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Self-Induced Recirculation Characteristics of a Swirl Burner

G.E. PLEASANCE

Research Engineer, State Electrictiy Commission of Victoria

SUMMARY Isothermal model tests conducted to investigate parameters influencing the self-induced external recirculation of hot combustion gases back to the throat of a swirl burner are described. Secondary air swirl, recirculation to secondary air duct radius ratio and recirculation flow circuit resistance were found to be the main variables which influenced the recirculation flow. An optimum combination of secondary air swirl and radius ratio was found to maximise the recirculation flow for a given secondary air supply pressure.

The results from the isothermal experiments were used to design a prototype burner and the recirculation flow predictions from the isothermal model verified during combustion tests on a model of this burner.

1 INTRODUCTION

Swirling jet burners are commonly used in oil, gas and black coal fired furnaces. Swirling secondary air admitted in the outer annulus of the burner generates a low pressure at the mouth of the burner which induces internal recirculation of hot combustion gases. Incoming fuel/transport gas/air mixtures admitted at the centre of the burner are deflected around the internal recirculation gases mixing with them to provide good conditions for stable intense combustion. The optimisation of the burner geometry and flow parameters has been established by extensive trials for oil, gas and black coal fuelled burners[1,2] and more recently for the relatively large central fuel pipe to outer annulus diameter ratio required to burn Victorian brown coal[3].

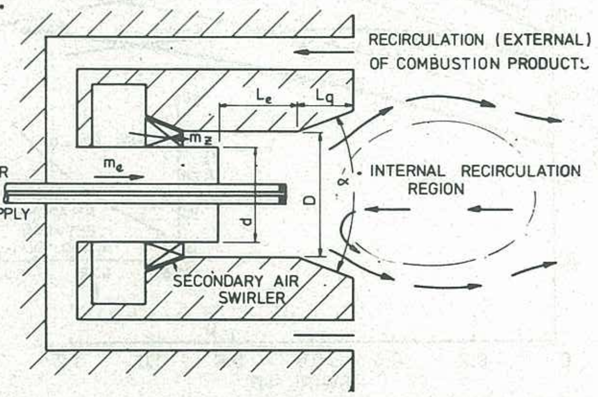

Figure 1 : External Recirculating Swirl Burner

Reducing the emissions of nitrogen oxides (NO_x) from combustion processes by controlling the combustion environment is the objective of much current research[4]. A reduction in NO_x emission without affecting carbon carryover has been achieved in small industrial oil burners by externally recycling hot combustion products back to the burner mouth. It was proposed to use this means of NO_x control for larger oil fired furnaces.

This paper presents the results of a model study to investigate the use of the low pressure developed at the throat of a swirl burner and the jet pump effect of the swirling secondary air to externally recycle hot combustion products (≈800°C) back to the burner mouth. Figure 1 shows the proposed burner together with the flow pattern which is characteristic of a swirl burner.

A preliminary isothermal model study was undertaken to determine the influence of burner geometry, secondary air swirl, burner radius ratio and recirculation duct resistance on the recirculation (external) mass flow and the characteristics of the internal recirculation region. The range over which the variables were tested was governed by the requirement to retrofit the burner to existing plant and to maintain its performance as an effective stable burner.

The results from the preliminary model tests were used to design a prototype burner and a one-half scale model of this burner was constructed for combustion trials. Isothermal tests were conducted on the latter model to verify the design and initial combustion tests were used to verify the scaling criteria.

2 PRELIMINARY AERODYNAMIC MODEL INVESTIGATION

2.1 Description of the Model

The test rig used in the preliminary model study is shown in Figure 2. The burner geometry is defined in Figure 1. The secondary air was supplied from

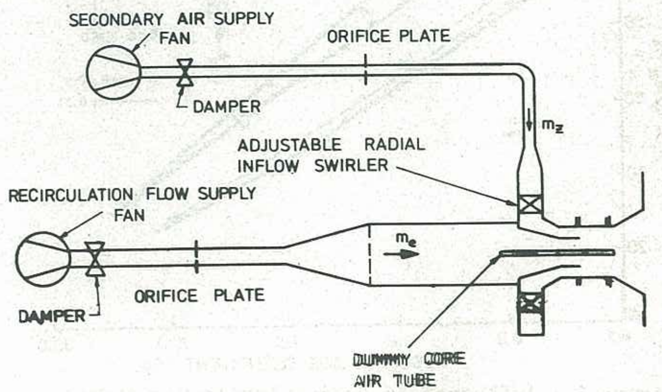

Figure 2 : Preliminary Model Test Rig

a high pressure fan to the outer annulus of the burner via an orifice meter and a radial inflow swirler. The swirl blade angle could be changed to vary the secondary air swirl. The recirculation flow was supplied from a second fan via a damper to control the recirculation circuit resistance and an orifice meter to measure the flow. The burner's core air and fuel gun tube was represented by a dummy tube.

The outer air annulus was terminated with a flange to allow various combinations of quarl geometry and cylindrical extensions to be attached for testing.

2.2 Tests Conducted

The recirculation to secondary air mass flow ratio was measured for

- no quarl ($\alpha = 0$) and quarl angles of 20^o, 30^o and 60^o;
- quarl length to diameter ratio (L_q/D) of 0.35 and 0.7;
- mixing length (L_e) between the secondary and recirculation flow of 0.2 D and 0.7 D (0 and 0.5D for no quarl);
- burner radius ratios ($R = d/D$) of 0.81 and 0.9;

over a range of secondary air swirls and recirculation circuit resistances. The influence of these parameters on the size and relative strength of the internal recirculation flow was determined from a tuft survey.

The air swirler was calibrated to give the swirl number (S) as a function of the swirler blade angle. The swirl number is defined as the ratio of the axial flux of angular momentum to the product of the axial flux of axial momentum and burner throat radius (D/2).

2.3 Results and Discussion

2.3.1 Recirculation circuit resistance

The total pressure loss coefficient of the recirculation circuit is defined as

$$C_p = \frac{P_i - P_e}{\rho_e v_e^2/2} \qquad (1)$$

where P_i is the total gauge pressure at the inlet to the recirculation duct, equal to zero in the test rig but equal to the furnace pressure in a boiler; P_e is the total gauge pressure at the exit from the recirculation duct; and $\rho_e v_e^2/2$ is the velocity head at the exit from the recirculation duct.

In a boiler, pressure loss in the recirculation circuit due to possible screening of the inlet from the furnace by furnace wall tubing and the 180^o bend required to direct the flow back to the burner throat will be unavoidable. Figure 3 shows that significant increases in the recirculation flow can be achieved by minimising this pressure loss.

Increasing the recirculation flow by reducing the recirculation circuit resistance decreased the length and strength of the internal recirculation region.

2.3.2 Secondary air swirl

Figure 4 shows the influence of secondary air swirl on the recirculating flow. Increasing the swirl from zero to a swirl number of 1.2 approximately doubled the recirculation flow with the incremental increase reducing at higher swirl numbers. As the required secondary air wind-box pressure increases with the square of the swirl number, swirl numbers beyond 1.2 could not be justified in a full size burner to obtain further increase in the recirculating flow.

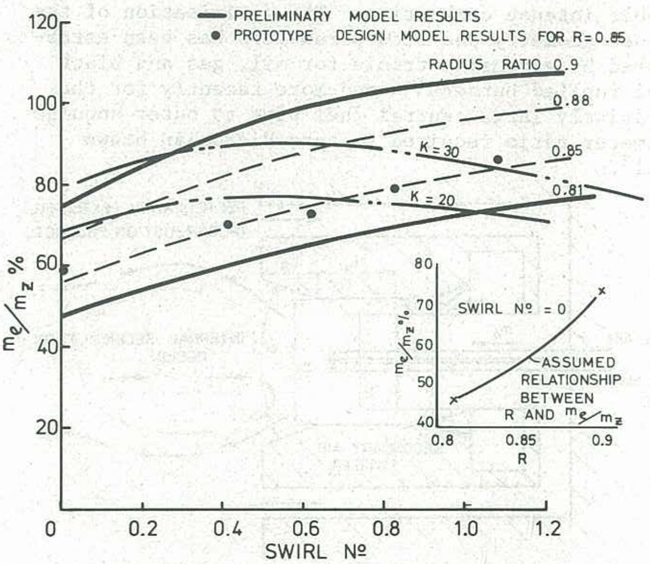

Figure 4 : Influence of Swirl in Recirculation Flow

Increase in swirl whilst maintaining a constant recirculation flow increased the length of the internal recirculation but did not appear to increase its strength. However, with a constant recirculation circuit resistance the increase in recirculation flow with swirl resulted in a decrease in the internal recirculation strength. This corresponds to results from isothermal model measurements made by the State Electricity Commission of Victoria[3] on a swirl burner of similar radius ratio. These showed that the internal recirculation flow decreased as the inner flow increased and that secondary air swirl had no influence on the internal recirculation mass flow.

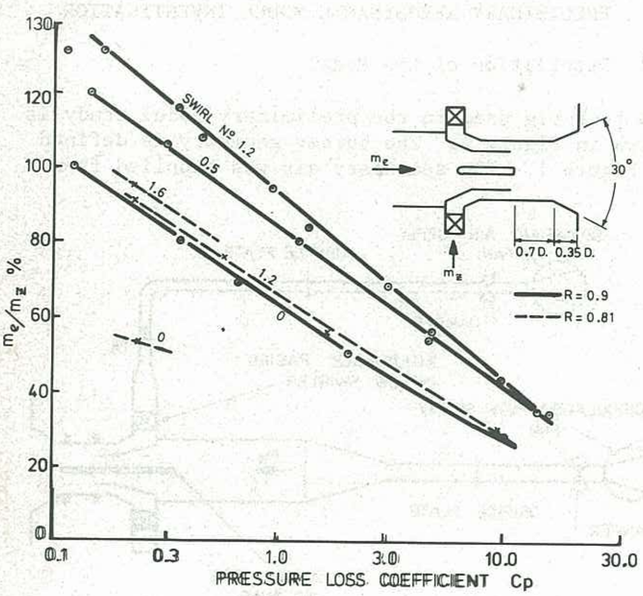

Figure 3 : Influence of Recirculation Resistance on Recirculation Flow

2.3.3 Quarl angle and quarl length

The influences of quarl angle and quarl length on the recirculation flow are shown in Figure 5. For the quarl length of $L_q/D = 0.35$ (0.4 for the 60^o quarl) an optimum quarl angle of between 20^o and 30^o is indicated. Increasing the quarl length to 0.7 D with the 20^o quarl increased the recirculation flow by up to 5 percentage points.

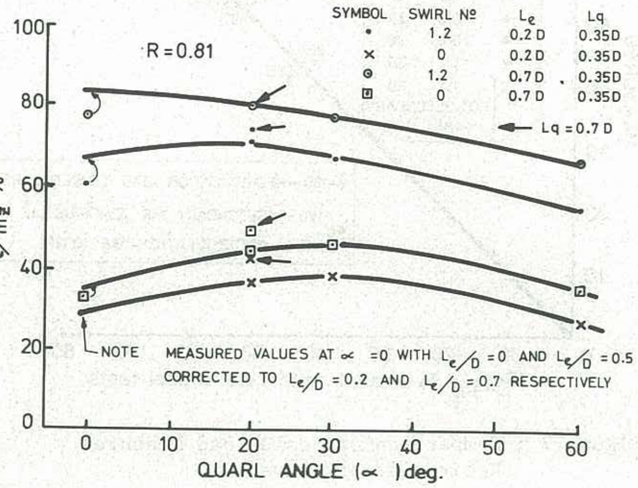

Figure 5 : Influence of Quarl Angle and Quarl Length on Recirculation Flow

The size and strength of the internal recirculation region increased with quarl angle. With no quarl and a high recirculation flow (high swirl) the internal recirculation region was small and turbulent with virtually no recognisable mean flow pattern.

2.3.4 Mixing length

Figure 6 shows that increasing the mixing length from $L_e/D = 0.2$ to 0.7 increased the recirculation flow by approximately 20%. It would be expected from data on ejector performance[5] that the incremental increase in entrained flow would decrease with further increase in mixing length.

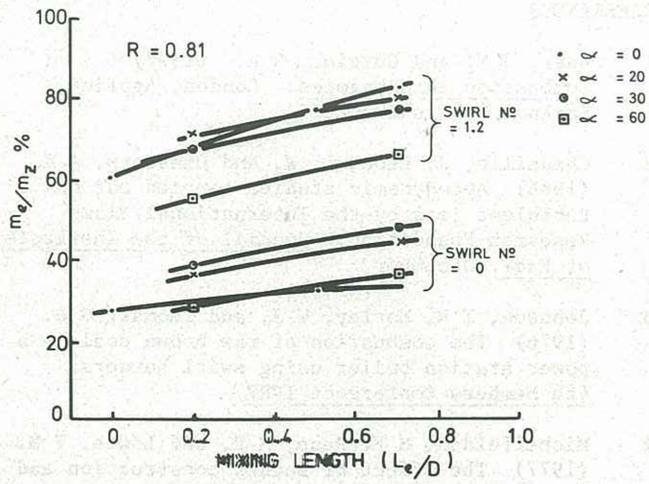

Figure 6 : Influence of Mixing Length on Recirculation Flow

2.3.5 Radius ratio

Increasing the radius ratio from 0.81 to 0.9, and consequently the secondary air velocity by a factor of 1.8 for constant secondary air mass flow, increased the recirculation flow by between 25 and 30 percentage points for the cases tested (see Figures 3 and 4). However, as with increasing swirl, the required wind-box pressure is increased, in this case by a factor of 3.2.

For equivalent recirculation flow rate, geometry and swirl, the larger radius ratio appeared to give a stronger internal recirculation.

3 DESIGN OF A PROTOTYPE BURNER

3.1 Optimisation of Radius Ratio Against Secondary Air Swirl

Both radius ratio and secondary air swirl have a significant influence on the recirculation mass flow at the expense of secondary air supply pressure. For a free vortex superimposed on a uniform axial flow the mean velocity head in the secondary air annulus is

$$\frac{\rho_z}{2} q^2 = \frac{\rho_z u^2}{2} \left(\frac{2S^2}{1-R^2} \ln \frac{1}{R} + 1\right) \qquad (2)$$

where ρ_z is the secondary air density, u is the secondary air axial velocity, and q is the secondary air absolute velocity.

For all cases tested this velocity head was approximately equal to the pressure difference between the secondary air wind-box and furnace. For a given wind-box to furnace pressure difference, secondary air temperature, burner diameter and secondary air flow applicable to a particular burner retrofit, equation 2 can be expressed as

$$\frac{\rho_z q^2}{2} \cdot \frac{2\rho_z}{m_z^2} \cdot \left(\frac{\pi D^2}{4}\right)^2 = \frac{1}{(1-R^2)^2} \left(\frac{2S^2}{1-R^2} \ln \frac{1}{R} + 1\right)$$

$$= \text{constant } K \qquad (3)$$

Figure 4 shows lines of constant K superimposed on the experimental results of swirl versus recirculation flow for radius ratios of 0.81, 0.9 and interpolated intermediate radius ratios. An optimum radius ratio and swirl can be selected for given burner design conditions although the recirculation mass flow is not greatly dependent on this optimisation.

3.2 Scaling Model to Prototype

Dynamic similarity between the isothermal model and the prototype burner will be achieved if the axial momentum flux ratios between the secondary and recirculation flow, the swirl numbers and the recirculation duct pressure loss coefficients (the Reynolds number in this duct was sufficient in both to ensure turbulent flow) are equal. Therefore for geometrically-similar model and prototype operating with equal secondary air swirls

$$\left.\frac{\rho_e v_e^2}{\rho_z v_z^2}\right| \text{model} = \left.\frac{\rho_e v_e^2}{\rho_z v_z^2}\right| \text{prot}$$

and hence the prototype recirculation to secondary air mass flow ratio is

$$\left.\frac{m_e}{m_z}\right| \text{prototype} = \left.\frac{m_e}{m_z}\right| \text{model} \left.\sqrt{\frac{\rho_z}{\rho_e}}\right| \text{prototype} \qquad (4)$$

For conditions expected in the proposed operating burner (secondary air temperature 30^oC and recirculation gas temperature of 800^oC) the prototype recirculation mass flow ratio would be approximately half that measured in the model.

3.3 Burner Geometry

The size of the prototype burner was governed by restraints of the proposed retrofit installation. Locations of neighbouring burners and positioning of the recirculation ducts limited the burner diameter; a requirement that the oil gun be removable dictated the clearance required behind the burner and limited the mixing length; the possibility of combustion in the quarl and the consequent flame resistance due to the momentum sink across the flame front restricting the recirculating flow limited quarl length; and requirements for a flame size compatible with the furnace but at the same time with sufficient internal recirculation for ignition stability set the upper and lower limits of secondary air swirl.

Within these restraints a prototype burner similar to that shown in Figure 1 was designed for a secondary air pressure of 2.5 kPa. The burner parameters were D = 600 mm, $L_e/D = 0.67$, $\alpha = 30^o$, $L_q/D = 0.33$ and R = 0.85 with a swirl number of 0.6.

4 ISOTHERMAL AND NON-ISOTHERMAL TESTS OF THE PROTOTYPE BURNER DESIGN

A 1/2 geometric scale model of the prototype burner was built for combustion tests to evaluate the burner's potential to reduce NO_x formation. Isothermal tests on this model to calibrate the secondary air swirler and to calibrate the 180^o bend in the recirculation duct as a flow meter for combustion tests, were also used to determine the influence of core air (which enters the burner mouth via a central tube and swirler) on the recirculation flow and to verify the design. Core air quantities of up to 15% of the total air flow to the burner did not influence the recirculation to total (core plus secondary) mass flow ratios. The measured recirculation flow rates are superimposed on Figure 4 and are seen to agree well with the first model results.

Discussion of the combustion tests are beyond the scope of this paper. However, during the initial combustion tests the scaling relationship (4) was checked by measuring the recirculation gas to total air mass flow ratio both whilst the combustion chamber was cold ($\rho_e = \rho_z$) and immediately following combustion tests whilst the recirculation gas was still hot ($\rho_e < \rho_z$). Tests were also conducted with preheated secondary air. The results shown in Figure 7 confirm the scaling criteria.

5 CONCLUSIONS

Isothermal model tests have shown that relatively large recirculation mass flows can be achieved in a swirl burner by optimising the mixing length, secondary air swirl and burner geometry. The recirculation mass flow is limited in practice by recirculation duct resistance, available air supply pressure and the available space into which the burner is to be installed.

These results have been used to design a self-recirculating burner for a commercial installation and isothermal tests on a one-half, geometrically-scaled model of this burner have verified the design objectives. Non-isothermal tests have confirmed that the model results can be scaled to prototype conditions for geometrically-similar burners by maintaining equivalence of the recirculation gas to secondary air momentum flux ratios.

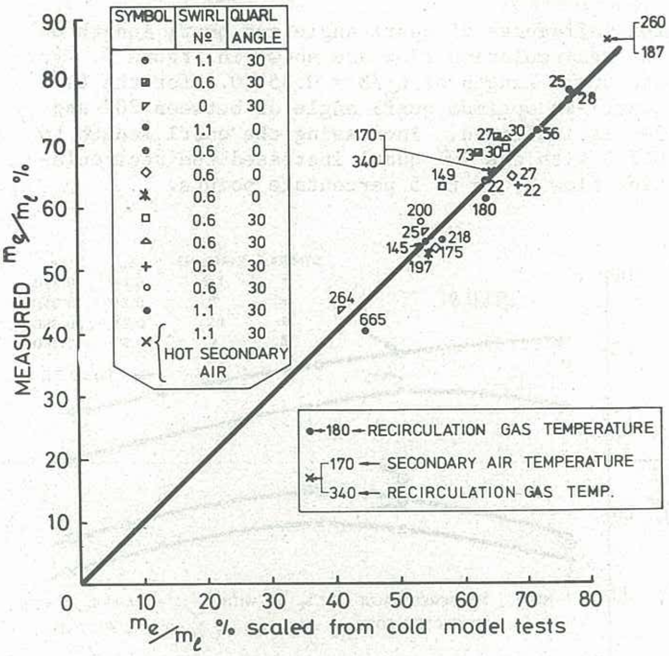

Figure 7 : Comparison of Scaled and Measured Recirculation Flows

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