INFRARED BURNERS AND AERODYNAMICS OF THE FLOW WITH VARIABLE MASS ALONG THE WAY

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At the end of the SO^{s} and the beginning of the 60^{s} the infrared radiation type gas burners (IRB) were used in many countries The principle of their operation is based on the combustion of prior-made gas/air mixture over a porous or a perforated ceramic nozzle or between two metallic nets Originally developed in England by Bonn (IRB) with porous nozzle were replaced from industrial use by the burners with perforated nozzle thoroughly developed by G. Shvank (FRG) and which were widely used in the countries of Europe and the USA.

For the first time in the Soviet Union IRB were developed and industrially used for the combustion of natural and liquified gases at the Institute of Gas (Academy of sciences of Ukraine) 1955-1959 and later at the Research Institute of Gas (Saratov) 1959-1964 Similar systems were then developed and investigated at Mosgasproject and the Union Research Institute of Gas etc In the early 70s appeared IRB with metal grid radiators. The first types of such burners were developed in the Soviet Union by A.M. Levin and S.S. Salikhadzhayev in 1962-1964. At Polotsk State University the authors of the article and V.V. Dunin and V.A. Terentyev have developed and are developing new types of IRB with grid type radiators. The literature on the subject is extensive and covers the principle of operation, methods of calculations and applications of IRB. We are going to mention here the major specific features of IRB.

The combustion of prior prepared mixture almost completely eliminates the emission of CO with the combustion products. And lowering of the combustion temperature (due to heat radiation from metallic net and flame) permits to avoid the formation of NOx (Nitric Oxide) It is one of the most ecologically acceptable gas burners.

About 50-60 per cent of all amount of heat released during combustion is transmitted from IRB by radiation. It is advisable to heat large industrial premises, cattle-breeding farms and poultry farms with the aid of IRB. In contrast with convection heating such systems provide the saving of heating energy from 40 to 50 per cent and the reduction of capital investment for the construction of heating systems by 50 per cent.

In the CIS countries considerable experience has been gained in using IRB for heating large industrial premises, cattle-breeting farms, poultry farms, for drying and heat-treatment of grain, flour, marmalade, confectionary, tobacco, fish etc.

They are also used for drying plaster in the buildings under construction, for warming up the carters of automobiles in winter, for defrosting several cargoes in railroad wagons, for defrosting tram’s breaking systems and working parts, for defrosting tram-way switches, in paint bake ovens, for drying raw-fur, for drying moulds, for static heat trials of different parts of aircraft etc.

For the last few years has been done a great job on the introduction of IRB for bread baking by the Polotsk State University.

The development of IRB with extended fire nozzle for bread-baking industry and heating of the industrial shops caused particular attention for the use of even rates of gas/air mixture through the holes of a fire nozzle along the burner manifold.

The methods of calculation described in literature [4, 5] were negative as the results of calculations differed from the experiment both in qualitative and quantitative aspects. The authors did not consider all specific features of movement with variable mass as the peculiarities of transition at the dead end of a perforated pipework.

The investigations are based on classical works of G.A. Petrov [6, 7] dealing with the flow movement with variable mass.

The equation that has been deducted by him differs from the commonly used Bemoully's equation in one additional item which considers the effect of varying of the flowing mass on specific energy of the flow

The following equation has been deducted for the established streams in a horizontally positioned pipe at the perpendicular direction of the mass to be separated towards the axis of the main flow

$$\frac{d(\alpha_0 v^2)}{2g} + \frac{dP}{\gamma} + i_f dx + \frac{\alpha_0 v^2}{g}\frac{dQ}{Q} = 0$$

(1)

For the quadratic law of resistance $i_f = \frac{Q^2}{k^2}$. Considering α_0 = *const* after the integration we receive the following equation:

$$\frac{P_{in} - P_x}{\gamma} = \frac{\alpha_0}{gw^2}\left(Q_x^2 - Q_{in}^2\right) + \int_0^x \frac{Q_x^2}{k^2}dx$$

(2)

Having transit Q_t and way Q_w flows the flow value Q_x equals to

$Q_x = Q_t + Q_w - Q_w \frac{x}{l}$ and

$$\frac{P_{in} - P_x}{\gamma} = -\frac{\alpha_0 Q_w^2}{gw^2}\frac{x}{l}\left[2\left(\frac{Q_t}{Q_w}+1\right) - \frac{x}{l}\right] + \frac{1}{k^2}\left[\left(Q_t + Q_w\right)^2 x - \left(Q_t + Q_w\right)\frac{Q_w}{l}x^2 + \frac{Q_w^2 x^3}{3l^2}\right]$$

(3)

When there is no transit flow ($Q_t = 0$) and

$$\frac{P_{in} - P_x}{\gamma} = \frac{Q_w^2}{k^2}\left(x - \frac{x^2}{l} + \frac{x^3}{3l^2}\right) - \frac{\alpha_0 Q_w^2}{gw^2}\frac{x}{l}\left(2 - \frac{x}{l}\right)$$

(4)

for the whole piping (x = *l*)

$$\frac{P_{in} - P_x}{\gamma} = Q_w^2\left(\frac{l}{3k^2} - \frac{\alpha_0}{gw^2}\right)$$

(5)

We equate derivative $\frac{dP_x}{dx}$ = 0 and receive the following equation:

$$\frac{2\alpha_0 k^2}{gw^2 l}\left(1 - \frac{x}{l}\right) - \left(1 - \frac{x}{l}\right)^2 = 0$$

(6)

the roots of which $x_1 = l - \frac{2\alpha_0 k^2}{gw^2}$ equals to minimum and x2 = *l* equals to maximum of piezometric pressure.

The correlation $\frac{l}{3k^2} > \frac{\alpha_0}{gw^2}$ with $P_{in} > P_e$ corresponds to a long piping The correlation $\frac{l}{3k^2} = \frac{\alpha_0}{gw^2}$ with $P_{in} > P_e$ corresponds to a short piping. The correlation $\frac{l}{3k^2} < \frac{\alpha_0}{gw^2}$ with $P_{in} > P_e$ corresponds to a very short piping.

The literature on ventilation [4,5] gives the following equation for the uneven flow of a horizontal pipe and the quadratic resistance law is given as

$$\frac{P_{in} - P_e}{\gamma} = \frac{w_{in}^2}{2g}\left(\frac{\lambda l}{3D_e} - 1\right) \tag{7}$$

At the same time regarding the peculiarities of the movement with variable mass when using similar values gives the following equation:

$$\frac{P_{in} - P_e}{\gamma} = \frac{w_H^2}{2g}\left(\frac{\lambda l}{3D_e} - 2\right) \tag{8}$$

So, the first conclusion is the above mentioned equations when we have even distribution of air flows along the pipeline are not correct It differ from the real ones in the equation describing the pressure drops equal to one pressure head $\left(-\frac{w_{in}^2}{2g}\right)$ instead of $\left(-2\frac{w_{in}^2}{2g}\right)$. That is why the distance from the inlet part of a pipeline till the point where the minimum static pressure is observed is defined in a wrong way.

But there is one more important factor that has been considered neither by G.A. Petrov nor by other authors. All their calculations are made for turbulent flow, not to say about the fact that very often in practice we may come across the cases when the stream of the main flow is subjected to the law of laminar flow. Let us consider the peculiarities of the flow with continuous distribution along the way having no transit flow. In this case the liquid flow changes along the way from initial to zero. So at the end the flow will always be of a laminar type. But at the point where the static pressure has minimum value the movement is a positive longitudinal gradient of static pressure. As a result of this a kind of a diflusor effect takes place. In this case the flow becomes turbulent. And the critical value of Reynolds' number goes down drastically. And having very high levels of positive gradient of pressure the inner layer of flame may be detached.

In both cases when the flow is laminar due to the decrease of a flow rate and turbulization due to a diflusor effect may sometimes level each other Sometimes there is a great difference between experimental results and the results predicted theoretically.

Having the main flow of a laminar type one can integrate a complete equation of the movement with variable mass (I), where $\alpha_0 = 1$ and $\lambda = \frac{64\nu}{w_{in}D_e}$.

The equation is as follows:

$$\frac{P_{in} - P_x}{\gamma} = \frac{w_{in}^2}{2g}\left(\frac{32\nu l}{D_e^2 w_{in}} - 2\right)\frac{x^2}{l^2}$$

(9)

and if $x = l$

$$\frac{P_{in} - P_x}{\gamma} = \frac{w_{in}^2}{2g}\left(\frac{32\nu l}{D_e^2 w_{in}} - 2\right) \qquad (10)$$

Equation (9) proves that static pressure having the flow of a laminar type with even distribution along the way does not have extremum and is a monotonically decreasing function.

The table below presents the main aspects of the flow movement with even distribution of the flow rate along the way.

The basic equations for the flow movement with even distribution of the flow rate along the way.

Type of movement	Quadratic regime (turbulent movement)	Laminar Regime
Unvariable Movement	$\frac{P_{in} - P_e}{\gamma} = \frac{w_{in}^2}{2g}\lambda\frac{l}{D_e}\frac{1}{3}$	$\frac{P_{in} - P_e}{\gamma} = \frac{w_{in}^2}{2g}\frac{32\nu l}{D_e^2 w_{in}}$
Variable Movement	$\frac{P_{in} - P_x}{\gamma} = \frac{w_{in}^2}{2g}\left(\frac{\lambda x}{3D_e} - 1\right)\frac{x^2}{l^2}$ $\frac{P_{in} - P_e}{\gamma} = \frac{w_{in}^2}{2g}\left(\frac{\lambda l}{3D_e} - 1\right)$	$\frac{P_{in} - P_x}{\gamma} = \frac{w_{in}^2}{2g}\left(\frac{32\nu l}{D_e^2 w_{in}} - 1\right)\frac{x^2}{l^2}$ $\frac{P_{in} - P_e}{\gamma} = \frac{w_{in}^2}{2g}\left(\frac{32\nu l}{D_e^2 w_{in}} - 1\right)$
Movement with Variable Mass	$\frac{P_{in} - P_x}{\gamma} = \frac{w_{in}^2}{2g}\left(\frac{\lambda x}{3D_e} - 2\right)\frac{x^2}{l^2}$ $\frac{P_{in} - P_e}{\gamma} = \frac{w_{in}^2}{2g}\left(\frac{\lambda l}{3D_e} - 2\right)$	$\frac{P_{in} - P_x}{\gamma} = \frac{w_{in}^2}{2g}\left(\frac{32\nu l}{D_e^2 w_{in}} - 2\right)\frac{x^2}{l^2}$ $\frac{P_{in} - P_e}{\gamma} = \frac{w_{in}^2}{2g}\left(\frac{32\nu l}{D_e^2 w_{in}} - 2\right)$

The following conclusion can be drawn from the analysis of the above mentioned equations The static pressure at the end of the piping under the above-mentioned circumstances *(Q_t* = 0) will certainly decrease even if at the inlet part the pipeline regime will be turbulent

Conclutions

1. This article presents IRB, their use and applications
2. Presents the solving of equations for a transfer flow with equal rate of dischange along the pipe.

Nomenclature

Q - flow through the given section of the main stream, Q_{in}, Q_x - flows in the inlet section and the section at the distance x from the inlet part of the piping; P - static pressure; P_{in} and P_e - pressure in inlet and outlet sections of the piping, D_e - equvalent diameter, g - acceleration of gravity; v - average velocity at a given section of the main flow; x - horizontal coordinate at the centre of gravity of the given section; i_f - hydraulic tilt; w_{in} - average velocity in the inlet section; α_0 - Cariolis factor; γ - specific gravity; ω - piping cross-section area; λ - resistance factor.

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