Theoretical description of preheating a gas-air mixture in a gas burner with a heat divider

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Abstract. The article is devoted to the study of thermal processes occurring during the operation of a household gas burner. The preparation of gas fuel for combustion is one of the important functions that a gas burner performs. At the same time, when designing gas burner devices, it is important to determine the final temperature of the gas-air mixture inside the burner body, since an increase in the temperature of the resulting mixture makes it possible to intensify the combustion process. We have proposed a solution that allows to intensify the transfer of heat in the body of a gas burner from the lid heated due to contact with the flame to the gas flow using the thermal shape of the conical divider installed in the center on the inner side of the lid. It has been established that the location of the heat divider in the center on the inner side of the cover allows minimizing or completely eliminating the formation of a stagnant zone. The conical shape of the heat divider provides less resistance to the oncoming flow movement, and also because of its own side surface, it allows you to increase the useful area of the heat exchange surface. An expression is obtained for determining the final temperature of the gas-air mixture at the outlet from the firing holes of the burner body.

1. Introduction

Improving the life and well-being of the population requires the development of infrastructure, the development of regional energy and industry, as a result, leads to an increase in energy consumption, including natural gas.

In 2019, production of natural and associated petroleum gas in Russia amounted to 738.4 bcm, which is 1.7% more than in 2018 and is a record figure for the entire period of Russian gas production [1].

In the structure of natural gas use in Russia at the end of 2019, with a share of 60.2% and a consumption volume of 289 bcm, the communal and industrial sector is in the lead, the level of gasification reached 70.1% [2].

One of the main devices of industrial and domestic gas-using devices is a gas burner device. It is designed to convert the energy of the combustible gas process into thermal energy, while the gas must be completely burned [3-7].

In this regard, gas supply specialists are faced with new complex tasks related to the development of new systems and development and reconstruction of use, increasing their reliability, more economical use of gas and minimal environmental impact.

Investigations of combustion processes and the development of effective gas-burner devices are the subject of many works by domestic and foreign scientists. An example is the work [8], in which he investigates the efficiency of heat recirculation in the operation of a gas burner using the porous medi-
um technology. In [9] the author was aimed to study the influence of four significant parameters, namely degrees of outer and inner ports and the number of outer and inner ports of high-pressure gas burners, to the thermal efficiency by the experiments. In [10] the author of this experimental investigation was to see the effects on using different design burner heads on the performance of LPG cooking stove. Also, burners of different material were used to study the effects of burner material on LPG stove performance. [11] Author this paper discusses the performance investigations of a conventional burner with different porous media used in LPG cooking stove.

The burner device for burning gaseous fuel has the following main functions: gives the necessary components and the flow rate of gas and air; ensures mixing of gas fuel with air, sometimes heats the resulting mixture to the ignition temperature - thereby preparing the resulting mixture for subsequent combustion; provides stabilization of combustion, creates conditions for the combustion of the mixture without separation and flame breakthrough; provides a minimum content of toxic gases in the composition of combustion products [12].

Consequently, the development of methods for increasing the efficiency of gas-burner devices, obtaining expressions for calculating the average temperature of the gas-air mixture inside the gas burner with a mathematical description of the thermal processes taking place is urgent.

2. Materials and Methods
To theoretically describe the process of heating a gas-air mixture in a burner body equipped with a heat divider, we used the equations of jet aerodynamics, as well as heat transfer equations based on the laws of thermal processes of Newton-Richman and Stefan-Boltzmann.

3. Results and Discussion
Let us consider the processes of movement of the flow of the gas-air mixture inside the burner, which have a significant effect on the distribution of heat.

In the general case, the flow pattern of the gas flow inside the burner body is similar to the pattern of motion of the shock gas jet [13-15] (Figure 1.).

![Figure 1. Shock gas jet scheme.](image)

The gas stream flowing out of the gas burner nozzle strikes at right angles against the flat inner surface of the cover, which is distant at a distance H, as a result of which it forms a stagnant zone - a certain area of vortices that prevents further free movement of the jet (Figure 2.). This area is characterized by a lower intensity of heat transfer, and when calculating heat transfer from the plate to the inci-
dent transverse gas flow, the plate surface occupied by the stagnant zone is excluded - thereby reducing the useful heat transfer surface and, accordingly, the heat transfer efficiency. For example, in [16], in calculating heat transfer, the stagnant zone area was taken to be equal to the area limited by the diameter of the outlet part of the Venturi tube, and thus the useful area corresponded to the annular part, where the outer diameter is equal to the diameter of the burner cover, and the inner diameter is the diameter of the Venturi tube.

We have proposed a solution that makes it possible to intensify the transfer of heat in the body of a gas burner from the lid heated due to contact with the flame to the gas flow using a conical heat divider installed in the center on the inside of the lid (Figure 3.). Placing a heat divider in the center on the inner side of the lid minimizes or completely eliminates the formation of a stagnation zone. The conical shape of the splitter provides less resistance to the oncoming flow of the flow, and also due to its own lateral surface allows to increase the useful area of the heat exchange surface.
The design of the gas stove burner with increased efficiency has been developed. The main structural elements of the proposed gas burner are: body, gas nozzle, mixer, firing holes for the outlet of the gas-air mixture, a cover with a preheating device and a primary air regulator [17-18] (Figure 4.).

To describe the operation of a gas-burning device, we draw up the heat balance equations, the distribution scheme of the main heat losses of a gas-burning device is represented by Figure 5.:  

\[ Q_2 = Q_1 + Q_3 + Q_4 + Q_5 \]  

(1)

where \( Q_2 \) is the heat released during the combustion of gaseous fuel; \( Q_1 \) is heat losses caused by radiative heat exchange between the volume of burnt gas and the environment; \( Q_3 \) the total amount of heat absorbed by the test vessel; \( Q_4 \) the heat loss with flue gases, after heat exchange with the side surface of the test vessel; \( Q_5 \) the total amount of heat absorbed by the body of the gas burner.

Figure 4. Original gas burner with preheating of the gas-air mixture: 1 - housing, 2 - gas nozzle, 3 - axial outlet, 4 - mixer, 5 - outlet end of the mixer, 6 - cover, 7 - fire holes, 8 - heat divider, 9 - primary air regulator.

Figure 5. Scheme of distribution of heat losses of the heat balance of a gas burner device.
The total amount of heat generated by the combustion of gaseous fuel is calculated:

\[ Q_2 = m_{gas} \cdot Q_f \]  

(2)

where \( m_{gas} \) is the mass fuel consumption, kg/s; \( Q_f \) is the lowest heat of combustion of the fuel, kJ/kg.

Heat losses caused by radiative heat exchange between the volume of burnt gas and the environment are expressed according to the Stefan-Boltzmann law by the formula [19-22]:

\[ Q_i = \varepsilon \cdot \sigma \cdot (t_{flame}^4 - t_{amb}^4) \cdot F_{flame} \]  

(3)

where \( \varepsilon \) is the emissivity (emissivity); \( \sigma \) is the Stefan-Boltzmann constant \( (\sigma = 5.67 \cdot 10^{-8} \text{ W/(m}^2\text{-K}^4)) \), \( t_{flame} \) and \( t_{amb} \) is the flame and ambient temperature, °C, \( F_{flame} \) is the flame surface area, m².

We represent the processes of heat transfer between the surfaces of solids and a gaseous medium by expressions based on the Newton-Richmann law [23-24].

The total heat absorbed by the test vessel is determined:

\[ Q_2 = (c_{p(water)} \cdot m_{water} + c_{p(vesSEL)} \cdot m_{vesSEL}) \cdot (t_{in(water)} - t_{fin(water)}) \]  

(4)

where \( c_{p(water)} \), \( c_{p(vesSEL)} \) mass heat capacity at constant pressure of water and the material of the test vessel (aluminum), kJ/(kg°C), \( m_{water} \), \( m_{vesSEL} \) is the mass of water and the test vessel itself, kg; \( t_{in(water)} \), \( t_{fin(water)} \) is the initial and final water temperature in the test vessel, °C.

Heat loss with flue gases after heat exchange with the side surface of the test vessel:

\[ Q_3 = m_{gas} \cdot c_{p(gas)} \cdot (t_{in(gas)} - t_{amb}) \]  

(5)

where \( m_{gas} \) mass flow rate of flue gases, kg/s; \( c_{p(gas)} \) is the mass heat capacity at constant pressure of flue gases, kJ/(kg°C); \( t_{in(gas)} \), \( t_{amb} \) is the temperature of flue gases after heat exchange with the side surface and ambient temperature, °C.

Heat absorbed by the body of the gas burner device:

\[ Q_4 = c_{p(burner)} \cdot m_{burner} \cdot (t_{fin(burner)} - t_{in(burner)}) \]  

(6)

where \( c_{p(burner)} \) is the mass heat capacity at constant pressure of the material of the gas burner device, kJ/(kg°C); \( m_{burner} \) is the mass of the gas burner device, kg; \( t_{in(burner)} \), \( t_{fin(burner)} \) is the initial and final temperature of the gas burner device, °C.

\[ \eta_{(th)} = \frac{Q_4}{Q_2} \]  

(7)

Thus, the thermal efficiency of a gas burner device is the ratio of the useful heat absorbed by the test vessel to the total amount of heat released during the combustion of gaseous fuel.

Preparation of gas fuel for combustion is one of the important functions that a gas burner performs. At the same time, it is useful for developers of gas burner devices to determine the final temperature of the gas-air mixture inside the burner body, since an increase in the temperature of the resulting mixture makes it possible to intensify the combustion process. Since combustion is a chemical reaction of oxidation of gaseous fuel with oxygen contained in the air, this feature of the process is explained, for example, by the Van't Hoff rule, which states that when heated, the rate of most chemical reactions increases [25-27].

The process of operation of the gas burner device is accompanied by the absorption by the body of the gas burner device of a certain part of the heat released during the combustion of gas. This type of heat loss is unavoidable for atmospheric burners. At the same time, the heated surfaces of the gas burner transfer heat to the gas-air mixture flowing in the housing [28], thereby ensuring its preheating.
Let us consider the processes of heat transfer during steady state operation, occurring in the housing of a gas burner.

In steady-state operation, when the temperature of the burner cover has reached a constant value of \( t_{\text{cover}} \) and does not change over time, the amount of heat absorbed by the flow of the gas-air mixture when the flow temperature changes from \( t_{\text{in(flow)}} \) to the temperature \( t_{\text{fin(flow)}} \) will be expressed:

\[
Q_{\text{flow}} = G_{\text{flow}} \cdot c_{p(\text{flow})} \cdot (t_{\text{fin(flow)}} - t_{\text{in(flow)}}) \tag{8}
\]

where \( G_{\text{flow}} \) the mass flow rate of the gas-air mixture, kg/s, \( c_{p(\text{flow})} \) is the mass capacity at constant pressure of the gas-air mixture, kJ/(kg·°C); \( t_{\text{in(flow)}} \), \( t_{\text{fin(flow)}} \) respectively, the initial temperature and the temperature of the gas-air mixture at the outlet from the firing holes, °C.

In this case, as a result of convective heat transfer from the burner cover with a temperature \( t_{\text{cover}} \) to the gas flow with an initial temperature \( t_{\text{in(flow)}} \) the amount of heat is transferred, determined by the formula:

\[
Q_{\text{cover}} = F_{\text{h.t.a.}} \cdot \alpha_{\text{cover}} \cdot \Delta t_{\text{flow}} \tag{9}
\]

where \( F_{\text{h.t.a.}} \) is the area of the inner surface of the cover participating in the heat exchange with the flow of the gas-air mixture, \( m^2 \), \( \alpha_{\text{cover}} \) is the heat transfer coefficient of the inner surface of the cover participating in the heat exchange with the flow of the gas-air mixture, W/(m²·K), \( \Delta t_{\text{flow}} \) is the logarithmic mean temperature difference, °C.

Logarithmic mean temperature difference

\[
\Delta t_{\text{flow}} = \frac{(t_{\text{cover}} - t_{\text{in(flow)}}) - (t_{\text{cover}} - t_{\text{fin(flow)}})}{\ln\left(\frac{t_{\text{cover}} - t_{\text{in(flow)}}}{t_{\text{cover}} - t_{\text{fin(flow)}}}\right)} \tag{10}
\]

Thus, setting the equality condition, the amount of heat given off by the heated lid to the flow of the gas-air mixture and the amount of heat absorbed by the flow of the gas-air mixture \( (Q_{\text{cover}} = Q_{\text{flow}}) \) we obtain the following expression:

\[
\ln\left(\frac{t_{\text{cover}} - t_{\text{in(flow)}}}{t_{\text{cover}} - t_{\text{fin(flow)}}}\right) = \frac{F_{\text{h.t.a.}} \cdot \alpha_{\text{cover}}}{G_{\text{flow}} \cdot c_{p(\text{flow})}} \tag{11}
\]

By taking the logarithm of the right-hand side of the expression and performing the appropriate abbreviations, we obtain the equation for determining the final temperature of the gas flow (temperature at the outlet from the firing holes of the gas burner):

\[
t_{\text{fin(flow)}} = t_{\text{cover}} - \left(t_{\text{cover}} - t_{\text{in(flow)}}\right) \cdot \exp\left(-\frac{F_{\text{h.t.a.}} \cdot \alpha_{\text{cover}}}{G_{\text{flow}} \cdot c_{p(\text{flow})}}\right) \tag{12}
\]

This equation makes it possible to determine the final temperature of the gas flow \( (t_{\text{fin(flow)}}) \) after contact with the inner surface of the lid participating in heat exchange \( (F_{\text{h.t.a.}}) \), with the temperature \( (t_{\text{cover}}) \). Taking into account the design of the burner with a heat divider, the useful heat exchange area consists of the area of the side surface of the cone and the annular surface of the cover.

4. Conclusion

The design of a gas burner with increased thermal efficiency has been developed, equipped with a conical-shaped heat divider installed in the center on the inner side of the cover. Installation of a heat divider allows preheating of the gas-air mixture in the burner housing.
The heat balance of the burner equipped with a heat divider is compiled, taking into account the heat absorbed by the body of the gas burner device with a heat divider. The heat balance allows you to determine the thermal efficiency of the burner of the developed design.

An expression is obtained for determining the final temperature of the gas-air flow after heat exchange with the cover of the gas burner equipped with a heat divider.

It was found that the location of the heat divider in the center on the inner side of the cover allows minimizing or completely eliminating the formation of a stagnation zone. The conical shape of the heat divider provides less resistance to the oncoming flow movement, and also because of its own side surface, it allows you to increase the useful area of the heat exchange surface.

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