INSTALLATION FOR GAS MIXTURE HOMOGENIZATION IN PULVERIZED COAL COMBUSTION

CIOCANEA ADRIAN [1]*, DRAGOMIRESCU ANDREI[1], BUDEA SANDA[1], LEPADATU IOAN[2], KREVEI PETRICA[2], ALEXANDRU MARIN[1]

[1] *University "Politehnica" of Bucharest, Faculty of Power Engineering, Department of Hydraulics and Hydraulic Machinery, Splaiul Independentei 313, Bucureşti, România*

[2] *National Institute of Optoelectronics INOE 2000 IHP, Cutitul de Argint, Bucuresti, Romania*

Abstract: This paper presents the energy performances of a pneumatic installation designed to improve the homogenization degree of the mixture consisting in recirculated flue gas and primary air used in pulverized coal combustion. The installation consists in a series arrangement of a centrifugal fan and a side channels blower. The experimental setup is intended to equip the burner of a 2.5 MWh pilot boiler, where the influence of the mixture parameters on the reduction in NO_x emissions is studied. The performance curves of the centrifugal fan and those of the side channels blower were obtained experimentally. Based on them, the performance curves of the series arrangement were obtained analytically, in order to assess the variation of pressure depending on the flow rate required by the burner. Of a particular interest was the degree in which the usage of the side channels blower could eliminate the risk of surge typical for centrifugal fans operating at low discharges.

Keywords: coal combustion, fan, side channels blower, surge, gas mixture, NO_x emissions

1. INTRODUCTION

The prediction of NO_x emissions is difficult unless both coal blending characteristics and boiler geometry are simultaneously considered. Still, research on coal combustion is highly focused on reduction of NO_x emissions from power plants, since more than 30% of the total nitrogenous compounds in the atmosphere are due to this process. For practical reason, research was conducted for a better assessment on the efficiency and quality of the coal burning process [1-4]. Conclusions of the studies highlighted the difficulty to identify the species of NO_x emissions in conjunction with a turbulent burning mixture.

The main issue in turbulent combustion research is to assess how the magnitude of turbulence influences the flame structure (front thickening and/or wrinkling, quenching, etc.) and then how heat emission and hydrodynamical instabilities influence the turbulence parameters [5-7].

In order to obtain low NO_x emissions, a large number of technologies are available, which can be divided into two major classes: primary technologies that minimize or prevent NO_x formation in the combustion zone by controlling the combustion process [8-10] and secondary technologies that make use of chemicals to reduce NO_x

*Corresponding author, email: adrian.ciocanea@yahoo.com

formed in the combustion zone to molecular nitrogen [11]. The present paper approaches one of the primary technologies, continuing a previous research [12-15].

2. THEORETICAL ASSUMPTIONS

Theoretical models for the kinetics of coal combustion are widely described in the literature. A schematic of such a model is presented in Figure 1a. Coal-nitrogen conversion process in pulverized coal flames could be described by tens of species and hundreds of chemical reactions involved. For practical reasons simplifications like that presented in Figure 1b are required to achieve computational expediency [16].

The reactions are dependent on the coal combustion conditions – turbulent two-phase flow with significant heat release, char combustion, devolitalization, volatile burning, etc. Simplifications are required due to the incomplete understanding of the phenomena and because CFD models failed to be fully reliable. Hence, pilot-scale installations are used in order to validate the sensitivity of several fuel-dependent parameters [17, 18].

Different theoretical models assume the unburned and burned gas as incompressible. After the expansion has taken place – as the gas has burned while crossing the flame – both density and viscosity are modified. The density of the burned gas is generally much lower than that of the unburned gas and the viscosities on either side of the flame are constant but unequal, the viscosity of the burned gas being usually higher than that of the unburned gas.

Considering the complex phenomena described previously, the present paper proposes a new approach for improving one of the classical methods to reduce the NO_x emissions in the pulverized coal combustion. The method under discussion involves flue gas recirculation. Because of the reduced oxygen intake, the temperature decreases, thus inhibiting the formation of nitrogen compounds as presented in Figure 2 where for the same temperatures (isothermal lines for 1413 K and 1351 K) the fields of high temperature around the flame are more limited in the case of homogenous mixture between primary air and re-circulated flue gas in comparison with inhomogenous mixture situation. The main drawback of this method is the non-uniform concentration of the flue gas in the air-coal mixture. To alleviate this, a mechanical process using a specially designed side channels blower is added to the jet method in order to obtain a homogenous mixture of recirculated flue gas, air, and coal. The expected results were presented in [12] for a furnace of a 2.5 MWh pilot boiler, where the burning volume was considered to correspond to a thermal load of 360 kW, which is equivalent to a lignite flow rate of net heating

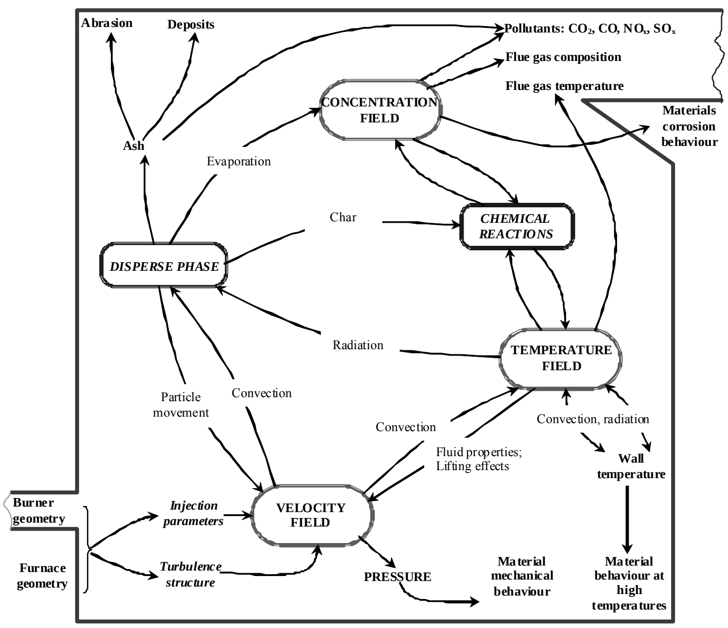


Fig. 1.a. General synoptic map of the complex phenomenon of the heterogeneous combustion process [12].

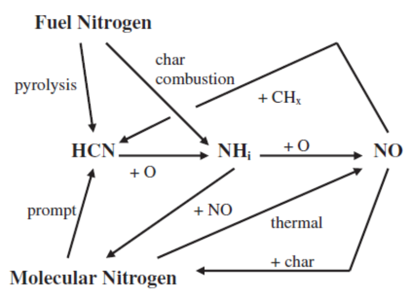
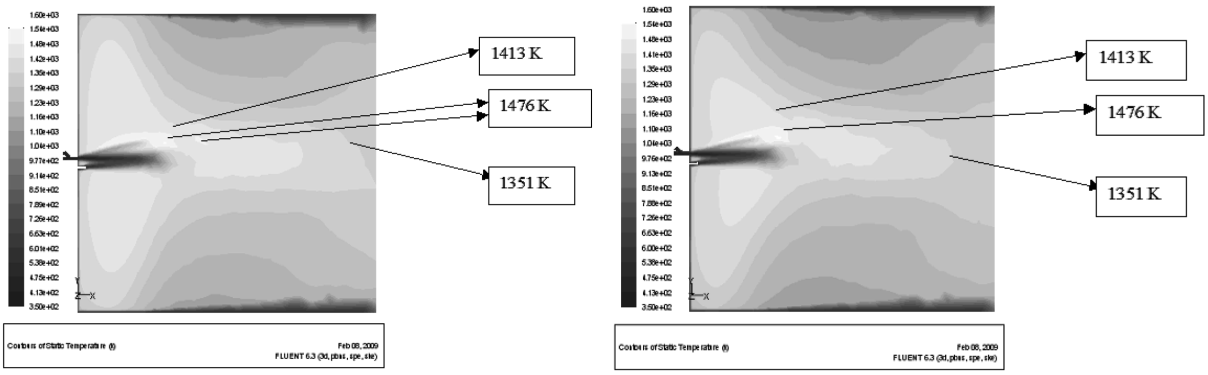


Fig. 1.b.Simplified NO formation and destruction pathways for coal flames [17].

value under 6.8 MJ/kg; the excess of air in front of the burner was considered to be 1.25 and the total air mass flow rate provided by the two burner nozzles was of 0.249 kg/s; the temperature of the recirculated flue gas was of 350 K and that of primary air was of 523 K; the percentage of recirculated gases was of 10…20%.

To validate the theoretical results, it is necessary to plot the performance curves of the pneumatic installation that feeds the burner in order to make possible the adjustment of its operating parameters according to the requirements of the burner.



a) b)

Fig. 2. Temperature distribution for: a) inhomogeneous mixture and b) homogenous mixture (vertical section).

3. EXPERIMENTAL SETUP AND RESULTS

The experimental setup consists in a series arrangement of a centrifugal fan with the rated discharge and pressure q_v = 1200 m^3/h and p_f = 7.5 kPa, respectively, and a side channels blower of a special design [13] having the rated discharge and pressure q_v = 400 m^3/h and p_f = 50 kPa, respectively (Figure 3). The main purpose of the blower is to mix as uniformly as possible the preheated primary air and the recirculated flume gas with which the burner is fed. In a classical installation, a good mixing is hindered by the differences in density and viscosity of the two gases. The proposed mixing method is expected to reduce also the effects of the hydrodynamic instabilities of the combustion.

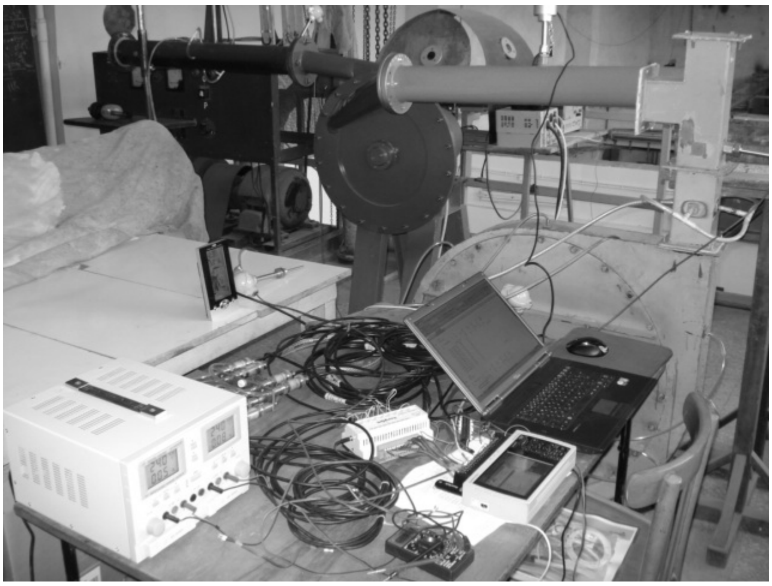


Fig. 3. Experimental setup – regenerative blower, centrifugal fan, transducers and data acquisition system.

The research involved the testing of the centrifugal fan and of the side channels blower. The performance curves of the two turbomachines were plotted according to SR EN ISO 5801:2009, SR EN ISO 13349:2009, and EUROVENT 1/4. The compressibility of the gas under the effect of both pressure and temperature was considered. At inlets, the pressures were measured with absolute pressure transducers having accuracies better than 0.1% FS. At fan outlet a similar transducer was used, while at blower outlet the pressure was measured with a relative pressure transducer having the accuracy better than 0.5% FS. The temperatures at inlets and outlets were measured with Pt100 temperature sensors.

The main parameters of the turbomachines were computed with the following relationships:

- stagnation pressure at inlet/outlet:

$$p_{sg1,2} = p_{1,2}\left(1+\frac{\kappa-1}{2}\mathrm{Ma}_{1,2}^2\right)^{\frac{\kappa}{\kappa-1}} \text{ [Pa]} \tag{1}$$

where the indexes 1 and 2 denote inlet and outlet, respectively.

- fan/blower pressure (pressure rise):

$$p_f = p_{sg2} - p_{sg1} \text{ [Pa]} \tag{2}$$

- specific compression work:

$$W = \frac{p_2 - p_1}{\rho_m} + \frac{v_{m2}^2 - v_{m1}^2}{2} \text{ [J/kg]} \tag{3}$$

The mass flow rate of the centrifugal fan was measured with an orifice plate installed in the suction pipe and was calculated according to SR EN ISO 5167-2:2004 with the formula:

$$q_m = \alpha\,\varepsilon\,\beta^2\,\frac{\pi D^2}{4}\sqrt{2\,\Delta p\,\rho} \text{ [kg/s]} \tag{4}$$

where Δp is the pressure drop caused by the orifice plate.

The mass flow rate of the side channels blower was measured with a hot-film anemometer whose sensor was placed in the centre of the suction pipe. The flow was assumed turbulent. Hence, the velocity profile inside the pipe has the form [18]:

$$v = v_{max}\left(1 - \frac{r}{R}\right)^{1/n} \tag{5}$$

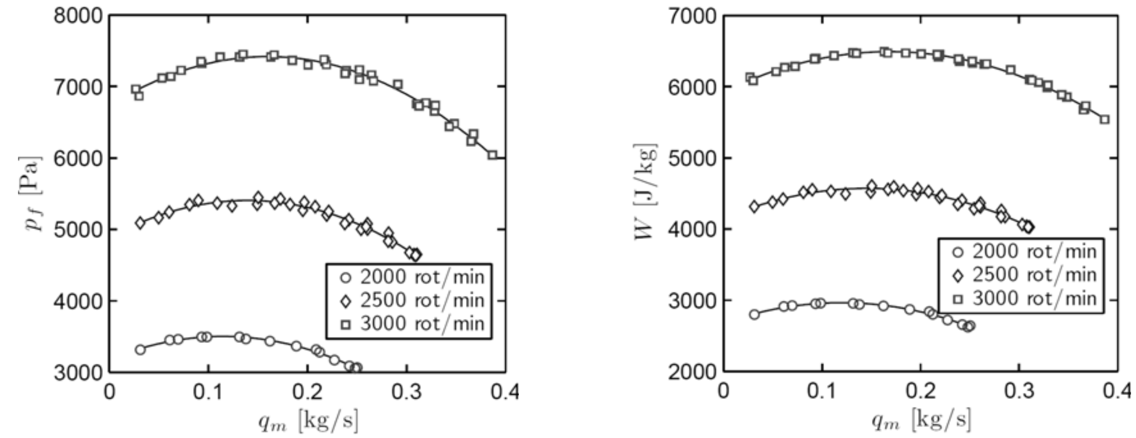
where the maximum velocity $v_{\max}$ is at the centre of the pipe.

The average velocity used to calculate the volume flow rate has the expression:

$$v_m = \frac{2n^2}{(1+n)(1+2n)}v_{\max} \tag{6}$$

Since the exponent n depends on the Reynolds number, the volume flow rate was computed iteratively. The mass flow rate was derived by multiplying the volume flow rate with the gas density in the suction pipe.

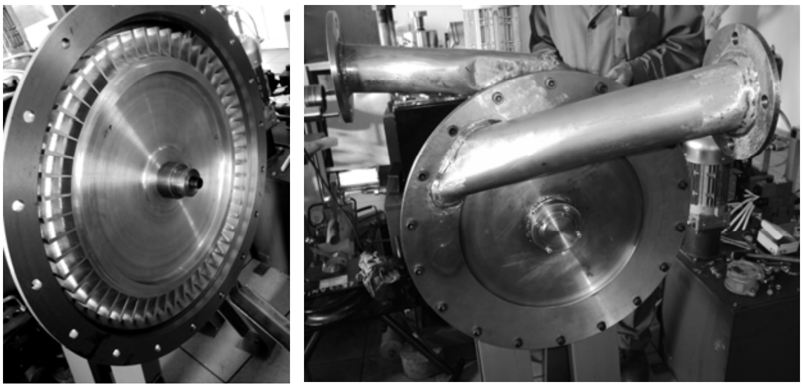
The measurements were made for three rotational speeds: 2000 rot/min, 2500 rot/min, and 3000 rot/min. To change the speeds, frequency converters were used. Figure 4 presents the performance curves of the centrifugal fan: fan pressure and specific compression work depending on mass flow rate. It can be seen that the curves are flat, showing only small variations over the entire flow rate range. This is an important advantage when feeding the burner, since the pressure can remain almost constant regardless of the flow rate required by the operating regime. However, the curves show an ascending slope at small flow rates, which indicates that the fan is prone to surge at such operating regimes.



a) b)

Fig. 4. Performance curves of the centrifugal fan: a) fan pressure, b) specific compression work.

The side channels blower is depicted in Figure 5. Its casing was designed so that the inlet and the outlet divert the flow of the gas mixture in as little as possible. It was assumed that a discharge path directed along the propagation direction of the main flow vortex helps to maintain the internal energy of the gas mixture, so that a separation of the gases between blower and burner does not occur.



a) b)

Fig. 5. Construction of the side channels blower: a) runner; b) inlet (front) and outlet (back).

Inside the blower, the preheated primary air and the flume gas are continuously recirculated between runner blades and casing, the mixture being thus homogenized. The part of installation used for testing the blower is shown in Figure 6.

Figure 7 presents the performance curves of the side channels blower. They have a descending slope over the entire flow rate range, which means that the danger of surge is not present. Moreover, the steepness of the slope, which is typical for side channels blowers, should be noticed and considered especially when the feeding of the burner should be finely adjusted. At large flow rates, the pressure delivered by the blower tends to zero, which means that the blower alone is not well suited for feeding the burner.

The performance curves of both the blower and the fan were fitted with second order polynomials of the forms:

$$p_f = \pi_1 q_m^2 + \pi_2 q_m + \pi_3 \quad \text{and} \quad W = \lambda_1 q_m^2 + \lambda_2 q_m + \lambda_3 \tag{7}$$

The performance curves for the series arrangement of the fan and the blower were obtained analytically, by adding the corresponding polynomials of each machine. Figure 8 presents the diagrams obtained.

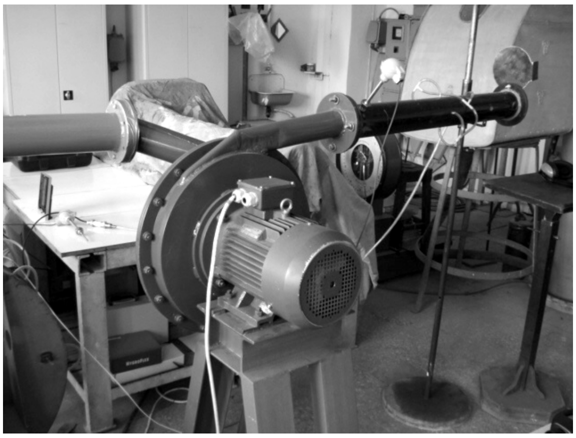
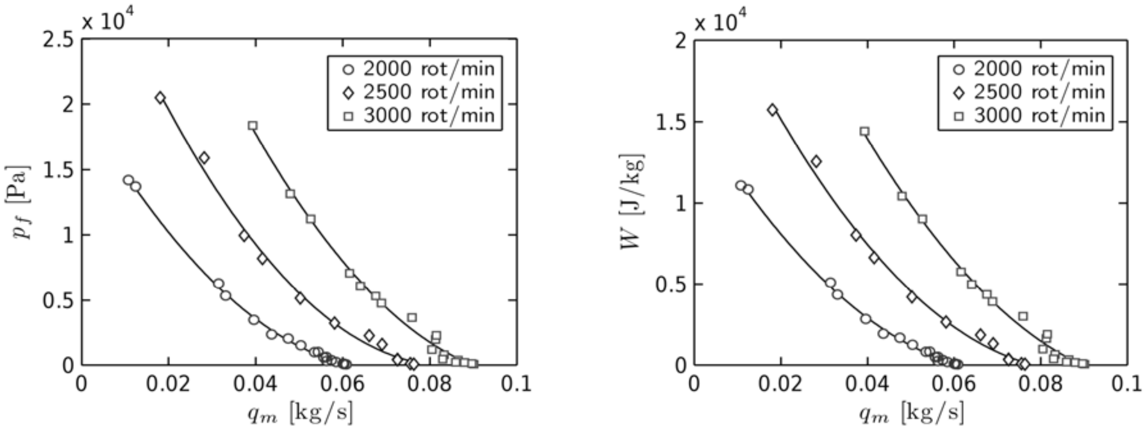
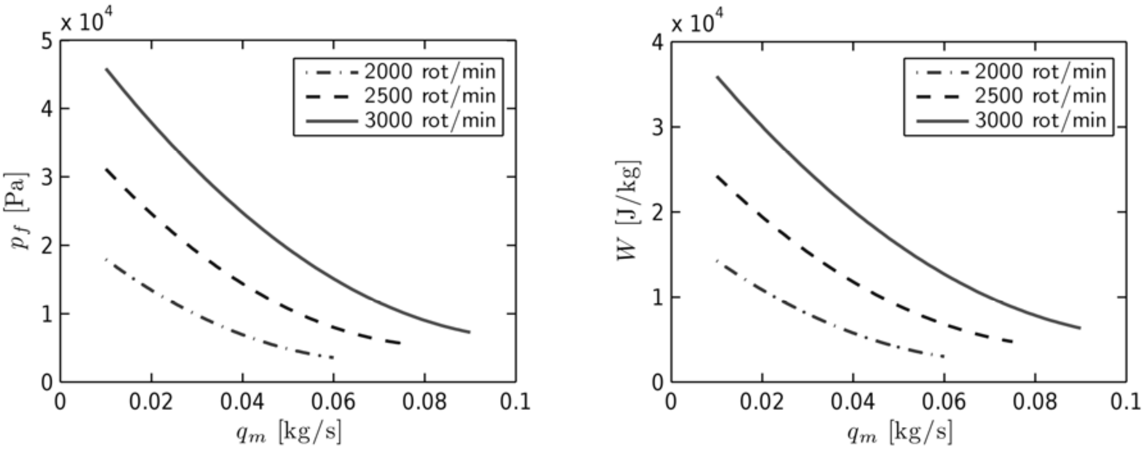


Fig. 6. Part of the experimental setup used for testing the side channels blower.



a) b)

Fig. 7. Performance curves of the side channels blower: a) blower pressure; b) specific compression work.



a) b)

Fig. 8. Performance curves of the series arrangement formed by the centrifugal fan and the side channels blower.

It should be noticed that by coupling in series the two turbomachines the surge region disappears. The results suggest that when the fan and the blower operate at different rotational speeds the combination between fan and blower allows the obtaining of a larger range of operating conditions that can be better adjusted to the feeding requirements of the burner.

In order to assess the homogeneity of the mixture with which the burner is fed, flow visualizations were made at blower discharge. For this purpose, the discharge pipe was replaced with a transparent one, made of plexiglass, and supplemental piping was used in order to build a closed circuit inside which air is recirculated by the blower. Sawdust was introduced into this circuit. For better visualizing the flow of air and sawdust, a sheet of red light was created with a 5 mW laser perpendicular to the discharge pipe. Pictures of flow patterns observed under working conditions corresponding to the best operating points at rotation speeds of 2500 rpm and 3000 rpm are presented in Figure 9. As it can be seen, a vortex forms in the upper part of the discharge section. Its size undergoes continuous changes during the flow and seems to be strongly influenced by the rotation speed. The vortex is larger at the lower rotation speed and decreases as the rotation speed increases. In the rest of the discharge section, a high homogeneity degree of the mixture of air and sawdust can be observed.

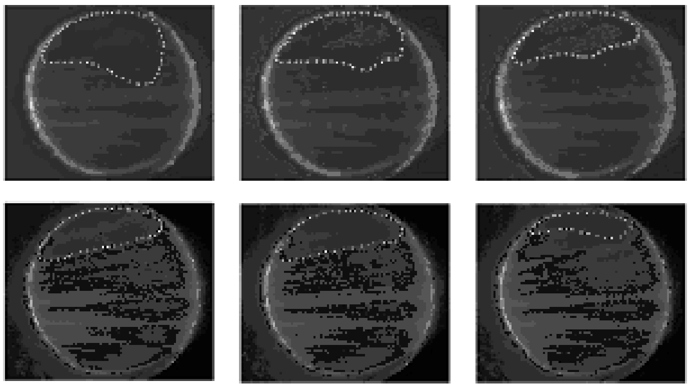


Fig. 9. Flow patterns at the discharge of the regenerative blower obtained at 2500 rpm (top) and 3000 rpm (bottom).

4. CONCLUSIONS

A solution to the requirements for reducing emissions of nitrogen compounds resulting from burning coal is to feed the burners with a mixture of preheated primary air and recirculated flume gas. This leads to a decrease in burning temperature, which inhibits the formation of nitrogen compounds. The key element of this solution is the proper mixing of the gases, which is rendered difficult by the differences in density and viscosity. To fulfil the main desiderata, namely the feeding of the burner at the required pressure and flow rate and the homogenization of the fed gases, an installation consisting of a centrifugal fan and a side channels blower coupled in series was proposed. The main role of the blower is to assure a good mixing of the gases due to the continuous recirculation between runner and casing. Experimental research was conducted in order to obtain the performance curves of the machines and of their series arrangement.

The results obtained show that the performance curves of the fan used are flat, the fan being well suited from this point of view for feeding the burner at different operating regimes. However, the fan is prone to surge which, as it is known, could become a serious problem during operation. For the blower, the surge is not an issue, but the machine working alone cannot deliver the required pressure at high flow rates.

The coupling in series of the two pneumatic turbomachines, beside the fact that provides a proper mixing of the gases, seems to solve also the aforementioned problems. At high discharges, the pressure required for the proper operations of the burner is provided by the centrifugal fan. At lower discharges, the risk of surge, present when using only centrifugal fans, is eliminated by the side channels blower. It can be thus concluded that the installation studied could cover successfully the entire domain of flow rates and pressures requested by the process to which it is destined, namely the pulverized coal combustion inside the furnace of a 2.5 MWh boiler.

REFERENCES

[1] Schweitzer, Y., Cârţînă, Gh., Grigoraş, Gh., Aspects of NO_x in coal combustion, The 6th International Power Systems Conference, Timişoara, Romania, 2005, p. 507-514.
[2] Song, Y.H., Pohl, J.H., Beer, J.M., Sarofim, A.F., Nitric oxide formation during pulverized coal combustion, Combustion Science and Technology, vol. 28, 1982, p. 31-39.
[3] Su, S., Pohl, J.H., Holcombe, D., Hart, J.A., Techniques to determine ignition, flame stability and burnout of blended coals in p.f. power station boilers, Progress in Energy and Combustion Science, vol. 27, 2001, p. 75-98.
[4] Van der Lans, R.P., Glarborg, P., Dam-Johansen, K., Influence of process parameters on nitrogen oxide formation in pulverized coal burners, Progress in Energy and Combustion Science, vol. 23, 1997, p. 349-377.
[5] Qian, J., Tryggvason, G., Law, C. K., A Front Tracking Method for the Motion of Premixed Flames, Journal of Computational Physics, vol. 144, 1998, p. 52-69.
[6] Queiros-Conde, D., Feidt, M., Entropic Skins Geometry Applied to Dynamics of Turbulent Reactive Fronts, International Journal of Thermodynamics, vol. 11, no. 1, 2008, p. 11-19.
[7] Poinsot, T., Veynante, D., Theoretical and numerical combustion, Eduards, 2005.
[8] Xuea, S., Huia, S., Liub, T., Zhoua, Q., Xua, T., Hua, T., Experimental investigation on NOx emission and carbon burnout from a radially biased pulverized coal whirl burner, Fuel Processing Technology, vol. 90, no. 9, 2009, p. 1142-1147.
[9] Cvoro, V., Easson, W.J., Optimisation of furnace design for in-furnace NOx reduction: the effect of pitch spacing on multiple burner jet interactions, Fuel, vol. 82, no. 15-17, 2003, p. 2075-2085.
[10] Jinga, J., Li, Z., Zhua, Q., Chena, Z., Rena, F., Influence of primary air ratio on flow and combustion characteristics and NOx emissions of a new swirl coal burner, Energy, vol. 36, no. 2, 2011, p. 1206-1213.
[11] Kobayashi, H., Bool, I., Lawrence E., U.S. Patent 10,436,164, 2003.
[12] Ciocănea, A., Prisecaru, T., Mixed jet-vortex method for reducing NO_x emissions of pulverized coal combustion, Environmental Engineering and Management Journal, vol. 9, no. 10, 2010, p. 1421-1425.
[13] Ciocănea, A., Prisecaru, T., Budea, S., Special designed regenerative blower for biphase fluid flows, International Conference for Energy and Environment CIEM, Bucharest, Romania, 2009, p. 65-72.
[14] Postelnicescu, M., Theoretical research and experiments for designing a dust burner for solid fuel using jets (pilot flame), Scientific Bulletin of I.P.B., Series in Mechanics, vol. 43, 1981, p. 39-46.
[15] Prisecaru, T., Ene, A.S., Semenescu, A., Prisecaru, M., Mihăescu, L., Considerations regarding the application of the turbulence models in the case of burning systems with strong swirling, Metalurgia International, vol. 5, 2000, p. 10-16.
[16] Wendt, J.O.L., Fundamental coal combustion mechanisms and pollutant formation in furnaces, Progress in Energy and Combustion Science, vol. 6, no. 2, 1980, p. 201-222.
[17] Diez, L.I., Cortes, C., Pallares, J., Numerical investigation of NOx emissions from a tangentially-fired utility boiler under conventional and overfire air operation, Fuel, vol. 87, no. 7, 2008, p. 1259-1269.
[18] Chui, E., Gao, H., Estimation of NOx emissions from coal-fired utility boilers, Fuel, vol. 89, 2010, p. 2977-2984.
[19] Zucrow, M.J., Hoffman, J.D., Gas Dynamics, John Wiley & Sons, New York, 1976.