A study of burning processes of fossil fuels in straitened conditions of furnaces in low capacity boilers by an example of natural gas

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Abstract. The aim of this work is to research operations of modern combined low-emission swirl burner with a capacity of 2.2 MW for fire-tube boiler type KV-GM-2.0, to ensure the effective burning of natural gas, crude oil and diesel fuel. For this purpose, a computer model of the burner and furnace chamber has been developed. The paper presents the results of numerical investigations of the burner operation, using the example of natural gas in a working load range from 40 to 100%. The basic features of processes of fuel burning in the cramped conditions of the flame tube have been identified to fundamentally differ from similar processes in the furnaces of steam boilers. The influence of the design of burners and their operating modes on incomplete combustion of fuel and the formation of nitrogen oxides has been determined.

1. Introduction
Combustion of organic fuel in swirling jets is widely used in a variety of power plants. Fuel burning always occurs in a confined space, but the degree of limitation in different types of units is significantly different. There is a large number of studies, devoted to the burning in combustion chambers of large energy boilers (in condition of small tightness) [1, 2] and to the processes taking place in combustion chambers of gas turbines (very high tightness) [3, 4]. The intermediate position is occupied by a fire-tube and some water-tube boilers with a capacity of 2-5 MW. As practice has shown, the regularities of processes in such units have their own distinctive features in comparison with the two cases cited above. In [5, 6], it is noted that toroidal vortex regions are created in cylindrical furnaces. However, the regularities of their formation and the influence on heat and mass transfer processes have not been sufficiently studied yet. Therefore, the purpose of this work was to study the processes of ignition and burnout of various types of fuel, heat and mass exchange and emission of nitrogen oxides in the constrained conditions of small-sized furnaces of boilers.

2. Main approaches
Preliminary studies have shown that the solution of this problem in isolation from the specific equipment is not correct, therefore the main studies were carried out for a 2 MW fire-tube boiler (Figure 1, a) equipped with a vortex burner. This combination of equipment is widespread in the domestic energy sector.
The burner device (Figure 1, b) used in the work is equipped with one annular channel for supplying secondary air, which is equipped with an axial blending device for its swirling. The outlet part of the
burner contains a diffuser-confuser section and a perforated disk that can change its position to influence the characteristics of the flow. In the central channel, which is cooled by a small fraction of air, a fuel injector is installed to spray the liquid fuel into the furnace. Natural gas (Table 1) is fed into the annular air channel after the vortex blades.

![Figure 1. Fire-tube of boiler (a) and vortex burner device (b)](image)

ANSYS software was used to simulate the jet flow with fuel combustion. The problem of gas fuel burning was solved using the program ANSYS CFX, and for liquid fuel the ANSYS FLUENT was used. The obtained and processed results were presented in the form of fields of velocity, temperature and concentration variables, mean integral values in different cross and longitudinal sections, etc. In this form they were used for further analysis.

The geometric model was prepared in the SolidWorks program. The ANSYS MESHING software complex with a combination of Delaunay, Sweep and Advancing Front methods were used to build an unstructured calculation grid.

The solution of the known differential equations (continuity, motion, diffusion, energy, etc.) is based on gas dynamics modeling in ANSYS packages and is supplemented by a number of special models. To solve the motion problem in the ANSYS FLUENT program, the RANS method supplemented with the Realizable k-ε model was used [7, 8]. Heat transfer by radiation was calculated using the Discrete Ordinates model [9], and liquid fuel spraying process was simulated using the Pressure Swirl Atomizer model (based on the LISA model) [10] and a number of submodels (Stochastic Collision, Coalescence, TAB Breakup).

The motion of the droplets of liquid fuel, formed as a result of spraying, was described by the Discrete Phase Model, which was supplemented by the Particle Radiation Interaction model. The formation of nitrogen oxides (thermal, fuel, prompt) was described by well-known mechanisms [11]. To describe chemical reactions a diffusion-kinetic model was used [12, 13].

The combustion process was investigated for gas, oil and diesel fuel. Preliminary studies had shown differences in the flame structure for different loads of the boiler, i.e. fuel-air mixture, so the studies were carried out for a range of loads from 40% to 100%. The excess air in the furnace was maintained at 1.17. The inlet air temperature was 30°C.

The realization of a full-scale experiment for verification of the mathematical model was not possible; therefore, to confirm the reliability of the obtained results, the traditional approach - the study of grid convergence - was used. Previously, the problem was solved with a fixed number of elements, which then increased approximately twice. A conclusion about their reliability was made based on the comparison of the obtained results, and further the grid refinement was carried out, if necessary. This approach allowed achieving acceptable results with the reasonable use of mathematical models,
included in the modeling process. After analyzing the obtained data and generalizing the values, a conclusion was made about the adequacy of the grid detail with its dimension of about 17 million elements, since most of the studied parameters agreed with the data for a 35 million element grid with an error of less than 2.5%, which is an acceptable indicator.

3. Results
The obtained results for the natural gas combustion indicate that the ignition of the gas-air mixture at maximum loads (80-100% of nominal) occurs due to the creation of two recirculation high-temperature zones in the flame tube (Figure 2). The central return flow zone, which is located not far from the burner section on the axis of the flame tube, brings high-temperature combustion products to the root of the flame and provides a steady ignition in this area. The second torus-shaped recirculation zone is located on the periphery, near the diffuser part of the flame tube (Figure 2). High-temperature combustion products from this zone ensure the fuel-air mixture ignition at the external boundary of the swirling flow in the immediate nearness of the burner section. The maximum temperature region, shaped like a torus, is located approximately one caliber from the burner (that is, just behind the main burning-out zone) and borders directly with the wall of the flame tube, which indicates a decrease in the reliability of its operation. Thus, at maximum loads a stable ignition of the mixture at the outer and inner boundaries of the fuel-air flow is achieved. This is confirmed by the fields of concentrations of oxygen and carbon monoxide, showing that almost complete burn-out of the fuel is provided in the first half of the flame tube at maximum loads.

The efficiency of the processes of ignition and the air-fuel mixture burnup decreases with decreasing load, that is, with a decrease in the swirling flow momentum. If at maximum loads the return flow zones and the ignition borders are practically symmetrical, then when the thermal power decreases, the flame is "stalled" to the wall of the flame tube (Figure 2). The zone of maximum temperatures loses the torus shape and begins to rotate around the circumference near the wall of the flame tube. As a result, the symmetry of all zones and boundaries is violated, and the processes of ignition of the fuel-air mixture are delayed. At the same time, due to the formation of a larger number local active combustion zones with significantly smaller volumes, the maximum temperatures in them noticeably increase to the level of 1950-1990°C. This process, in particular, is significantly affected by the uneven supply of gas fuel through the annular channel of the peripheral air flow.

The calculations have not shown the presence of appreciable concentrations of CO (Table 2) at the exit from the calculated combustion region, even at reduced loads, but the concentration of nitrogen oxides (Figure 3) first decreases, and then increases up to greater values than obtained for the nominal operating mode. This is due to the fact that when the load decreases, the consumption of combustion products decreases. This, on the other hand, reduces the mean integral temperature level at the exit from the calculation area (at the exit from the turnover chamber) and the speed of gas motion inside the flame tube. On the other hand, it increases the time of their stay within the calculated volume. These conflicting trends have different effects on the level of nitrogen oxides emission in the volume of the flame tube.

When the load decreases from 100 to 80% of the nominal value, the maximum temperature level practically does not change (Table 2), while the volume of the zone with a temperature above 1500 °C decreases. This results in an appropriate decrease in the emission of nitrogen oxides. With a further load decrease the symmetry of the aerodynamic organization of combustion was destroyed, which led to the "stalling" of the flame to the wall of the flame tube. Despite occupying smaller volumes in comparison with the maximum loads, high-temperature zones on these regimes were characterized by a rather high level of maximum temperatures T = 1950-1990°C. At the same time, reduced gas velocities led to an increase in the residence time of the combustion products in these high-temperature zones. As a result, both of these factors positively influenced the NO emission, the yield of which in reduced loads decreased by almost 70%, compared to the nominal mode operation, and reached 113-115 ppm.
4. Discussion

Due to the limited volume of the article, the results of modeling of the combustion processes of oil and diesel fuel are not presented, however, the flow pattern is similar to that for the case of gas combustion, despite the existing differences in the ignition mechanisms [14]. The process of ignition is tightened due to the need for heating and evaporation of the droplets. Combustion of liquid fuel and bottom water is described in detail in [15].

The flow structure in the furnace is regulated by adjusting the burner, or rather the position of the central disk (Figure 1, a). In many respects, the negative effects observed with a reduced load (Figures 2, 3) can be avoided, providing adjustment of the gap at the root of the ring flow. However, in practice

![Diagrams of axial velocities and gas temperatures fields in fire tube depending on boiler load N.](image-url)

Figure 2. Axial velocities (a-d) and gas temperatures fields (e-g) in fire tube depending on boiler load N: a, e) $N = 100\%$; b, f) $N = 80\%$; c, h) $N = 60\%$; d, g) $N = 40\%$. 
this is not used, due to the complexity of implementing such a solution. In this way, the burner is set up once at rated power. That is why it is very important to realize all the subtleties of the furnace operation in the entire range of loads. If the position of the ring at a low load is adjusted, the aerodynamic resistance of the burner at its nominal operating mode will increase substantially, which is unacceptable.

The work of burners of other designs should be studied to obtain more complete data, but most burners for boilers of this type have similar design solutions: one annular channel, a central disk, adjustment of the width of the annular gap between the disc and the outer wall of the channel, preliminary mixing of the gas and spraying of liquid fuel through central nozzle. Therefore, the swirling flow created by such burners should have a very similar configuration.

![Figure 3](image)

Figure 3. NO concentration fields in fire tube depending on boiler load $N$:
- a) $N = 100\%$
- b) $N = 80\%$
- c) $N = 60\%$
- d) $N = 40\%$

| Boiler load,  | Maximum gas | Gas temperature, | NO concentration | CO concentration |
|--------------|-------------|------------------|------------------|------------------|
| %            | temperature, $^\circ$C | $^\circ$C | ppm | ppm |
| 100          | 1709        | 950              | 67.3             | 0.39             |
| 80           | 1718        | 882              | 47.5             | 0.43             |
| 60           | 1954        | 787              | 114.7            | 0.54             |
| 40           | 1991        | 673              | 113.4            | 0.74             |

5. Conclusion
1. The considered flow structure created by a typical swirl burner in the furnace of a flame tube boiler provides almost complete fuel burn-up throughout the all range of workloads from 40 to 100% of the nominal.
2. Combustion of natural gas at reduced boiler loads is accompanied by an increased yield of nitrogen oxides, which exceeds the normative values (GOST 30735-2001), that is associated with a loss of stability of the swirling flow.
3. Avoiding negative effects at low loads may be ensured by constant adjustment of the annular gap, depending on the current load (fuel consumption), while observing the necessary level of excess air (i.e., its flow rate).
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