20th International Congress of Chemical and Process Engineering CHISA 2012
25 – 29 August 2012, Prague, Czech Republic

Experimental determination of the swirl burner laboratory models hydraulic resistance

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Abstract

Performed investigation of pulverized fuel combustion in swirl flows has shown that there are two important groups of influential parameters: fuel and burner characteristics. The most important conclusion of these investigations is that each type of fuel needs a defined burner for its combustion and concrete operating parameters for its optimal work. Optimal operating parameters can be determined only by performing the appropriate experiments. Selection of optimal swirl burner design is dependent on energy loss which is used for air transport through burner. Increasing of swirl intensity is convenient for fuel ignition and flame stability, but energy loss for air transport increases, too. Swirl economy is estimated using hydraulic resistance parameter which represents energy loss due to swirling. In the paper are presented the methods and experimental data of hydraulic resistance determination for three laboratory models of swirl burners for pulverized fuel combustion. In the tangential burner the swirling is accomplished by tangential inflow of secondary air and fuel/air mixture. Axial-blade burner of type 1 enables swirling by a set of 18 blades. Axial-blade burner of type 2 is with a central lead of gas and coaxial lead of fuel/air mixture, primary air and recirculating gases. In movable swirl block burner the secondary air swirling is enabled by 8 stationary and 8 movable blocks fixed at movable and stationary plate.

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1. Introduction

Combustion of pulverized fuel with swirl burners utilizing the swirl effect is very significant and widespread. Swirl flows are formed as a result of inducing the proper rotation motion to current flow such as: guide vanes, swirl device or otherwise. This method of combustion is used extensively because of their favorable influence on the stabilization and intensification of the combustion process. In addition to increase the efficiency of combustion, swirl flows provides an environmentally cleaner fuel combustion by reducing the forming of harmful pollutants, oxides SOx, NOx, CO_2 and others.

Swirl burners are widely used in furnace technique for different types of combustion of fossil fuels. These burners are installed in the energy and industrial boilers and furnaces, furnace in cement kilns, brick yards and lime kilns, metallurgical furnaces and in the combustion chambers of gas turbines, gas-turbine engines, both petrol and diesel engines and other devices [1]. The complex processes that occurs in non isothermal stream which is getting out from the burner prevents the establishment of calculation according to associate the structural parameters of the burner with the aerodynamic characteristics of the flame. It is impossible to establish universal criteria for defining cost-effectiveness and safety of the burner which makes selecting the optimal burner design. Therefore, the experimental determination of this dependence is the best way for reliable burners calculation.

Swirling flows are commonly used in processes with and without combustion. In engineering set-ups with combustion swirling flows are commonly used for stabilizing and intensifying the combustion process. Also, they are used for environment protection from pollutant oxides SO , NOx, CO etc. The main effect of swirl is to improve the flame stability. When rotating motion is imparted to a fluid upstream of an orifice, the fluid flow emerging from the orifice has a tangential component of velocity besides the axial and the radial components. The presence of swirl induces radial and axial pressure gradients which affect the flow field. In the case of strong swirl, reverse axial pressure gradient becomes too large and produces a reverse flow-internal recirculation zone (RZ). This zone which recirculates heat and active chemical species to the root of the flame has a great influence on stabilization of the combustion process [2].

RZ formation, dimensions and shape depend on several factors: swirl intensity, presence of a central pipe, geometrical burner and furnace characteristics, presence of combustion in the flow, blade shape etc. Performed investigations have shown that swirl exerts a strong influence on the aerodynamic processes within the flow. The most commonly used parameter for characterizing the swirling flows is the swirl number (S), a non-dimensional parameter. Investigations have shown that the swirl number is a significant similarity criterion for swirling flows.

Aerodynamic characteristics of swirling flows are different for weak (S<0,6) and strong (S≥0,6) swirl. In swirling systems with weak swirl the jet spread is stronger, but they are not of extensive use in practice. In the strong swirl case the axial pressure gradients cause internal recirculation, and they have a wide practical use.

2. Swirl burner models for pulverized fuel combustion

Performed investigation of pulverized fuel combustion in swirl flows has shown that there are two important groups of influential parameters: fuel and burner characteristics. The most important conclusion of these investigations is that each type of fuel needs a determinate burner for its combustion and concrete operating parameters for its optimal work. Optimal operating parameters can be determined only by performing the appropriate experiments.

To enable detailed study of the combustion process of pulverized fuels (biomass, coal etc.) using swirl burners of different types and strengths, and compare the results of research in laboratory and real conditions fireplaces, Laboratory for Thermal Engineering and Energy Institute of Nuclear Sciences Vinca made four models of swirl burners with heat power of 0.1-0.7 MW. Their basic design and operating characteristics are given in Table 1. For experimental tests of burner model was used laboratory facility, which allows the implementation of pulverized fuel combustion test in a wide range of operating parameters. This facility can conduct the following tests: determination of the optimal area of the burner for the selected type of fuel, to determine the effect of fuel quality, fineness of grinding, moisture content, ash and volatile matter in pulverized fuel combustion, determining the influence of temperature on combustion and determination of additional parameters essential for the vortex burner construction and design.

Table 1. Basic design and working characteristics of the burner model

	TG	ALG1	GB	ALG2
Burner geometry				
Gas tube diameter (mm)	-	-	-	30/23,6
Central tube diameter (mm)	60.2/52.7	60.2/52.7	42.4/37.2	63.5/57.1
Primary tube diameter (mm)	103/92	103/92	60.3/54.5	108/100
Secondary tube diameter (mm)	160/145	160/145	95.0/87.8	146/138
Diffuser length (mm)	320	320	100	160
Diffuser half angle ($^\circ$)	33	33	33	26.6
Blade angle of secondary air ($^\circ$)	-	30-75	0-90	15-75
Units				
Primary air velocity, wp (m/s)	1.8-4.5	2.3-9.4	5.3-23.4	2.3-4.5
Secondary air velocity, ws (m/s)	3.4-6.6	2.7-9.4	10.5-25.6	6-44.9
Central air velocity, wj (m/s)	1.4	-	-	-
Ratio, ws / wp (-)	2-4	0.53-3.68	0.6-3.42	1.8-14.1
Ratio, Gp / Gs (-)	0.15-0.40	0.16-0.89	0.09-0.52	0.09-0.73
Momentum ratio, M_R (-)	0.056-0.194	0.054-1.77	0.039-0.869	0.004-0.28
Swirl number, S (-)	3.0-6.4	0.41-1.10	0.12-1.12	0.55-7.50
Reynolds number, Re (-)	0.17×10^5	0.25×10^5	0.44×10^5	0.84×10^5
Fuel consumption, Bg (kg/h)	27-150	20-140	19-130	20-140
Burner power, Qg (KW)	110-774	172-635	106-720	89-621

TG – Tangential burner; ALG1 – Axial-blade burner type 1; GB – Movable block burner; ALG2 - Axial-blade burner type 2

The most reliable and simplest way to carry out the examination of various fuel and burner types is on experimental facilities. The basic conveniences of utilizing these facilities in comparison to experiments on real plants are [3]: easier and more efficient experimental work, lower cost of research, possibility of variation and detailed examination of main parameters in wide ranges, possibilities of real processes simulations are not lessened, modelling of processes, operation of industrial facilities is not hampered, possibility to use the pulverized fuel with same characteristics etc.

In designing the experimental facilities it is necessary to satisfy certain conditions [3]: to ensure temperature conditions of process modelling, equality of volume heat flux, to use fuel of the same characteristics etc. In accordance with these conditions the experimental facility was designed.

Four models of swirl burners were made: tangential, axial-blade type 1 and type 2 and with movable blocks (Fig. 1). In the tangential burner model (Fig. 1a) the swirling is accomplished by tangential inflow of secondary air through the annular channel (3) and fuel/air mixtures through the channel (2). On the front side are positioned movable blades (4) that, if necessary, are inserted to regulate the overall level of swirl in the burners. The burner has been designed in such a way that particle residence time in the flame is equal, compared to a real burner, while velocities are several times lower.

Axial-blade model of swirl burner type 1 (Fig. 1b) enables swirling by a set of 18 moveable blades (4). The fuel-air mixture is introduced at an angle of 30^0 and the output is set round with straight blades (5) to reduce disparities in the concentration of the mixture. The central tube (1) provides supply of central air and storage of oil burners. Modelling was performed as for tangential burner.
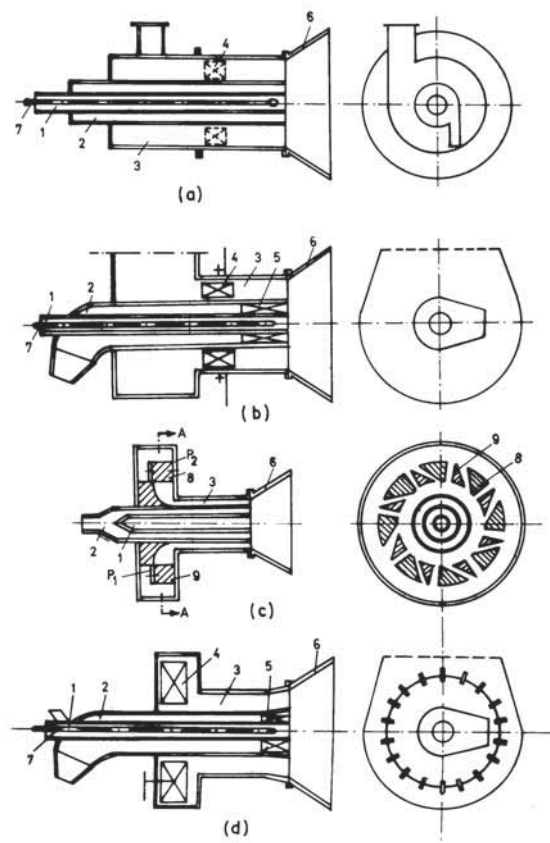

Fig. 1. Swirl burner models: a) tangential; b) axial-blade type 1; c) with movable blocks; d) axial-blade type 2
1-central air, 2-primary air, 3-secondary air, 4,5-blade vane, 6-diffuser, 7-gas burner,
8,9-stationary and movable blocks, P_1, P_2 -movable and stationary plate.

Axial-blade model of swirl burner type 2 (Fig. 1d) was prepared so that they prefer the existing design of swirl burners on boilers installed by some local boiler company. By its construction it is swirl type

burner, with bringing gas from the central ring and bringing the fuel-air mixture around the central tube of air. Mixture passes through the tube of a mixture of pulverized fuel, primary air and gas recirculation. It is anticipated wheel whirling mixture of 16 blades wreath. There are three sets with angle of the blades of 15, 30 and 45^0. The secondary air is introduced around the pipe of the mixture. Swirling of the secondary air is realized with a wreath of 16 blades with a mechanism for the continuous change of angle of the blades.

Movable swirl block burner (Fig. 1c) has been designed in such a way that the velocities of pulverized fuel/air mixture flow and secondary air can vary from the lowest to the velocities used in industrial burners. The secondary air swirling is enabled by 8 stationary and 8 movable blocks fixed at P_1 movable and P_2 stationary plate. The appropriate mechanism to enable the continuous movement of blocks and thus the light changes the level of swirl.

In designing the model and prototype burners are used appropriate similarity criteria [4]. The basic criterion for burner modeling was used retention time of particles in the flame. The condition of the same residence time of particles in the flame is obtained the ratio of air velocity in the model and prototype burner in inverse proportion to their relative diameters. Other conditions are the equality of modeling: length of the jet, power burner and heat input capacity of the flame. Meeting some criteria causes a form of flame. Also it is taken into account the geometric similarity model and the prototype burner.

3. General characteristics of swirl flows

3.1. Efficiency of swirl flows

At choosing of the type of swirl in burner one of the key criteria is its efficiency. Loss of power in the burner and the pressure that must be achieved at the entrance of the swirl is one of the characteristics of the burner of practical importance. It is clear that only a portion of available energy of the pressure exceeds the kinetic energy of the formed swirl flow, while the rest of the energy is lost by the action of viscous forces. The kinetic energy at the exit of the burner is [4]:

$$E_k = G u^2 (1 + \delta S^2) / 2 \tag{1}$$

where: G - mass flow of air through the burner, u - the average axial velocity in the outlet section of the burner; δ - coefficient depending on the type of swirl, on the ratio between tube diameter in the output section of the burner and on the axial velocity distribution, if is not uniform; S – burner swirl level.

Efficiency of swirl flows is evaluated by coefficient of kinetic energy flow in the form of annular flow depending on the geometrical characteristics of the swirl burner and ways of swirl. Also, efficiency of swirl flow is usually defined [4, 5] as the ratio of the kinetic energy flow through the outlet section of the burner and the static pressure drop through the burner. This parameter is used for assessing the formation of swirl intensity and is not a measure of efficiency of flow. At the same time swirl intensity of the various swirl devices can form different types of flow. Therefore, in practice, the parameter S is used for flow formed by axial and tangential swirl devices, and the angle of the blades β for blade devices.

3.2. The coefficient of hydraulic resistance of the burner

Selection of the optimal burner design depends on the loss of energy used to "blow" air through the burner. With increasing of swirl intensity enhances the security of fuel ignition and flame stability, but growing energy losses in the blowing air. Therefore, in choosing the optimum swirl intensity or swirl

device design must take into account both factors. Cost-effectiveness of swirl burner is assessed using hydraulic resistance burners coefficient ζ which is determined by the expression [5, 6]:

$$\zeta = 2\Delta p_{st} / (\rho w^2) + (Ae / A_i)^2 \qquad (2)$$

where: Δp_{st} - static pressure drop between the inlet section of the channel (Ai) and the output section of the duct behind the swirl device (Ae); ρ - air density; w - velocity at the exit of the channel (Ae).

The coefficient of hydraulic resistance of the burner reflects energy losses due to turbulence and experimental way can be related to the swirl intensity. Performed experiments Shagalova et al. [5, 6, 7] have shown that it is not possible to generalize the dependence of S and ζ for all types of swirl device.

In Fig. 2 is showed a dependence ζ of S in the auto model (self preserving) field of Re number for different swirl types [5, 6]. For most types of burners for pulverized fuel auto modelity (self preserving) performed at Re = $w*d_{hidr} / \nu \leq 2x10^5$, where: d_{hidr} – burner hydraulic diameter. In the non auto modeled area resistance coefficient ζ increases. The ratio of actual coefficient ζ_ϕ and coefficient ζ in auto modeled field is shown in Fig. 2b. Of the surveyed construction of swirl device least resistance ζ at the same S has an axial-blade swirl device with profiled blades.
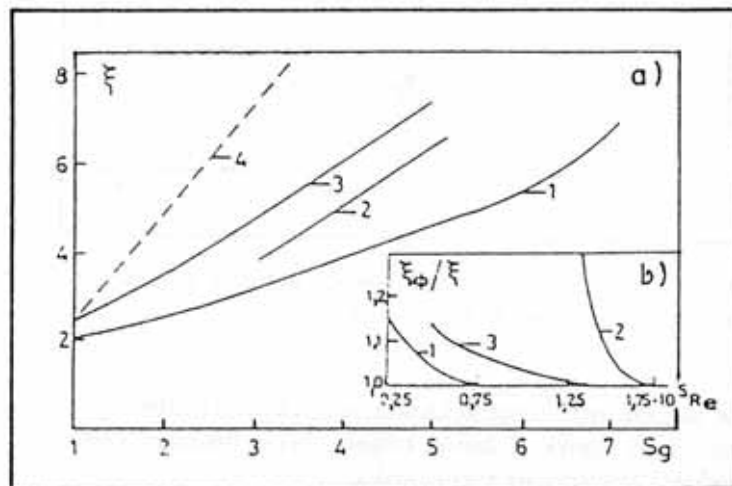

Fig. 2. Hydraulic resistance coefficient in auto model (self preserving) region (a) and correction for non auto modeled (b)
 for different swirling types: 1-axial with profiled blades, 2-with entrance device, 3-tangential with strait blades,
4-axial with strait blades.

3.3. Swirl intensity (degree) for burner models

Since the hydraulic resistance to a large extent depending on the swirl intensity of burner, it will be analyzed shortly the way of determining the swirl intensity of burners for different burner models. In swirling free jets, isothermal or with combustion, applies the law of conservation of momentum. Therefore, can be writing that:

$$G_\phi = \int_{r_1}^{r_2} (wr)\rho u 2\pi r dr = const. \tag{3}$$

$$G_x = \int_{r_1}^{r_2} u\rho u 2\pi r dr + \int_{r_1}^{r_2} p 2\pi r dr = const. \tag{4}$$

where: G_ϕ - the flux of angular momentum in the axial direction, G_x - the flux of momentum in the axial direction; u and w axial and tangential velocity component at a radius r; p - static pressure; r_1 and r_2 - the radial border of the outlet cross section of burner.

Maintaining momentum is the aerodynamic characteristics of the development swirl flow, and as a dimensionless criterion of this type of flow Beer and Chigier [4] proposed a swirl intensity, so-called swirl number defined as the ratio:

$$S = G_\phi / (G_x r_g) \tag{5}$$

where rg – exit radius of the burner nozzle. A number of studies analyzed in [1] showed that the swirl number is an important criterion of similarity of swirl flows, caused by the geometric similarity of the way of swirl, and that the other criteria of similarity, whether it is on the isothermal flow or combustion, can be applied in conjunction with the swirl number.

Calculation of swirl number by using the expression (3)- (5) requires knowledge of, or measurement, velocity and static pressure profiles at the exit of the burner, at the jet cross section. In most cases it is impossible to do, so the swirl number is determined using the known burner design data.

Axial-blade burner models: For the multi-component coaxial flow which is exit from burner channel the swirl number is calculated as the integral, according to the expression:

$$S_g = (\sum_{i=1}^{n} S_i D_{ei} \rho_i A_i w_i^2) / (\sum_{i=1}^{n} D_{ei} \rho_i A_i w_i^2 D_{eg}) \tag{6}$$

where: S_i - appropriate flow stream swirl number; A_i - area of the i-th channel; ρ_i - coaxial flow density; w_i – velocity of coaxial flow; D_{ei} - the equivalent diameter of the i-th coaxial channel; D_{eg} - the equivalent diameter of the burner according to the output section of the burner. Swirl burner model ALG1 hasn't primary flow swirl, so that $S_p = 0$, and the swirl number is calculated as:

$$S_g = (1.7397 * tg\beta_s) / (1 + 0.4645 (\rho_p w_p^2 / \rho_s w_s^2)) \tag{7}$$

For a burner model ALG2 which has primary flow swirl the swirl number is calculated as:

$$S_g = (S_p D_{ep} \rho_p A_p w_p^2 + S_s D_{es} \rho_s A_s w_s^2) / ((\rho_p A_p w_p^2 + \rho_s A_s w_s^2) D_{eg}) \tag{8}$$

where: $S_p = 1.89 * tg\beta_p$ and $S_s = 3.6 * tg\beta_s$.

Tangential burner model: For this burner model the swirl number is calculated depending on the geometric dimensions of the burner [7].

Movable blocks burner: For the burner with blocks which containing un swirled primary and secondary swirl flow, swirl number is equal to:

$$S_g = (0.83 * S_s) / (1 + 0.288 (\rho_p w_p^2 / \rho_s w_s^2)) \tag{9}$$

where the secondary air swirl number is determined depending on the axial width of the channel, the number of blocks and geometric characteristics of the burner.

4. Experimental results

Experimental determination of hydraulic resistance performed for all four laboratory models of swirl burners. Change of the burner operating parameters was performed within the plant boundary features. The coefficient of hydraulic resistance was calculated according to the expression (2) whereby certain size determined as follows:

- Δp_{st} (Pa): pressure drop between the inlet and outlet section of the burner was measured using digital micro manometer,

- ρ (kg/m3): air density was determined from the appropriate table based on the measured air temperature,

- w (m/s) air velocity at the outlet section of the burner was calculated based on the measured flow (using a measuring aperture) and air temperature,

- Ai / Au (-): the ratio of surface area of the output and input section of burner was: for the burner with blocks - 0.653, tangential burner – 14.39; axial-blade type 1 burner - 1.447 and for the axial-blade type 2 burner - 0.992.

Experimentally determined values of the hydraulic resistance coefficient of burner models are shown in Fig. 3-6. For the movable blocks burner (Fig. 3) was determined the correlation of coefficient ζ and the burner swirl number Sg for more air velocities, i.e. for more secondary air flow. Measurements showed that on the coefficient ζ have a similar impact change of the swirl intensity and air velocity, i.e. to their increase resistance value of burner increases. The high value of the coefficient ζ from 8-12 was obtained because, the blocks which fulfill the air swirl are the major resistance to air flow and cause considerable loss of kinetic energy flow.
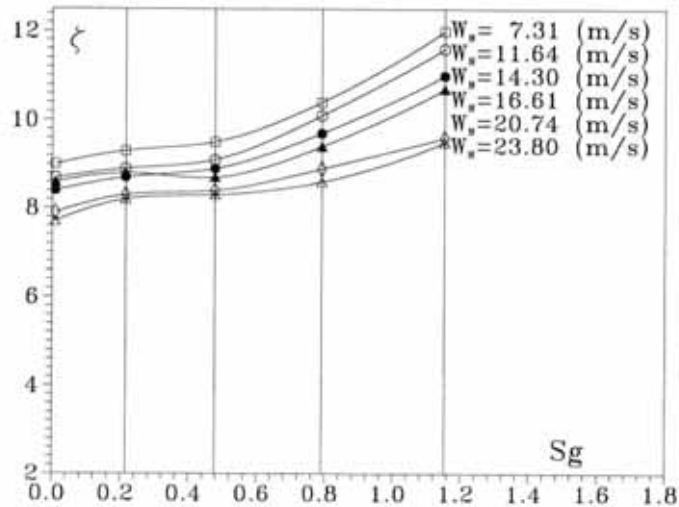

Fig. 3. Hydraulic resistance coefficient for movable blocks burner model.

In Fig. 4, for the tangential burner, is shown the change in the coefficient ζ depending on the velocity i.e. secondary air flow. The values were obtained at approximately constant value of the burner swirl

number Sg = 5.0 because this type of burner has not the ability noticeable change in the intensity of air swirl intensity. It may be observed weak dependence of the coefficient ζ change of air velocity. The high average value of the coefficient ζ of $\approx 20{,}7$ is the result of a small section of the input channels for the introduction of air conditioned to the possibilities of bringing air at the pilot plant.

   For axially-blade burner type 1 was performed determination of the coefficient ζ (Fig. 5) for the three angles (position) of secondary air blades β of 30, 45 and 60^0. Measurements have been made for the case of un swirled primary air (Sp = 0) and for the constant velocity of primary air $w_p = 5.5$ m/s. Measurements showed that the angle of the blades of secondary flow significantly influence the change of coefficient ζ, but also that this model of burner has a huge loss of energy to blow air through the burner. By this, it is obtained the following average values of the coefficient ζ: from angle $\beta = 30^0$ ($S_S = 1{,}42$) ζ = 16,0; for $\beta = 45^0$ ($S_S = 2{,}47$) ζ = 12,0 and for $\beta = 60^0$ ($S_S = 4{,}27$) ζ = 18,5.

   Axial-blade burner type 2 had the lowest experimental values of the coefficient ζ of all tested burner models. The explanation of this lies in the fact that this burner model has profiled blades of primary and secondary air, which considerably reduce air flow resistance. For this burner model (Fig. 6) performed the determination of the coefficient ζ in function of continuous changes of blade angle and velocity, i.e. secondary air flow. Measurements were performed for different angles (positions) of secondary air blades $\beta = 15, 30, 45, 60$ and 75^0. These values of blade angle and actual secondary air velocities i.e. secondary air flow rates correspond to swirl numbers of secondary air of Ss = 0.853, 1.838, 3.184, 5.515 and 11.882.
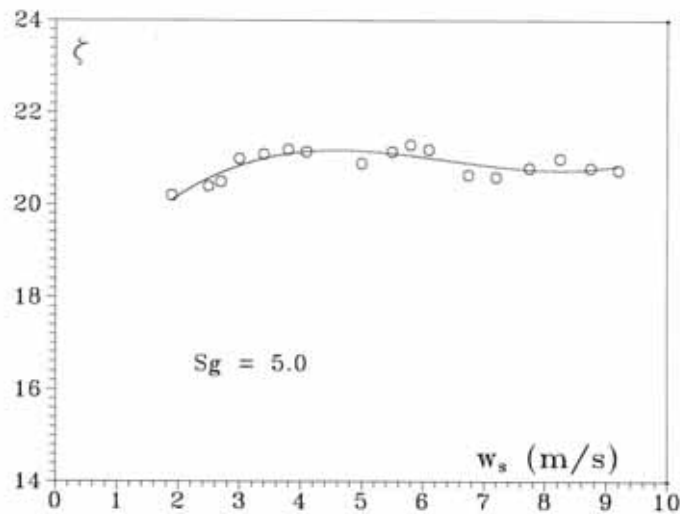

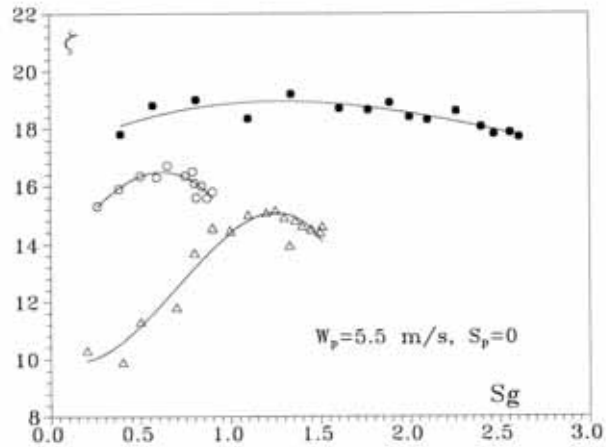Fig. 4. Hydraulic resistance coefficient for tangential burner model.

Fig. 5. Hydraulic resistance coefficient for axial-blade type 1 burner model.

Measurements showed that on the change of coefficient ζ the same effect has a changing the blade angle or shifting secondary air velocity i.e. flow through the burner. So, to their increase growth experimentally determined values of the coefficient ζ. The obtained values of the coefficient ζ which ranging from 2 to 4 fully correspond to the actual values of the coefficient of hydraulic resistance of the burner installed on real plants.

Experimentally determined values of the coefficient ζ correspond to a specific geometry of burner. The results can be replicated in real burners of the same or similar geometry. High values of the coefficient ζ obtained for some burner models are due to small burner input section of the burner, which is convenient for the model but not for real burner. In practice, it is common for the air injection through air ducts significantly greater at lower values of the coefficient ζ, as well as production of profiled blades.
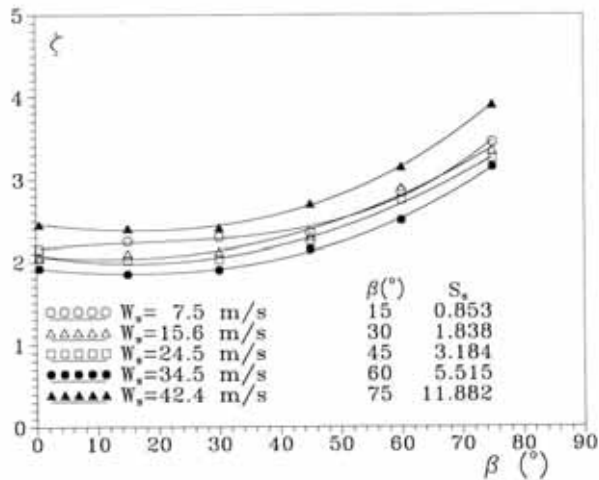

Fig. 6. Hydraulic resistance coefficient for axial-blade type 2 burner model.

5. Conclusions

Selection of the optimal swirl burner design depends on the loss of energy used to blow air through the burner. Cost of swirl device is evaluated using the so-called coefficient of hydraulic resistance of the burner, which reflects the energy losses due to swirling effect. This paper presents a method and experimental data to determine the burner hydraulic resistance of four laboratory model of swirl burners for pulverized fuel combustion (biomass, coal etc.). Investigation was performed for four swirl burner models where swirl of air is establish by movable blocks, with tangential air introduction and using a blades. Experimental measurements show that the hydraulic resistance of burner is very dependent on the geometric characteristics of the burner. The results can be successfully replicated in real burners of the same or similar geometry.

Acknowledgements

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