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PERFORATED HOUSEHOLD BURNER STRESS MODELLING

Biljana Prochaska[1], Nebojša Radić[2], Predrag Živković[3], Mladen Tomić[4], Jovan Škundrić[1]

1 University of Banja Luka Faculty of Mechanical Engineering,

2 University of East Sarajevo, Faculty of Mechanical Engineering

2 University of Niš, Faculty of Mechanical Engineering

3 University of Novi Sad, Faculty of Technical Sciences

Abstract. *Development and improvement of the premixed gas burners for home heating systems are ongoing challenge. In the era of the fight against pollution, the reduction of undesired combustion products has made these devices very popular. Flexible power modulation of such devices is a great design challenge. Among the basic steps in solving the mentioned problems is modelling of the gas burner's perforated cylinder thermal stress. Accordingly, this study started from a closed analytical form that yields initial values of the stress for uniform temperature distribution, as a base for further research.*

Key words: *Thermal stresses, Stress modelling, Gas burner, Perforated cylinder*

1. INTRODUCTION

The research presented in this paper is focused on the development of new and improved gas burners for condensing boilers in household heating systems. Improvements of the burner construction are subjected to the investigation of thermal stresses, both in steady and unsteady mode of operation. These burners must be flexible and able to achieve large power modulation which requires a construction resistant to varying thermal loads. Different speeds of the gas flow and different heat loads on the cylindrical burner's surface are causing the outer thin-

***Received: January 10, 2023 / February 10, 2023.**
Corresponding author: Biljana Prochaska
University of Banjaluka, Faculty of Mechanical Engineering
e-mail: biljana.prochaska@gmail.com

walled cylinder stresses, which leads to significant deformations and shortening of the burner lifetime. The model yields initial stresses for the burner segment closed by the cover.

2. Research Review

Research related to the improvement of condensing boilers with premixed flame burners are the scope of many ongoing studies. The subject of interest of scientists around the world is the improvement of these boiler's construction considering different aspects. Many are considering the combustion temperature, flow regimes, CO distribution and NOx emissions of the gas burner at different ratios of primary and secondary air as in the paper [1]. Some authors emphasize the necessary optimization of these gas burners construction [2] due to large variations in the gas quality, which affects the combustion, as well as all structural elements of a such device. In the paper [3], the authors emphasize the necessity of designing such a structure that would enable formation of a thin premixed surface flame. This kind of flame ensures low emission of CO and NOx as well as the applicability of a low-level combustion, all of which require a compact construction design. They numerically calculated the flow inside the burner, and the calculated data were used for designing the entire burner system, including the inner thin-walled cylinder. They stressed the importance of uniform flow distribution in order to form a surface flame. The geometry of the burner's inner thin-walled cylinder is studied, which enables the even gas distribution on the outer thin-walled cylinder. The importance of an appropriate choice of the pattern of openings and the burner holder construction was pointed out in [4]. It prevents the flame from rising at the edges of a burner outer cylinder. The authors in [5] investigated outer thin-walled cylinder diameters and the importance of shape perforations on the external distribution network of the plate flat burner due to equalization of the gas flow and the maximum of achieved temperatures, simultaneously considering the reduction of NOx emissions. In the paper [6], the authors emphasize the necessity of understanding burners with premixing as a multidisciplinary problem that requires the equivalent multidisciplinary approach. In the study [7] authors present an analytical solution of stresses and strains in a long operational graded hollow cylinder that is subjected to uniform heat generation and internal pressure. The authors in the paper [8] dealt with the temperatures shift distribution function and thermal stresses of a thick 3D hollow cylinder. They modified the conceptual idea proposed by Khobragade for temperature distribution, stress and strain functions of the hollow cylinder. In the paper [9], the authors performed an analytical solution for radial, tangential and axial thermal stress in a hollow cylinder with uniformly generated heat for convective thermal boundary conditions for heating on the inner surface of the cylinder and cooling on the outer surface of the cylinder. In the paper [10], the author provides an overview of existing approaches and their assumptions in mathematical modelling. It indicates the importance of mathematical and mechanical modelling on different levels, from micro to macro levels. On the other hand, it shows very complex practical requirements for real-time simulations. In the paper [11], the accuracy of the simulated against measured results was analyzed.

3. CONDENSING BOILER COMBUSTION CHAMBER WITH CYLINDER BURNER

Boilers of condensing type have the gas burner within the combustion chamber, as a part of the heat exchanger, Fig. (1).

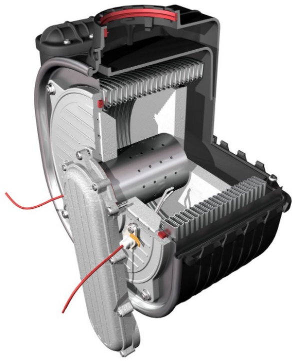
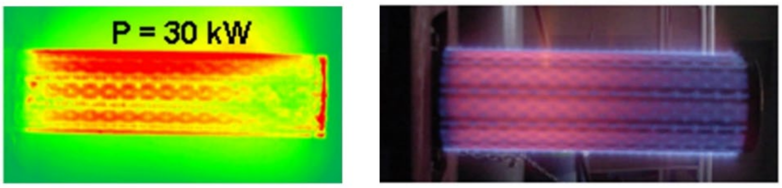


Fig. 1 Combustion chamber with a cylinder burner [12]

Fig. (2) shows a 30 kW atmospheric premixed gas burner, shaped as a perforated cylinder. The photos on the left were taken with a thermal camera and on the right with an optical camera. One can notice the arrangement of the temperature field on the surface of the perforated cylindrical burner. The part of the burner exposed to higher than operating temperatures is red and the part that is under the operating temperatures, is green. The goal is the air/fuel mixture to be ideally prepared in order to reach the goal of the emission levels of NOx, CO and other undesired combustion products, lower than 50 mg/kWh. Fig. (2.b) shows the uneven distribution (decreasing) of the gas flow, which results in the zones of the radiation (yellow) and convective heat transfer (blue). This causes the uneven heat load of the burner outer thin-walled cylinder, which increases the thermal stresses.



a) Thermal imaging *b) Optical imaging*

Fig. 2 Hot and cold areas on the atmospheric premixed burner surface, taken by a thermal (a) and an optical camera (b) [13]

The subject of the stress analysis is a gas burner, a cylinder with diameter of ϕ70 mm and height of 154 mm (Fig. 3). One can notice the required external casing of the gas burner body with a pattern of small circular and slot openings.

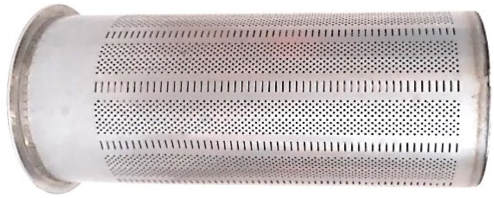


Fig. 3 Cylindric gas burner

4. Stress Model for an Unperforated Cylinder at Constant Thermal Load

As an initial approximation for the stress analysis, a smooth unperforated cylinder was considered. In this paper, a closed analytical model which solves the radial stresses of a homogeneous smooth cylinder depending on the initial temperatures on the outer and inner surfaces of the smooth thin-walled cylinder was considered.

The model was used to test some of the boundary conditions needed for the numerical simulation, which was carried out under several different circumstances. The boundary conditions that do not allow the expansion of the inner and outer radii correspond to some extent to the conditions on the cylinder segment near the top of the burner, which is closed with a cover. This represents a real burner which in steady mode does not change its temperature and dimensions during operation.

In the model, an additional heat flux corresponding to an independent source can be introduced. In a specially conducted numerical simulation, it represents the radiative heat flux in the combustion process simulation. These values are obtained from the fluid dynamic part of the numerical simulation, on the outer shell of the cylinder at a given temperature (gray body radiation effects). Therefore, the stress results obtained do not correspond to a linear temperature increase, as seen in Fig. (6). A good behavior of the model in describing the segment of the cylinder closed by the lid was as expected.

The model described the state of the specified cylinder section, as well as the additional influences on the convection. The stress results obtained by the model differ by about 10% compared to the numerical simulation of the segment near the real cylinder cover results.

The basic nomenclature used is:

$\theta(R) = \dfrac{T(r) - T_1}{T_1 - T_2}$ - Dimensionless temperature

$\theta^* = \dfrac{T_2}{T_1}$ - Dimensionless temperaturne ratio

$R = \dfrac{r}{r_1}$ - Dimensionless radial coordinate

$\rho = \dfrac{r_2}{r_1}$ - Inner to outer radius ratio

h_1, h_2 - Outer and inner convection coefficients

$Q = \dfrac{\dfrac{dq}{dt} r_1^2}{k(T_1 - T_2)}$ - Dimensionless heat flux parameter

$B_i = \dfrac{h_1 r_1}{k}$ - Outer cylinder surface Biot number

$B_{i_2} = \dfrac{h_2 r_2}{k}$ - Inner cylinder surface Biot number

R, t, L - Cylinder mean radius, thickness and length

r, ϕ, z - Cylindric - polar coordinates

$\sigma_r, \sigma_\phi, \sigma_z$ - Radial, tangential and axial stress component

E - Elasticity modulus

ν - Poisson coefficient

α - Thermal dilatation coefficient.

For the initial analysis of the thermal stress, a model of an unperforated cylinder with a given outer r_1 and inner radius r_2 and thermal conductivity k was chosen [9] (Fig. 4).

The stresses were calculated depending on the cylinder outer and inner radius ratio.

The outer and inner surfaces are in contact with the gas with temperature T_1 and T_2 and the associated convection coefficients h_1 and h_2, respectively.

Initial assumption was that the temperature on both surfaces is uniform, so the stresses and strains can be presented in an analytical form.

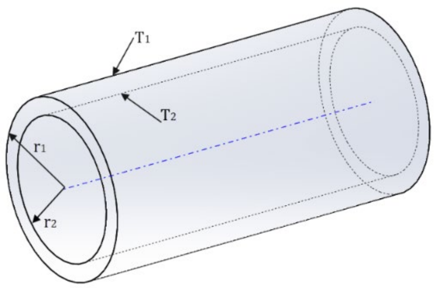


Fig. 4 Homogeneous temperature on the outer T_1 and inner thin-walled cylinder surface T_2

Obtained results are suitable for comparison with a more complex stress/strain model of a thin unperforated cylinder.

The model considered in this paper has the inverse heat flow direction i.e., the gas flowing inside the burner cools the inner surface of the cylinder, while the outer surface is heated, so in this case is $T_1 > T_2$.

The stress components are constant at an arbitrary, fixed radius and depend on the temperature and the corresponding heat flux.

As the temperature in this model is uniform and constant on both the outer and inner cylinder surfaces, the stress is independent of the axial and tangential coordinates z, ϕ.

One-dimensional conduction in the radial direction of the cylindrical-polar coordinate system with uniform heat flow is described by the energy equation:

$$\frac{1}{r}\left(\frac{d}{dr}T(r)\right)+\frac{d^2}{dr^2}T(r)+\frac{1}{k}\frac{d}{dt}q=0 \tag{1}$$

The assumption is that the cylinder is cooled from the inside convectively by a room temperature gas and heated from the outside by convection - radiation effects.

The boundary conditions are:

$$r=r_1: \quad -k\left(\frac{d}{dr}T(r)\right)=h_1(T_1-T(r)) \tag{2}$$

$$r=r_2: \quad -k\left(\frac{d}{dr}T(r)\right)=h_2(T_2-T(r)) \tag{3}$$

All stress coordinates are a function of radial conduction, Poisson's ratio and the radius. By substitution of:

$$\theta(R)=\frac{T(r)-T_1}{T_1-T_2},\ \theta^*=\frac{T_2}{T_1},\ R=\frac{r}{r_1},\ \rho=\frac{r_2}{r_1},\ Q=\frac{\frac{dq}{dt}r_1^2}{k(T_1-T_2)},\ Bi_1=\frac{h_1r_1}{k},\ Bi_2=\frac{h_2r_2}{k}$$

the Eq. 1. can be presented in the dimensionless form:

$$\frac{1}{R}\left(\frac{d}{dr}\theta(R)\right)+\frac{d^2}{R^2}\theta(R)+Q=0 \tag{4}$$

To calculate the thermal stress, we assume that the ends of the cylinder are fixed, that the axial deformation is negligible, and that the radial stresses on the inner and outer surfaces are equal to zero. For these conditions, the radial, tangential and axial stress components σ_r, σ_ϕ, σ_z can be obtained as follows:

$$\sigma_r=\frac{\alpha E}{(1-\nu)r^2}\left(\frac{r^2-r_1^2}{r_2^2-r_1^2}\int_{r_1}^{r_2}T(r)rdr-\int_{r_1}^{r}T(r)rdr\right) \tag{5}$$

$$\sigma_\phi=\frac{\alpha E}{(1-\nu)r^2}\left(\frac{r^2-r_1^2}{r_2^2-r_1^2}\int_{r_1}^{r_2}T(r)rdr-\int_{r_1}^{r}T(r)rdr-T(r)r^2\right) \tag{6}$$

$$\sigma_z=\frac{\alpha E}{(1-\nu)}\left(\frac{2\nu}{r_2^2-r_1^2}\int_{r_1}^{r_2}T(r)rdr-T(r)\right) \tag{7}$$

Inserting the above expression for the dimensionless temperature into the equations for the dimensionless stress in the cylindrical-polar coordinates gives the expressions:

$$\sigma_r^* = \frac{\sigma_r(1-\nu)}{T_1\alpha E} = \frac{1}{R^2}\left(\frac{R^2-1}{\rho^2-1}\int_1^{\rho}\Big((1-\theta^*)\theta(R)+1\Big)RdR + \int_1^{R}\Big((1-\theta^*)\theta(R)+1\Big)RdR\right) \quad (8)$$

$$\sigma_\phi^* = \frac{\sigma_\phi(1-\nu)}{T_1\alpha E} = \frac{1}{R^2}\left(\frac{R^2+1}{\rho^2-1}\int_1^{\rho}\Big((1-\theta^*)\theta(R)+1\Big)RdR + \int_1^{R}\Big((1-\theta^*)\theta(R)+1\Big)RdR\right) \quad (9)$$

$$\sigma_z^* = \frac{\sigma_z(1-\nu)}{T_1\alpha E} = \frac{1}{R^2}\left(\frac{2\nu}{\rho^2-1}\int_1^{\rho}\Big((1-\theta^*)\theta(R)+1\Big)RdR + \int_1^{R}\Big((1-\theta^*)\theta(R)+1\Big)RdR\right) \quad (10)$$

Using the substitutions and solving the system of differential equations by $\theta(R)$, and including the resulting solution in (8-10), after the integration we obtain explicit expressions for σ_r^*, σ_ϕ^*, σ_z^*.

Including the corresponding values for the basic model of the cylinder: the outer burner cylinder (the outer surface is at higher T_1 and the inner surface at lower temperature T_2), with a diameter of 70.4 mm and a wall thickness of 0.5 mm, so r_1 = 35.2 mm, r_2 = 34.70 mm; $Bi_1 = Bi_2 = 0.3$ are constant; $\theta(R)$, θ^* and Q are parameters that change with temperature, and are defined for each calculation separately. The dimensionless stresses σ_r^*, σ_ϕ^*, σ_z^* return to σ_r, σ_ϕ, σ_z.

In order to show the idea of how the problems of non-uniform thermal stress could be solved, when T_1 and T_2 of the cylinder are known, we will observe a narrow ring under different heat load values, shown as a diagram in Fig. (5). Each of these narrow rings has uniform temperature distribution. It can be used to simulate the process at the top of a real cylinder, in the segment near the cover where the diameters do not change.

Based on the analysis of several different ring lengths, temperature values and steps, it was concluded that there are no significant changes, so a more transparent division was chosen for the overall process representation. The chosen division is an acceptable model approximation. To display the obtained results, an increasing series of T_1 in 25 K steps was entered into the calculation of the cylinder outer surface. This increasing temperature series Fig. 5 is stacked starting from zero in steps of 10 mm on the abscissa, in order to clearly see the difference in the entered temperatures. The Fig. (5) represents a stacked series of narrow rings with different temperature values that appear on the real cylinder top near the lid. The temperature T_2 on the inner cylinder surface has values lower than T_1 by 5 K.

One can notice in Fig. (6) we see that stresses do not follow the linear temperature increase. The reason is that different combustion heat loads have different radiation contribution, which is not linear and is already included as an additional heat load.

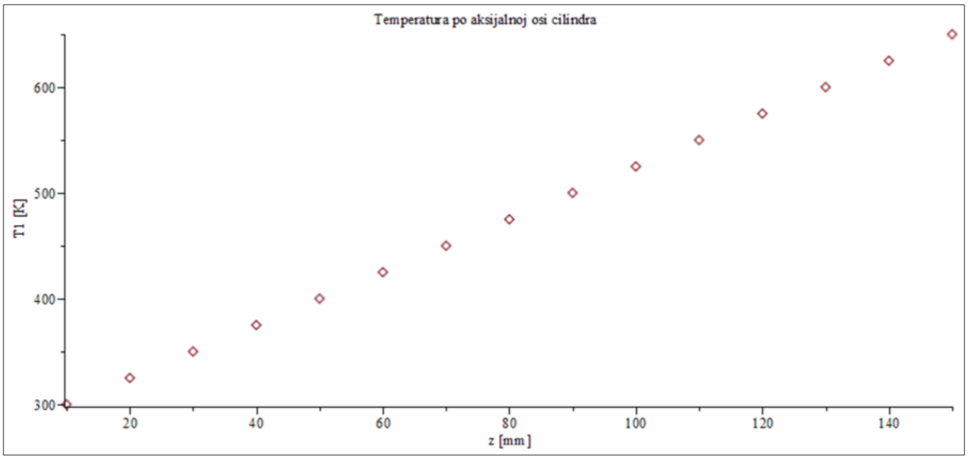


Fig. 5 Temperature distribution along the cylinder

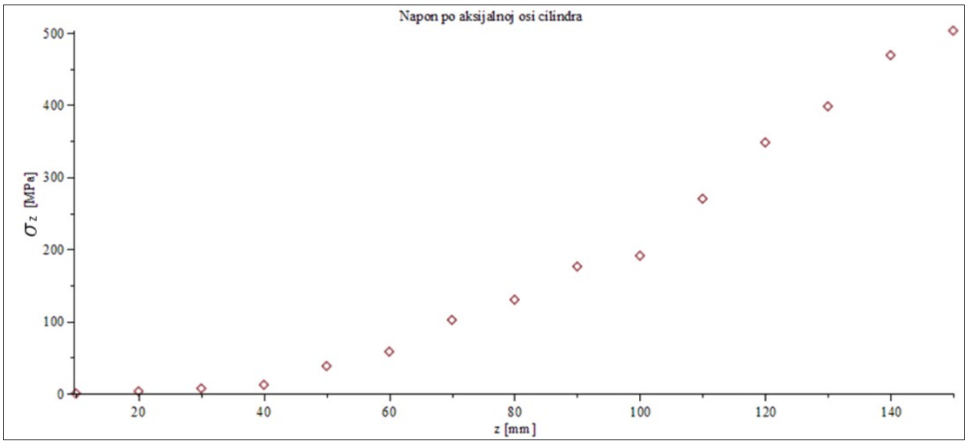


Fig.6 Stress distribution along the cylinder

5. Conclusion

Initial approximation of the gas burner perforated cylinder, with a smooth unperforated cylinder can be analyzed for non-uniform thermal stresses if the surface temperatures of the thin-walled cylinder are known. The model tested the boundary conditions for a series of numerical simulations. The boundary conditions that do not allow the expansion of the inner and outer radii correspond to the cylinder segment near the burner top, closed with a cover. This models a real steady mode burner with constant temperatures and dimensions.

An additional flux corresponding to an independent heat source is introduced. It represents the combustion process radiative heat flux. The values are obtained from the fluid dynamic simulation, on the outer surface of the cylinder (gray body radiation effects). This causes the non-linear stress distribution for linear temperature increase.

A good model behavior for describing the cylinder segment closed by the cover was presented, as the perforations are covered by the lid. For the rest of the cylinder, a model that includes perforations will be developed. The model described the state of the specified cylinder part, as well as the influence of convection and radiation. The stress values from the model deviate ~10% of the numerical simulation results for the same cylinder segment.

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