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THE INFLUENCE OF ORIFICE INSERTION IN RADIAL SWIRLING FLOW FOR LOW EMISSIONS COMBUSTION

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ABSTRACT

This paper presents the effect of inserting swirler outlet orifice plate of different sizes at the exit plane of the flat blade radial air swirler in liquid fuel burner system. Tests were carried out with three different orifice plates with area ratios (orifice area to swirler exit area ratio) between 1.0 to 0.7 using 280 mm inside diameter combustor of 1000 mm length. Tests were conducted using commercial diesel as fuel. Fuel was injected at the back plate of the 60^o vane angle swirler outlet using central fuel injector with single fuel nozzle pointing axially outwards. The aim of the insertion of orifice plates was to create the swirler pressure loss at the swirler outlet phase so that the swirler outlet shear layer turbulence was maximised to assist with fuel/air mixing. In the present work, orifice plate with smaller area ratios exhibited very low NO_X emissions for the whole operating equivalence ratios. NO_X reduction of more than 15 percent was achieved for orifice with 0.7 area ratio compared to area ratio of 1.0. Other emissions such as carbon monoxide increased with decreasing in orifice area ratios. This implies that good combustion was achieved using smallest area ratios of orifice plate.

Keywords: *Radial swirler, orifice plate, NO_X reduction, CO emissions*

1.0 INTRODUCTION

Burners are usually used in industrial applications such as starters for boilers, district heating and cooling and also for domestic central heating system. However, conventional burners, operating at or above stoichiometric air/fuel ratios, produce high flame temperatures that resulted in the production of nitrogen oxides, which is then emitted to the atmosphere [1].

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Global environmental problems such as greenhouse warming, acid rain and hole in the ozone layer have become serious problems all over the world. Where acid rain is essentially a regional phenomenon, green house warming is a global problem and is difficult to solve. In recent years, an increasing awareness of the environmental impact of combustion devices has led to legislation concerning their exhaust emissions. The role of NO_X formation in ozone has been the subject of many recent debates. NO_X emissions from combustion devices would also deplete the stratospheric ozone layer and this would increase ultraviolet radiation to the earth's surface and with it the occurrence of skin cancer in the population [2].

However, lowering NO_X emission by reducing flame temperature will lead to reduced flame stability or increased in Carbon Monoxide (CO) emission [3]. Therefore, a method must be found that will be able to reduce the time for peak temperature and will reduce the formation of NO_X.

Basically there are two techniques of controlling NO_X in burner applications: those that prevent the formation of nitric oxide (NO) and those that destroy NO from the products of combustion. The methods that prevent the formation of NO involved modifications to the conventional burner designs or operating condition. In this research, the burner will be designed to incorporate swirling flow to enhance turbulence and hence helps in mixing of fuel and air prior to ignition. Swirling flow induces a highly turbulent recirculation zone, which stabilises the flame resulting in better mixing and combustion [4]. It has been suggested that the large torroidal 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 [5].

Although the importance of fuel and air mixing has been recognized, methods of controlling the mixing in burners have received relatively little attention. Considerable attention has been given to the extreme case of premixed-prevaporized combustion. However, it has been noted that the fully premixed combustion has several severe problems such as flame stability, auto ignition, combustion efficiency and stabilizer durability [6,7]. Therefore, attention has been switched to rapid fuel and air mixing rather than premixing. Rapid fuel and air mixing can be achieved by using swirlers. Swirl is an important feature of burner design since it provides satisfactory flame stability by creating a central reverse-flow zone. This is obtained when a sufficiently high degree of swirl is imparted on the combustion air stream. The recirculation zone is formed in both cold and combusting flows. The formation of such a flow pattern in the vicinity of the fuel injector results in stable combustion and rapid heat release. This is attributed to the enhanced mixing rates between the fuel and air and supply of energy required for ignition.

Swirling flow is a main flow produced by air swirled in gas turbine engine. Such flow is the combination of swirling and vortex breakdown. Swirling flow is widely used to stabilize the flame in combustion chamber. Its aerodynamic characteristics obtained through the merging of the swirl movement and free vortex phenomenon that collide in jet and turbulent flow. This swirl turbulent system could be divided into three groups and they are jet swirl turbulent with low swirl, high jet swirl with internal recirculation and jet turbulence in circulation zone. Each and every case exists due to the difference in density between jet flowing into the combustion chamber and jet flowing out into the atmosphere from the combustion chamber.

When air is tangentially introduced into the combustion chamber, it is forced to change its path, which contributes to the formation of swirling flow. The balance in force could be demonstrated by the movement of static pressure in the combustion chamber and can be calculated by measuring the distribution of the tangential velocity. Low pressure in the core center of the swirling flow is still retrieving the jet flow in the combustion chamber and thus, produces the not-so-good slope of axial pressure. Meanwhile at the optimum swirl angle, the swirl finds its own direction and as a result, swirl vortex is formed.

The recirculation region in free swirl flow is shown in Figure 1. Due to assumption that the flow is axially symmetrical, thus only half of the flow characteristics are discussed. The recirculation region is in the OACB curve. The point B is known as stagnation point. The flow outside of the OACB curve is the main flow, which drives the recirculation along the AB solid curve. The ultimate shear stress could happen at points near to point A, along the boundary of recirculation. The condition of zero axial velocity is represented by hidden curve AB. Every velocity component decreases in the direction of the tip. After the stagnation point, the reverse axial velocity will disappear far into tip; the peak of velocity profile will change towards the middle line as an effect of swirling decrease.

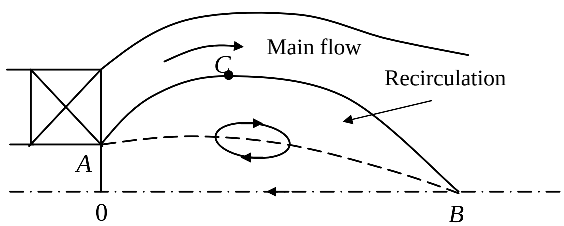

Figure 1: Recirculation zone in swirling flow

As the level of applied swirl increase, the velocity of the flow along the centerline decreases, until a level of swirl is reached at which the flow becomes stationary. As the swirl is increased further, a small bubble of internal recirculating fluid is formed. This, the vortex breakdown phenomenon, heralds the formation of large-scale recirculation zone that helps in stabilizing the flame. It has been suggested [8] that the large torroidal recirculation zone plays a major role in the flame stabilization 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.

In high velocity combustion system, the fuel and air mixing requires high turbulence levels and these result from the combustor pressure loss. Whether this pressure loss is generated by a jet flow system or swirl system, the air inlet aerodynamics generate shear layer which create the turbulence. In a conventional swirl burner the turbulence energy is mostly generated close to the central toroidal recirculation zone and is not fully utilised in an efficient way. In order to achieve enhanced flame stabilisation and better control of mixing process, a swirler shroud consisting of an orifice plate at the outlet of swirler throat was introduced. The aim of this was to create the main pressure loss at the outlet phase rather than in the vane passage so that the swirler outlet shear layer turbulence was maximised to assist with fuel and air mixing. Orifice plate insertion also helps to prevent fuel from entraining into the corner recirculation zone that will create local rich zone thus generates lower NO_X emission by eliminating locally rich region [10]. Locally rich region tends to generate locally high NO_X emission that contributes to overall high NO_X emission. Smaller orifice plate's outlet does increase the velocity of the air and fuel at the swirler shroud thus reduce the risk of flashback. However, this velocity should not be to high as lift off could occur and cause blow off of combustion. The increase in velocity also would increase the Reynolds number, which increases the strength of turbulence effect and thus reduces the combustions residence time. Other than that, from the aerodynamic factor, air and fuel mixing rate increases as the pressure drop in the swirler outlet increases.

The purpose of present approach was to investigate the significance of shroud swirler orifice plate assistance effect on swirling flame and their emissions and to exploit the additional mixing force for low NO_X combustor.

2.0 SWIRL NUMBER

The swirl number is usually defined as the fluxes of angular and linear momentum and it is used for characterising the intensity of swirl in enclose and fully separated flows. The parameter can be given as [5]:

$$S = \frac{G_{\phi}}{G_{\chi} . r_o} \tag{1}$$

where $G_ø$ is the axial flux of angular momentum:

$$G_{\phi} = 2\pi \int_0^{\infty} \rho U_x U_{\theta} r^2 dr \tag{2}$$

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

$$G_{\chi} = 2\pi \int_0^{\infty} \rho U_x^2 r dr + 2\pi \int_0^{\infty} p r dr \tag{3}$$

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

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:

$$S^{'} = \frac{G_\theta}{G_x^{'} r_o} \tag{4}$$

where

$$G_x^{'} = 2\pi \int_0^\infty \rho U_x r dr \tag{5}$$

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 [5] that the swirl number may be satisfactorily calculated from geometry of most swirl generator. According to [11] if a perfect mixing and conservation of momentum is assumed, then the swirls number can be defined in term of the geometry of the combustor.

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

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 (1989) and given as:

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

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 following Equation (5).

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

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

where

ΔP is the pressure drop

$\dot{m}$ is the volumetric air flow rates

The discharge coefficient, C_D is defined as:

$$C_D = \frac{1}{\sqrt{K_{th}}} \tag{9}$$

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:

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

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

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

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

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

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

3.0 EXPERIMENTAL SET-UP

The general rig set-up for the liquid fuel burner tests is shown in Figure 2. The rig was placed horizontally on a movable trolley. The air is introduced through the inlet pipe and flows axially before entering the combustor through the flat blade radial swirler of 50 mm diameter outlet and an orifice plate at the exit plane of the swirler outlet where the amount of air entering the combustor is controlled by the orifice plate minimum area. The equivalence ratio is varied in the combustion process by varying the primary air intake into the combustor.

Tests were conducted using comercial diesel as fuel. Fuel was injected at the back plate of the 60^o vane angle swirler outlet using central fuel injector with single fuel nozzle pointing axially outwards. Tests were carried out with three different orifice plates with diameter 45 mm, 40 mm and 35 mm that gave area ratios (orifice area to swirler exit area ratio) between 1.0 to 0.7 using 280 mm inside diameter combustor of 1000 mm length. The combustor was cooled by convection from the ambient air. The air entering the combustor was passed through a plenum chamber where the air swirler was installed at its exit plane and the fuel was introduced in this chamber. The exhaust sampling probe is mounted at the end pipe.

The gas analyzer used in these tests was the portable Kane May gas analyzer model Quintox 9106 that can measures oxides of Nitrogen, Unburned Hydrocarbon, Carbon Monoxide and Carbon Dioxide.

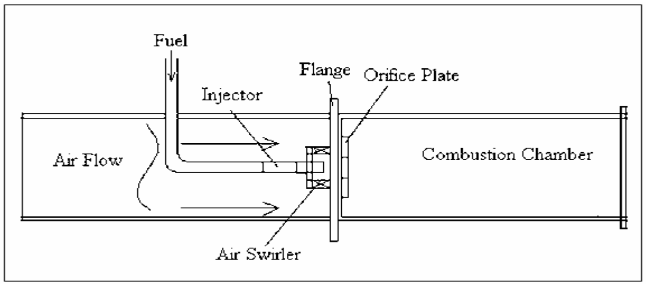

Figure 2: Experimental rig set up

4.0 RESULT AND DISCUSSIONS

4.1. Isothermal Performance

In order to achieve better mixing between fuel and air in 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 with orifice insertion, 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 for isothermal performance were plotted as a function of Reynolds number and presented in Figure 3.

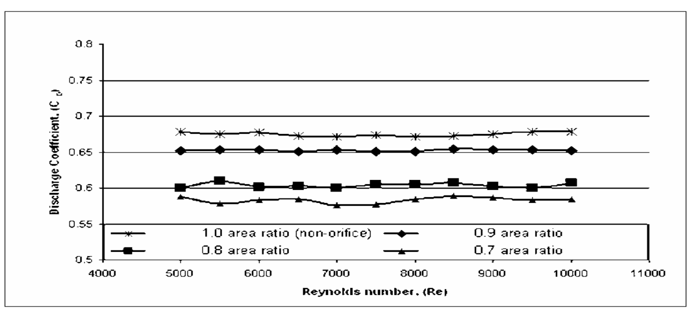

Figure 3: Discharge coefficient vs reynolds number for various area ratio, 60^{o} swirler vane angle

From Figure 3, it can be seen generally that all discharge coefficients were approximately constant with variation in Reynolds number. Thus the value of discharge coefficient may be concluded to be independent of Reynolds number. In the case of non-orifice plate (1.0 area ratio) for 60º vane angle swirler gave the highest C_D around 0.67. The C_D values were decreased with the decreasing in orifice area ratio, with the lowest area ratio, 0.7 having the C_D value of around 0.57. This may be attributed to the fact that the excessive swirl was generated by the restriction on swirler exit width.

4.2 Combustion Performance

Figures 4 to 7 show the effect of placing different sizes of orifice plate at the exit plane of the various vane angle radial air swirler outlet on exhaust emissions from burner system. Test on exhaust emission were carried out for 35 mm, 40 mm and 45 mm sizes of orifice plate diameter (0.7, 0.8 and 0.9 area ratio) using 60^o radial air swirler vane angle.

Figure 4 shows vast reduction in oxides of nitrogen (NO_X) emissions when smaller area ratio of orifice plate was used. This suggested that smaller orifice size enhances better mixing than the larger ones due to improve upstream mixing. This was apparent for the whole range of operating equivalence ratios. Emissions level of below 30 ppm was obtained for all range operating equivalence ratios. The graph clearly shows that placement of the orifice plate produces better emission reduction rather than without the orifice plate. For orifice plate with 0.7 area ratio, NO_X emissions reduction of about more than 15 percent was obtained at equivalent ratio of 0.833 compared to the air swirler without orifice plate at the same equivalence ratio.

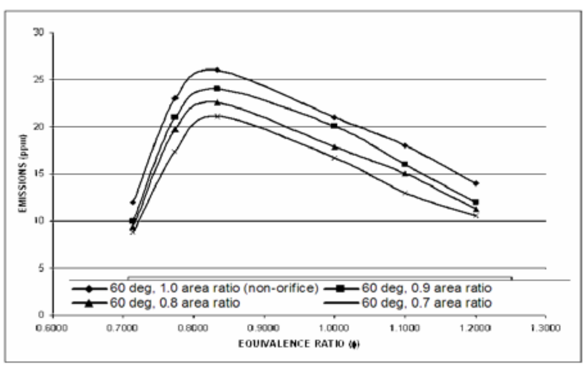

Figure 4 : NO_X emission vs equivalence ratio

Figure 5 shows Carbon Monoxide (CO) emissions versus equivalence ratio for all orifice plate diameters of 60^o air swirlers vane angle. Once again it is noted that smaller area ratio of orifice plate produce better emission reduction than larger ones. The emission reductions of carbon monoxide (CO) for 0.7 area ratio orifice plate was only 20 percent decrease at equivalent ratio of 0.833 compared to non-orifice assistance (1.0 area ratio) at the same equivalence ratio. However, this reduction could play a major role in pollution control. Figure 6 shows Carbon

Dioxide (CO_2) emissions versus equivalence ratio for all orifice plate diameters of 60^o air swirlers vane angle. There was a slight increase in carbon dioxide emissions when decreasing the area ratio. This was seen throughout the whole range of operating equivalence ratios. The decrease was very small compared to the reduction of NO_X emissions that was obtained. The increase of carbon dioxide emissions does not contribute to health problems, as carbon dioxide is more stable and non-toxic. However, CO_2 is a greenhouse gas and can contribute to global climate change.

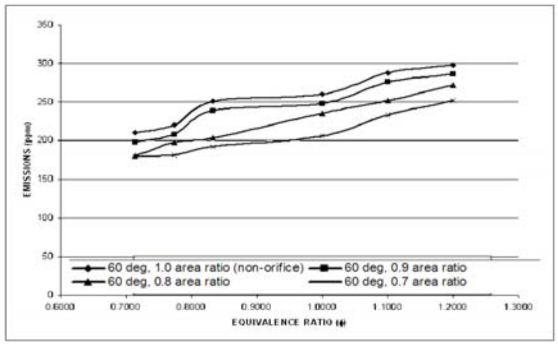

Figure 5: CO emission vs equivalence ratio

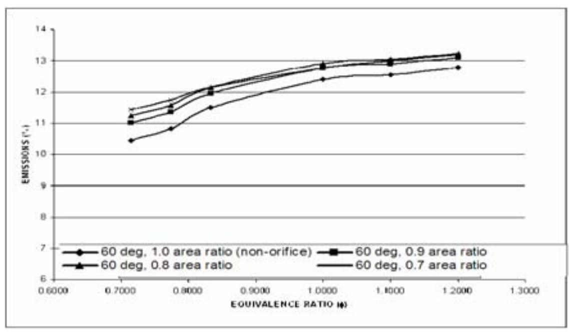

Figure 6: Percentage CO_2 vs equivalence ratio

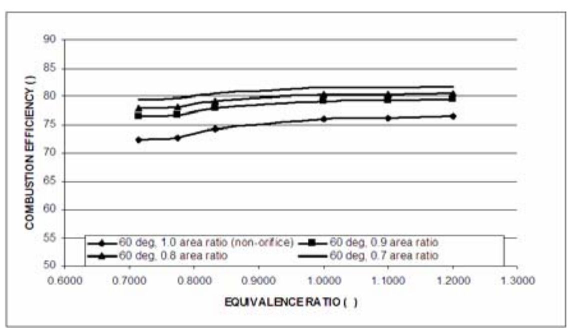

Figure 7: Combustion efficiency vs equivalence ratio

Figure 7 shows combustion efficiency versus equivalence ratio for all orifice plate diameters of 60^o air swirlers vane angle. Combustion efficiencies of greater than 70% were achieved throughout the whole range of operating equivalence ratios implying very good mixing of the fuel and air prior to ignition was achieved. This can be attributed to the insertion of orifice plate at the exit plane of the radial swirler.

5.0 CONCLUSIONS

The non-orifice plate (area ratio of 1.0) configuration showed the highest NO_X emission over a wide range of equivalence ratios suggesting that the insertion of the orifice plate helps in improving the mixing of fuel and air. NO_X emissions reduction of about 15 percent was obtained at equivalent ratio of 0.833 compared to the air swirler without orifice plate at the same equivalence ratio. Carbon monoxide (CO) for 0.7 area ratio orifice plate was only 20 percent decrease at equivalent ratio of 0.833 compared to non-orifice assistance (1.0 area ratio) at the same equivalence ratio. Other emissions such as carbon dioxide fluctuate constantly throughout the whole range of equivalence ratio. NO_X emissions of less than 30 ppm were achievable over the whole range of equivalence ratio.

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