Modeling of twofold flame behavior of ceramic foam surface burners

Citation for published version (APA):

Severens, P. F. J., Bouma, P. H., Ven, van de, C. J. H., Goey, de, L. P. H., & Drift, van der, A. (1995). Modeling of twofold flame behavior of ceramic foam surface burners. *Journal of Energy Resources Technology : Transactions of the ASME*, 117(1), 29-36. <https://doi.org/10.1115/1.2835317>

DOI:

[10.1115/1.2835317](https://doi.org/10.1115/1.2835317)

Document status and date:

Published: 01/01/1995

Document Version:

Publisher's PDF, also known as Version of Record (includes final page, issue and volume numbers)

Please check the document version of this publication:

- A submitted manuscript is the version of the article upon submission and before peer-review. There can be important differences between the submitted version and the official published version of record. People interested in the research are advised to contact the author for the final version of the publication, or visit the DOI to the publisher's website.
- The final author version and the galley proof are versions of the publication after peer review.
- The final published version features the final layout of the paper including the volume, issue and page numbers.

[Link to publication](#)

General rights

Copyright and moral rights for the publications made accessible in the public portal are retained by the authors and/or other copyright owners and it is a condition of accessing publications that users recognise and abide by the legal requirements associated with these rights.

- Users may download and print one copy of any publication from the public portal for the purpose of private study or research.
- You may not further distribute the material or use it for any profit-making activity or commercial gain
- You may freely distribute the URL identifying the publication in the public portal.

If the publication is distributed under the terms of Article 25fa of the Dutch Copyright Act, indicated by the "Taverne" license above, please follow below link for the End User Agreement:

www.tue.nl/taverne

Take down policy

If you believe that this document breaches copyright please contact us at:

openaccess@tue.nl

providing details and we will investigate your claim.

P. F. J. Severens

P. H. Bouma

C. J. H. van de Ven

L. P. H. de Goey

Eindhoven University of Technology,
Faculty of Mechanical Engineering
(WOC),
WH. 3.138, PO Box 513
5600 MB Eindhoven, The Netherlands

A. van der Drift

Netherlands Energy Research Foundation
(ECN),
Business Unit Fossil Fuels, PO Box 1,
1755 ZG Petten, The Netherlands

Modeling of Twofold Flame Behavior of Ceramic Foam Surface Burners

A model for the flame behavior of a ceramic foam surface gas burner is developed. The burner plate is divided into small segments. Each segment is assumed to behave independently and the transport phenomena in each segment are assumed to be one-dimensional. Relations between the local gas velocity and global pressure drop are derived. The model predicts the pressure drop over the hot burner plate and the radiating fraction of the plate reasonably well.

1 Introduction

Combustion of organic fuels plays a major role in the worldwide pollution of the environment. Some of the most important polluting chemicals arising from the combustion process are nitrogen-oxides (NO_x). The emission of NO_x strongly contributes to the acidifying of the environment. Much research on reducing flame temperature has been carried out recently, since the emission of NO_x increases strongly at increasing flame temperatures. As a result, the principle of the surface burner has been devised. Burner plates of this kind of burner are often fabricated of a suitable foam material. The porous structure of the concerning foam is nearly always inhomogeneous; the resistance to gases flowing through differs over the burner plate. This is of great importance for the flame behavior of the burner.

In Fig. 1, a schematical view of the experimental apparatus is given. The gas flow is controlled by mass flow controllers. The methane-air mixture is mixed before entering the plenum chamber. Above the burner plate, the gas mixture is ignited.

At a relatively low-velocity u_0 (<0.14 m/s in case the air ratio n of the premixed methane-air mixture amounts to 1.3) of the gas mixture flowing towards the burner plate, combustion takes place very close to the surface of the burner plate. In this low-load section of the operation range of the burner, the flame front stabilizes at a very close distance from, or even within, the ceramic material. The flame is cooled by a continuous conduction of heat from the burning zone to the burner plate, which heats up to about 1300 K. The hot burner plate radiates the incoming heat to the surroundings. As a result, the occurring flame tem-

peratures are considerably lower than in the case of most other conventional combustion processes, with smaller heat losses to the burner construction. The NO_x emission is significantly reduced by this method. Another advantage lies in the relatively small flame length. As a result, the combustion chamber can be built very compactly, even when the burner is used at high load operation. Although the flue gas temperature approaches the adiabatic flame temperature, the NO_x emission reduces by this method due to the short residence time.

The occurring flame phenomena are almost one-dimensional in case of a low-velocity u_0 . The burning zone is considered to be one square flat flame covering the entire burner surface. Most papers on this subject, e.g., Kendall and Desjardin (1989) and Beesteheerde et al. (1992), support the assumption of one-dimensional behavior at low-load operation.

At a relatively high-velocity u_0 (>0.30 m/s in case $n = 1.3$), the local velocity of the mixture exceeds the adiabatic burning velocity S_L (≈ 0.26 m/s in case $n = 1.3$; Law (1993)) in most areas of the burner surface and Bunsen-type flames occur. The flame front curves and heat is conducted to the burner plate at the stabilization points only. This is usually referred to as the blue-flame mode. The final flue gas temperature is similar to that in the conventional case and the NO_x production rate is comparable to the rate occurring in conventional combustion processes.

Between low-load operation and high-load operation, a transition behavior is observed. The flame behavior gradually changes from low-load behavior to high-load behavior while passing through this section with increasing u_0 . In case of operation in the transition section, the burner surface shows adjacent radiating and nonradiating zones. These zones have a typical length scale of order 10×10 mm. While increasing the gas velocity u_0 , these zones cool down one after another. The burner plate consists of high-resistance and low-resistance zones. At a

Contributed by the Petroleum Division for publication in the JOURNAL OF ENERGY RESOURCES TECHNOLOGY. Manuscript received by the Petroleum Division, June 21, 1993; revised manuscript received November 15, 1994. Associate Technical Editor: G. A. Karim.

Fig. 1 Experimental apparatus

particular u_0 , the local velocity of the gas mixture does not exceed S_L in a relatively high-resistance zone and the burner surface radiates in these zones. In the relatively low-resistance zones, the local velocity exceeds S_L and Bunsen-type flames occur. This twofold flame behavior is caused by the inhomogeneous structure of the burner plate.

This paper deals with investigations of the flame behavior of the surface burner during operation in the transition section. These investigations are carried out to improve the emission behavior of the gas burner. Especially, the rate of decrease of the radiating fraction γ from 1 to 0, when passing through this section, is of great interest, where the radiating fraction is defined as the radiating surface divided by the total burner surface. For example, if it is possible to reduce the load range of the transition section (maintaining the location of the point where $\gamma = \frac{1}{2}$), the low- NO_x section of the operation range of the burner may be optimized for the particular application of interest. This would offer some important possibilities; for instance, the modulation range of the low- NO_x section could be increased. This aspect is very inviting in view of application of the burner in domestic boilers and drying equipment.

Previous theoretical and numerical investigations on the flame behavior of surface gas burners are restricted to cases of low-load operation, by which the flame front stabilizes at very close distance (Kendall and Desjardin, 1989) or even within (Sathe et al., 1990) the porous burner plate. The models mentioned here are one-dimensional. An appropriate model is developed for the flame behavior in the transition section, since literature does not offer any modeling results regarding the twofold flame behavior in this section.

After a treatise concerning the pressure drop over the burner plate, which appears to be a basic quantity in view of the flame behavior of the surface burner, the combustion model is presented. This local one-dimensional model, which takes the dis-

tribution of local velocities into account, results in predictions for the pressure drop Δp over the burner plate and the radiating fraction γ of the burner surface, both as a function of u_0 and air ratio n . Experiments to verify the predictions as well as a discussion on obtained results are presented afterwards.

The experiments are performed on a ceramic-foam surface burner, developed at E.C.N. (Netherlands Energy Research Foundation); Beestheerde et al. (1992). The porosity ϕ of the ceramic foam is about 90 percent and the diameter of the pores varies between 0.2 and 0.8 mm. The inhomogeneity of the porous structure manifests itself both on the scale of the pores and on larger scales.

2 Pressure Drop Over the Burner Plate

The high temperature of the porous ceramic material at operation in the low-load and transition section induces an increase in the pressure drop Δp over the burner plate, compared to the magnitude of Δp in case of a cold gas flow through the porous structure in the absence of a combustion process. This indicates that the pressure drop plays an important role in the flame behavior of surface burners. A detailed description of the pressure drop of the burner, both in case of the absence of a combustion process ("cold operation") and in case of the presence of a combustion process ("hot operation"), will be presented in this section. In a first approximation, we neglect the effects of inhomogeneities and assume that the transport of mass, momentum, and heat within the burner plate is one-dimensional.

2.1 Cold Operation. The flow of the gas mixture towards the burner plate is uniform with velocity u_0 along the x direction, temperature $T_{g,0}$, and pressure p_0 . A Darcy-Forchheimer equation (Coulaud et al., 1988) describes the pressure gradient at cold operation

$$\frac{dp}{dx} = k_1 \mu u + k_2 \rho u^2 \quad (1)$$

where the Darcy term $k_1 \mu u$ dominates at low filtration velocities u (the velocity outside the burner plate). The inertia term $k_2 \rho u^2$ becomes more important at larger velocities. Quantities μ and ρ represent viscosity and mass density, and k_1 and k_2 are characteristic coefficients of resistance related to the internal structure of the porous material. The total cold pressure drop over the entire plate is given by

$$\Delta p_{\text{cold}} = \int_0^L \frac{dp}{dx} dx = k_1 L \mu_0 u_0 + k_2 L \rho_0 u_0^2 \quad (2)$$

Nomenclature

A = burner surface area (m^2)
 C = temperature coefficient (K)
 c_p = specific heat of gas ($\text{J/kg} \cdot \text{K}$)
 d = characteristic length scale (m)
 E_a = activation energy (J/mol)
 k_i = resistance coefficient ($\text{m}^{-2} \cdot \text{s}$)
 L = thickness of burner plate (m)
 n = air ratio ($n = 1/\phi$) (—)
 p = pressure (Pa)
 R = gas constant ($\text{J/kmol} \cdot \text{K}$)
 R_i = coefficient (m^{-1})
 S = specific area (m^2/m^3)
 T = temperature (K)
 T = tortuosity (m/m)
 T_b = adiabatic flame temperature (K)
 T'_b = nonadiabatic flame temperature (K)

u = velocity (m/s)
 S_L = adiabatic burning velocity (m/s)
 S'_L = nonadiabatic burning velocity (m/s)
 x = position (m)
 α = heat transfer coefficient ($\text{W/m}^2 \cdot \text{K}$)
 γ = radiating fraction of burner surface (—)
 δ = effective thickness (m)
 Δ = difference (—)
 ϵ = emissivity (—)
 λ = thermal conductivity ($\text{W/m} \cdot \text{K}$)
 μ = viscosity ($\text{N/s} \cdot \text{m}^2$)
 ϕ = porosity of ceramic material (m^3/m^3)
 ρ = density (kg/m^3)
 σ = constant of Boltzmann ($\text{W/m}^2 \cdot \text{K}^4$)

Indices

b = burning
cold = referring to cold operation of burner
crit = critical
 g = gas mixture
hot = referring to hot operation of burner
 i = position index
 j = position index
 L = laminar
 p = pores
 s = solid burner material
sink = referring to heat sink
surf = surface of burner plate
0 = referring to gas flow towards burner plate
 $\uparrow$ = referring to increasing load
 $\downarrow$ = referring to decreasing load

where L represents the thickness of the burner plate. The characteristic structure coefficients $k_1(\phi, d_p, d_s, T)$ and $k_2(\phi, d_p, d_s, T)$ are functions of the foam's porosity ϕ , the characteristic length scales of the pores d_p and d_s , where d_p is the pore diameter and d_s the diameter of the solid structure. The tortuosity T is the ratio between the effective flow length L and the actual flow length. From the experiments (see Fig. 2) it is clear that Eq. (2) describes the relation between Δp_{cold} and u_0 with satisfying accuracy.

2.2 Hot Operation. The issue is more complex when combustion takes place. The experimental relation between Δp_{hot} and u_0 is shown in Fig. 2 (upper curve) also. The gas mixture flowing through the burner plate is heated up at low u_0 , due to the high temperature of the ceramic material. As a consequence, the gas stream expands so that the velocity increases inside the foam. In addition, the viscosity μ of the gas rises with the temperature. At low-load operation of the burner, both effects produce an increase of pressure drop, compared to cold operation at the same initial velocity u_0 .

In the first instance, the temperature of the burner plate rises at increasing gas velocity, which causes an increase of the pressure drop. When the gas velocity approaches the adiabatic burning velocity S_L ($S_L \approx 0.26$ m/s), the temperature of the burner decreases, since the heat flux of the flame to the burner decreases. The increase of pressure drop then diminishes also. Finally, when u_0 exceeds S_L , the flame front detaches from the burner surface and Δp_{hot} approximates Δp_{cold} (see Figs. 2 and 4).

For the sake of completeness, it is remarked that the pressure drop over the flame front is neglected in these investigations. From an estimation, it appears that the value of this pressure drop is an order of magnitude smaller than the effects of viscosity and expansion on the difference between Δp_{hot} and Δp_{cold} .

To find an adequate model for Δp_{hot} the processes of heat transport within the burner plate have to be described. The Darcy-Forchheimer Eq. (1) will be used as a starting point.

The temperature dependence of the viscosity μ is taken into account by Bird et al. (1960)

$$\mu(x) = \mu_0 \left(\frac{T_g(x)}{T_{g,0}} \right)^m \quad (3)$$

where $\mu(x)$ and $T_g(x)$ represent the viscosity and the temperature of the gas mixture (methane and air) at position x , and m lies between 0.6 and 0.7 (Bird et al., 1960). The temperature dependence of the mass density is obtained by applying the ideal gas

$$\rho(x) = \rho_0 \frac{T_{g,0}}{T_g(x)} \quad (4)$$

and the filtration velocity u changes due to mass conservation

$$u(x) = u_0 \frac{\rho_0}{\rho(x)} \quad (5)$$

Applying Eqs. (1), (3), (4), and (5), one can derive

$$\frac{dp}{dx} = k_1 \mu_0 u_0 \left(\frac{T_g(x)}{T_{g,0}} \right)^{1+m} + k_2 \rho_0 u_0^2 \frac{T_g(x)}{T_{g,0}} \quad (6)$$

where it is assumed that the temperature increase has no influence on the geometrical structure coefficients k_1 and k_2 . It is also assumed that the flame zone is outside the burner, so the composition of the gas mixture is constant within the burner. By integrating Eq. (6) from $x = 0$ (inflow into the burner plate) to $x = L$ (outflow), the following relation between Δp_{hot} and u_0 is found (with $\Delta p_{\text{hot}} \equiv p(0) - p(L)$):

$$\Delta p_{\text{hot}} = k_1 \mu_0 u_0 \int_0^L \left(\frac{T_g}{T_{g,0}} \right)^{1+m} dx + k_2 \rho_0 u_0^2 \int_0^L \frac{T_g}{T_{g,0}} dx \quad (7)$$

Thus, in order to determine the relation between Δp_{hot} and u_0 , the two integrals in Eq. (7) have to be calculated. To do so, the

Fig. 2 Experimental pressure drop over the burner plate at cold operation (bottom curve) and at hot operation (upper curve)

behavior of the gas temperature $T_g(x)$ within the porous material has to be determined first. This is done by considering the energy balance within the burner plate.

Assume that the combustion processes take place outside the porous plate; then the energy conservation equations for the gas mixture and the solid (ceramic) material are written as follows:

$$\phi \left[\rho_0 \left(\frac{u_0}{\phi} \right) c_p \frac{\partial T_g}{\partial x} - \frac{\partial}{\partial x} \left(\lambda_g \frac{\partial T_g}{\partial x} \right) \right] = \alpha S [T_s - T_g] \quad (8)$$

$$-(1 - \phi) \frac{\partial}{\partial x} \left(\lambda_s \frac{\partial T_s}{\partial x} \right) = -\alpha S [T_s - T_g] \quad (9)$$

where gradients in ϕ are neglected. Note that $\phi = 1$ and $\alpha = 0$ for $x < 0$ or $x > L$. The quantities T_s , c_p , λ_g , λ_s , α , and S represent temperature of the solid, specific heat, conductivity of gas, conductivity of the solid, coefficient of transmission and specific area (wetted surface per unit volume), respectively. Heat radiation of the burner surface is assumed to take place only at the surface interface and is included in the model through the boundary condition at $x = L$. The effect of radiation inside the porous plate may be included in the effective thermal conductive coefficient λ_g of the porous material. When the temperature dependence of c_p , λ_g , λ_s , and α is neglected, a fourth-order linear differential equation for T_g and T_s can be derived

$$\begin{aligned} \rho_0 c_p u_0 \frac{\partial T_{g,s}}{\partial x} - [\phi \lambda_g + (1 - \phi) \lambda_s] \\ \times \frac{\partial^2 T_{g,s}}{\partial x^2} - \rho_0 c_p u_0 \frac{(1 - \phi)}{\alpha S} \lambda_s \frac{\partial^3 T_{g,s}}{\partial x^3} \\ + \lambda_g \lambda_s \frac{\phi(1 - \phi)}{\alpha S} \frac{\partial^4 T_{g,s}}{\partial x^4} = 0, \end{aligned} \quad (10)$$

where $T_{g,s}$ denotes T_g or T_s . The general solution (Jänich, 1983) of Eq. (10) is given by

$$T_{g,s} = C_{0,g,s} + C_{1,g,s} e^{R_1(x-L)} + C_{2,g,s} e^{R_2(x-L)} + C_{3,g,s} e^{R_3(x-L)}, \quad (11)$$

with R_i given by the roots of

$$\begin{aligned} \rho_0 c_p u_0 - (\phi \lambda_g + (1 - \phi) \lambda_s) R_i \\ - \rho_0 c_p u_0 \frac{(1 - \phi)}{\alpha S} \lambda_s R_i^2 + \lambda_g \lambda_s \frac{\phi(1 - \phi)}{\alpha S} R_i^3 = 0, \end{aligned} \quad (12)$$

with

$$R_1 < 0 < R_2 \approx \frac{\rho_0 u_0 c_p}{\phi \lambda_g + \lambda_s(1 - \phi)} < R_3 \quad (13)$$

and

$$C_{0,g,s} = T_0; \quad C_{i,g} = C_{i,s} \left(1 - \frac{(1-\phi)\lambda_s R_i^2}{\alpha S} \right) \quad (14)$$

The term $C_{1,g,s} e^{R_1(x-L)}$ gives only a small contribution in the first part of the burner plate. This term is neglected, as experiments show no significant temperature rise at the inlet. Further, R_3 is at least a factor of ten larger than R_2 . This means that the length scale on which the term with $C_{3,g,s} e^{R_3(x-L)}$ plays a role (θ (0.1 mm)) is much smaller than the average pore diameter d_p . The length scale in which the term with $C_{2,g,s} e^{R_2(x-L)}$ is important (θ (1 mm)) is larger than the pore diameter and is the leading term. The latter indicates that the increase of the gas temperature is significant only in the final part of the burner plate with thickness $\delta = 1/R_2$. The temperature profile can well be approximated by a block profile

$$T_g(x) = \begin{cases} T_{g,0} & \text{if } 0 < x < L - \delta \\ T_{g,\text{surf}} & \text{if } L - \delta < x < L \end{cases} \quad (15)$$

with δ given by

$$\delta = \frac{\phi \lambda_g + \lambda_s (1 - \phi)}{\rho_0 u_0 c_p} \quad (16)$$

The latter approximation is based on the observation that

$$\int_0^L (1 + e^{(x-L)/\delta}) dx = L + \delta(1 - e^{-L/\delta}) \approx L + \delta \quad \text{if } L \gg \delta \quad (17)$$

which can also be obtained using Eq. (15). This simplified temperature profile, Eq. (15), produces the following expression for the integrals in Eq. (7):

$$\int_0^L \left(\frac{T_g(x)}{T_{g,0}} \right)^{1+m} dx = L + \delta \left[\left(\frac{T_{g,\text{surf}}}{T_{g,0}} \right)^{1+m} - 1 \right] \quad (18)$$

$$\int_0^L \left(\frac{T_g(x)}{T_{g,0}} \right) dx = L + \delta \left[\left(\frac{T_{g,\text{surf}}}{T_{g,0}} \right) - 1 \right] \quad (19)$$

Note that the right-hand parts of Eqs. (18) and (19) consist of the result for the case of cold operation and additional terms induced by the temperature increase of the burner plate.

Application of Eqs. (18) and (19) requires data for the gas temperature at burner exit T_{surf} . Since αS is rather large, $C_{2,g} \approx C_{2,s}$. This means that we may use the burner surface temperature $T_{s,\text{surf}}$ instead of $T_{g,\text{surf}}$. The relation between $T_{s,\text{surf}}$ and u_0 can be derived by taking into account energy conservation of the total system and by assuming perfectly one-dimensional flame behavior. Assuming that the heat loss by radiation of the ceramic material takes place at $x = L$ only, then $T_{s,\text{surf}}$ can be calculated by equating the loss in the convective heat transport to the radiative heat transport at the interface $x = L$. $T_{s,\text{surf}}$ can be obtained from

$$\rho_0 u_0 c_p [T_b - T_b'] = \epsilon \sigma [T_{s,\text{surf}}^4 - T_{\text{sink}}^4] \quad (20)$$

where T_b , T_b' , and T_{sink} represent the adiabatic, nonadiabatic flame temperature and the temperature of the heat sink, respectively. For the sake of simplicity, the temperature of the heat sink is set equal to $T_{g,0}$. The quantity σ represents the Stefan-Boltzmann constant. Note that c_p is assumed to be constant in the temperature range between T_b' and T_b . Also, the value of the effective emissivity ϵ of the burner surface is assumed to be independent of the temperature of the ceramic material.

For a burner-stabilized flat flame, it is known that the mixture velocity u_0 is equal to the nonadiabatic burning velocity S_L' . A lot of research has been done on the relation between S_L' and T_b' for flat flames, both experimentally (Kaskan, 1967; Ferguson and Keck, 1979; Matsui and Yuuki, 1985) and theoretically (Williams, 1965; Yuuki and Matsui, 1987). Equation (21) is

used for simplicity, by which the value of an effective activation energy E_a has been fitted to experimental results (Kaskan, 1967)

$$u = k e^{-E_a/2RT_b'} \quad (21)$$

This relation only holds when the reaction zone is outside the foam. The preexponential factor k is related to the adiabatic flame temperature T_b and the adiabatic burning velocity S_L . The relation between S_L' and T_b' then finally reads

$$\frac{S_L'}{S_L} = \exp \left(-\frac{E_a}{2RT_b'} + \frac{E_a}{2RT_b} \right) \quad (22)$$

Note that Eq. (22) requires experimental values of S_L and T_b . Combination of Eqs. (20) and (22) gives expressions for T_{surf} at every value of the gas velocity u_0

$$T_{s,\text{surf}}^4 = \begin{cases} T_{g,0}^4 + \frac{\rho_0 u_0 c_p T_b}{\epsilon \sigma} \left[1 - \frac{E_a}{2RT_b \ln \left(\frac{u_0}{S_L} \right)} \right]^{-1} & \text{if } u_0 < S_L \\ T_{g,0}^4 & \text{if } u_0 > S_L \end{cases} \quad (23)$$

The result of Eq. (23) is shown in Fig. 3. Starting from low u_0 , the value of T_{surf} rises to a maximum at increasing load of the burner; after that, T_{surf} decreases. Although the flame temperature rises at increasing u_0 , less diffusive heat from the flame front reaches the burner surface; the convective transport by the (increasing) flow of the gas mixture dominates. The flame detaches from the surface when u_0 exceeds the adiabatic burning velocity S_L , so that the temperature of the burner plate reduces to $T_{g,0}$. The latter effect is observed experimentally in an indirect way. The measured difference between Δp_{hot} and Δp_{cold} is negligible in the high load section. This implies that $T_g(L) \approx T_{g,0}$, so $T_{\text{surf}} \approx T_{g,0}$.

Now that expressions are found for both integrals in Eq. (7) and for T_{surf} , both the cold and hot pressure drops can be determined, when values for the structure coefficients are known. In Fig. 4, experimental and theoretical curves of Δp_{cold} and Δp_{hot} are shown. The resistance factors k_1 and k_2 are obtained by fitting the experimental results of Δp_{cold} . As can be seen in the figure, the fitting results of Δp_{cold} are good, which indicates that the Darcy-Forchheimer relation holds.

It is found that the combination of Eqs. (7), (16), (18), (19), and (23) describes the relation between Δp_{hot} and u_0 well, both at operation in the low-load section ($u_0 < 0.14$ m/s) and at op-

Fig. 3 Calculated surface temperature of the burner plate, assuming one-dimensional flame behavior (numerical result of Eq. (23) with $n = 1.3$, $\rho_0 = 1.233$ kg/m³, $c_p = 1000$ J/kg·K, $T_b = 2.0 \cdot 10^3$ K, $S_L = 0.26$ m/s, $E_a = 1.38 \cdot 10^5$ J/mol (de Lange, 1992), $T_{g,0} = 293$ K and $\epsilon = 0.4$)

Fig. 4 Pressure drop versus velocity. Markers: measurement points of Δp_{cold} and Δp_{hot} . Lines: calculated curves of $\Delta p_{\text{cold}}(u_0)$ and $\Delta p_{\text{hot}}(u_0)$, assuming one-dimensional flame behavior, applying experimental values of k_1 and k_2 (numerical result of Eqs. (7), (18), (19), and (23) with $n = 1.3$, $\rho_0 = 1.233 \text{ kg/m}^3$, $\mu_0 = 1.78 \cdot 10^{-5} \text{ N/s} \cdot \text{m}^2$, $c_p = 1000 \text{ kJ/kg} \cdot \text{K}$, $k_1 = 2.9 \cdot 10^2 \text{ m}^2$, $k_2 = 2.4 \cdot 10^2 \text{ m}^{-1}$, $T_b = 2.0 \cdot 10^3 \text{ K}$, $S_L = 0.26 \text{ m/s}$, $E_a = 1.38 \cdot 10^6 \text{ J/mol}$, $T_{g,0} = T_g(0) = 293 \text{ K}$, $\epsilon = 0.4$, $L = 15 \text{ mm}$, $\phi = 0.9$, $\lambda_g = 0.03 \text{ W/m} \cdot \text{K}$, $\lambda_s = 2.0 \text{ W/m} \cdot \text{K}$ and $m = 0.65$).

eration in the high-load section ($u_0 > 0.30 \text{ m/s}$). However, at operation in the transition section the one-dimensional modeling of the flame behavior is inadequate. According to the model T_{surf} equals $T_{g,0}$ if u_0 exceeds the adiabatic burning velocity, so that $\Delta p_{\text{hot}} = \Delta p_{\text{cold}}$ if $u_0 > S_L$. Figure 4 shows that this result is not in accordance with the experimentally observed behavior of the burner. The one-dimensional model predicts a maximum in the pressure drop. This maximum is not observed experimentally. Furthermore, for values of u_0 exceeding S_L , several high-resistance zones of the burner plate are still hot, because the local gas velocity does not exceed S_L yet. Experimentally, the value of Δp_{hot} equals that of Δp_{cold} when all zones of the burner plate have cooled down.

To find a model for Δp_{hot} that describes the behavior in the transition section adequately, the twofold appearance of the flame behavior must be taken into account. The local one-dimensional model, that is presented in the next section, meets this need.

3 Local One-Dimensional Model

At operation in the transition section, the inhomogeneous resistance of the porous burner plate to gases flowing through gives rise to the simultaneous appearance of cold and hot areas on the burner surface. It is observed that the typical dimensions of these zones are of order $10 \times 10 \text{ mm}^2$. The flame locally detaches from the surface in the cold areas and the occurring combustion phenomena are obviously three-dimensional, which means that detailed flame modeling is an extremely complicated matter. The flow within the burner, however, is assumed to remain one-dimensional. The flame behavior can be modeled in the radiating areas (high-resistance areas) of the burner surface, since the occurring flame phenomena in these areas are comparable to those at operation in the low-load section of burner. The combustion in these areas may be treated one-dimensionally. The equations presented in Section 2 can be used to obtain the local relations between the local filtration velocity and the pressure drop.

In the remaining areas (low-resistance areas) of the surface, the local velocity is larger than the adiabatic burning velocity. The burner plate is then cold which gives reason to assume that the combustion is almost adiabatic. The local details of the flame structure in the cold areas are not of much interest, but only the average values of, for instance, the pressure drop, velocity and flame temperature. So it suffices to assume a uniform flame temperature in these areas. Since the heat transfer from the flame front to the burner plate is negligible in the cold areas, the local

flame temperature is assumed to be almost equal to the adiabatic value T_b .

Whereas the resistance of the porous material differs over the burner plate, the pressure drop Δp over the burner plate can be assumed equal in each zone of the plate. Note that the measured values of Δp are larger by two orders of magnitude than the actual dynamic pressure $\frac{1}{2} \rho u^2$. The combination of uniform pressure drop over the plate and inhomogeneity of the porous material causes the velocity of the gas mixture over the burner plate to fluctuate. This effect makes the relation between Δp and the local velocity one of the basic considerations for the flame behavior of the burner. The high-resistance and low-resistance zones of the burner plate are joined by the constant pressure drop over the burner plate. This forms the basis of a local one-dimensional flame model, by which the average flame behavior at operation in the transition section will be described.

3.1 Local Relations Between the Pressure Drop and the Velocity. The inhomogeneity of the porous structure induces differences in the local velocity of the gas mixture, flowing through the plate. When the statistical distributions of the structure coefficients k_1 and k_2 among the areas of a certain burner plate are known, Eq. (2), with a constant value for Δp_{cold} , offers the distribution of the local gas velocity at cold operation of the burner. The statistical distributions of k_1 and k_2 , in their turn, are governed by variations in geometric parameters like the porosity ϕ , the characteristic length scales d_p and d_s , and the tortuosity T . This means that the distribution of the local gas velocity can be determined by examining the distributions of ϕ , d_p , d_s , and T .

Alternatively, the distributions of k_1 and k_2 can be approximated by measurement of the relation between Δp_{cold} and the local gas velocity $u_{0,i,j}$. Then, the burner plate has to be separated in $N \times N$ small segments (i, j) with surface area $A_{i,j}$ and thickness L . The dimensions of the surface areas of the segments are chosen with the same order of magnitude as the cold and hot areas of the burner surface ($\theta(10^2 \text{ mm}^2)$). This method will be applied here to determine the distribution of the local gas velocity. The flow of the gas mixture in a small segment is assumed to be uniform and one-dimensional and the transport between adjacent segments of the burner plate is neglected. The local coefficients $k_{1,i,j}$ and $k_{2,i,j}$ are found by assuming that the cold pressure drop over a small segment can be described well by a Darcy-Forscheimer equation. This is permitted, because the dimensions of a segment are significantly larger than the size of the pores d_p .

At operation of the burner in the transition section, the distribution of the local resistance of the ceramic burner plate strongly differs from the distribution during cold operation (with the same initial velocity u_0). The high-resistance zones of the burner plate are hot, which produces a local increase of resistance to gases flowing through due to the high temperature of the unburnt gases. The low-resistance zones of the plate are relatively cold, so the local increase of the flow resistance is negligible. This phenomenon results in an increase of the velocity differences over the burner plate; the low gas velocities in the high-resistance zones are reduced and, in view of conservation of mass, the gas velocities in the low-resistance zones are increased.

To model this behavior, the segments are assumed to behave independently. We neglect exchange of both mass and heat between adjacent segments of the burner plate. The pressure drop over the different segments is the same for all the segments. Taking into account the latter arguments, the pressure drop over the segments is given by Eq. (7) combined with Eqs. (16), (18), (19), and (23). The average surface temperature $T_{\text{surf},i,j}$ for a cold segment (i, j) is assumed to be equal to $T_{g,0}$. The relation between Δp_{hot} and $u_{0,i,j}$ can be calculated for each small single segment (i, j) of the burner plate, whenever the values of $k_{1,i,j}$ and $k_{2,i,j}$ are known.

The one-dimensional model, presented in Section 2, can be applied on a single segment. As already mentioned, the modeled pressure drop curve (Fig. 4) shows that Δp_{hot} has a local maxi-

mum. Figure 5 shows this curve for one single segment with the local maximum of Δp_{hot} at $u = u_2 < S_L$. Starting at $u < u_2$ and by increasing load, a segment radiates until the local gas velocity reaches u_2 . Since the pressure drop over the burner plate increases at increasing load (see Fig. 2 or 4), the gas velocity can only stabilize at u_3 when the load is increased further. Starting at a value $u > S_L$ and by decreasing the load, the segment is non-radiating until the local velocity equals S_L . The flame will now stabilize with velocity $u \leq u_1$ after further reduction of the load. This gives occasion to define two critical pressure drops of a single segment, one for an increasing load ($\Delta p_{\text{crit};i,j} \uparrow$), and one for a decreasing load ($\Delta p_{\text{crit};i,j} \downarrow$); see Fig. 5.

A segment (i, j) is hot when $\Delta p_{\text{hot}} < \Delta p_{\text{crit};i,j}$ and cold when $\Delta p_{\text{hot}} > \Delta p_{\text{crit};i,j}$. The quantity $\Delta p_{\text{crit};i,j}$ is an important tool in view of modeling the average behavior of the burner. To evaluate $\Delta p_{\text{crit};i,j} \uparrow$, u_2 has to be calculated. Therefore, one has to solve

$$\left. \frac{d\Delta p_{\text{hot};i,j}}{du_{i,j}} \right|_{u=u_2} = 0 \quad (24)$$

$\Delta p_{\text{crit};i,j} \uparrow$ is then given by

$$\Delta p_{\text{crit};i,j} \uparrow = k_1 \mu_0 u_2 \left(L + \delta \left[\left(\frac{T_{\text{surf}}(u_2)}{T_{g,0}} \right)^{1+m} - 1 \right] \right) + k_2 \rho_0 u_2^2 \left(L + \delta \left[\left(\frac{T_{\text{surf}}(u_2)}{T_{g,0}} \right) - 1 \right] \right) \quad (25)$$

$\Delta p_{\text{crit};i,j} \downarrow$ is easier to evaluate, since it is found when $u = S_L$. It is given by

$$\Delta p_{\text{crit};i,j} \downarrow = k_{1;i,j} L \mu_0 S_L + k_{2;i,j} L \rho_0 S_L^2 \quad (26)$$

The difference between $\Delta p_{\text{crit};i,j} \uparrow$ and $\Delta p_{\text{crit};i,j} \downarrow$ causes a hysteresis effect in the flame behavior: the burner behaves differently at increasing and decreasing load. The magnitude of this effect depends on the thickness of the burner plate, since the additional pressure drop arises only in the final part of the burner plate. This means that the hysteresis effect is not observed when L is too large, since the cold pressure drop is then much larger than the extra hot contribution. To illustrate the hysteresis effect, three hypothetical pressure drop curves are shown in Fig. 6. At increasing load two of the three segments will radiate at the given Δp , while at decreasing load only one segment will radiate.

3.2 Radiating Fraction of the Burner Plate. For known $k_{1;i,j}$ and $k_{2;i,j}$ and given pressure drop Δp_{hot} , the value of $u_{0;i,j}$ is calculated in each segment independently. In the $(1 - \gamma)N^2$ segments where $\Delta p_{\text{crit};i,j}$ is lower than the specific value of Δp_{hot} , the ceramic material is cold and the values of $u_{0;i,j}$ can be calculated by inverting the Darcy-Forchheimer Eq. (2) for each cold

Fig. 5 Schematic plot of cold and hot pressure drop over one segment of the burner plate, with the two values for the critical pressure drop (see text)

Fig. 6 Schematic behavior of three segments of the burner plate at given Δp (see text)

segment (i, j) . The remaining γN^2 segments, with $\Delta p_{\text{crit};i,j} > \Delta p_{\text{hot}}$, are hot and the values of $u_{0;i,j}$ can be calculated by solving the inverse of Eq. (7) for each hot segment (i, j) . Finally, u_0 is given by

$$u_0 = \frac{\sum_{i,j} u_{0;i,j} A_{i,j}}{\sum_{i,j} A_{i,j}} \quad (27)$$

This procedure gives a result for u_0 for every value of Δp_{hot} and produces the final relation $\Delta p_{\text{hot}} = \Delta p_{\text{hot}}(u_0)$ for operation in the transition section.

The relation between the radiating fraction γ of the burner surface and the hot pressure drop over the burner plate is given by

$$\gamma = \frac{\sum_{i,j} A_{i,j} (\Delta p_{\text{crit};i,j} > \Delta p_{\text{hot}})}{\sum_{i,j} A_{i,j}} \quad (28)$$

Using the obtained relation between Δp_{hot} and u_0 , one may derive the radiating fraction as a function of the average velocity of the gas mixture: $\gamma = \gamma(u_0)$.

4 Experimental Results and Discussion

To verify the modeling results of the previous sections, a set of experiments has been carried out. After describing the experimental setup, the obtained results are presented and discussed.

The experiments have been carried out on flat square burner plates with thickness $L = 15$ mm and area 110×110 mm². To avoid interferences by the presence of adjoining construction parts of the test burner, measurements have been carried out on a restricted test area of the plate. The mixture velocity u_0 and the air ratio n are adjusted by mass-flow-controllers for regulating the air and methane flows.

In view of the local one-dimensional model presented in Section 3, $N \times N = 10 \times 10$ circular segments (i, j) are defined within the test area. Each segment has a diameter of 10 mm, as the magnitude of the smallest occurring hot radiating areas of the burner surface are of the same order of magnitude. To determine the 100 values of $k_{1;i,j}$ and $k_{2;i,j}$, local velocity measurements at various values for Δp_{cold} are carried out. Application of the burner model requires data of the local velocity $u_{0;i,j}$ for each segment (i, j) . The burner construction does not allow velocity measurements in the gas stream flowing through the burner plate. Since transport between adjacent segments is neglected, the average velocity of the mixture flowing through segment (i, j) equals the average velocity just after flowing out of the segment.

The local velocities $u_{0;i,j}$ are measured by hot wire probes. A cylindrical thin-walled tube with inside diameter 10 mm is placed on the surface of segment (i, j) . The unknown velocity profile of the gas flowing out of the porous material gradually develops into a parabolic profile while flowing through the tube. The tube

Fig. 7 Calculated curves of $\Delta p_{\text{cold}}(u_0)$ for six randomly chosen segments, applying experimental values of $k_{1,i,j}$ and $k_{2,i,j}$

is long enough to grant full development into parabolic flow. At the end of the tube, the top velocity is measured by a hot wire probe in the center of the tube.

The pressure drop Δp over the burner plate is determined by measuring the difference between the static pressure of the gas flow just before flowing into the plate and the atmospheric pressure. The radiating fraction γ of the test area is measured photographically. Photographs of the test area at operation in the transition section are analyzed for a large number of values for u_0 , and γ is determined for each single photograph, after dividing the pictures into 100 segments.

The pressure drop Δp_{cold} , as well as $u_{0,i,j}$ in each single segment, are registered at ten values for the mixture velocity u_0 through the burner plate. For each segment, the values of $k_{1,i,j}$ and $k_{2,i,j}$ are determined numerically by fitting a parabola through the measuring points of Δp_{cold} and $u_{0,i,j}$ (according to Section 3.1). Next, $\Delta p_{\text{crit},i,j}$ is calculated for each segment. For six randomly chosen segments, the shape of the profiles of $\Delta p_{\text{cold}}(u_{0,i,j})$ is shown in Fig. 7 based on the measured values of $k_{1,i,j}$ and $k_{2,i,j}$. The variations of the curves appears to be significant, which indicates that inhomogeneities are well observable on the measurement scale.

At availability of $k_{1,i,j}$, $k_{2,i,j}$ and $\Delta p_{\text{crit},i,j}$, the hot pressure drop $\Delta p_{\text{hot}}(u_0)$ over the total burner plate and the radiating fraction $\gamma(u_0)$ of the test area are calculated (according to Section 3.2). Note that the total area, where measurements are performed, is smaller than the total burner area which leads to errors in u_0 and γ . Finally, the experimental data are compared with calculated results.

Figures 8 and 9 show the results of the local one-dimensional model: the calculated curves of $\gamma(u_0)$ and the calculated $\Delta p_{\text{hot}}(u_0)$, while passing through the transition section, both at increasing u_0 and at reducing u_0 , are presented by continuous lines and dashed lines, respectively. The experimental results of both quantities (circles and crosses) are shown as well.

The experimental results of $\Delta p_{\text{hot}}(u_0)$ are well reproduced by the local 1-D model. Note that the hysteresis effect is hardly visible in modeling results given in Fig. 9. This is partly related to the small slope of the pressure as function of the velocity in the transition zone, i.e., relatively large variations in velocity give small changes in the pressure drop. Experimentally, there was no hysteresis observed within the error range.

Both experimental and theoretical hysteresis behavior are seen in Fig. 8 for $\gamma(u_0)$. The experimental relation between u_0 and Δp_{hot} is used to evaluate $\gamma(u_0)$ (see Section 3.2), to avoid propagation of modeling errors. It appears that the model gives an acceptable description of the real behavior of the burner at decreasing load. The deviations between model and observations in $\gamma(u_0)$ are larger at increasing load. Part of these differences originates from measuring errors. How-

Fig. 8 Radiating fraction as function of velocity. Markers: measurement points of $\gamma(u_0)$. Lines: calculated curves of γ , applying experimental values of $k_{1,i,j}$ and $k_{2,i,j}$

ever, another part may be contributed by the neglect of exchange between the segments. The transport among the segments is neglected in the model, but a simple analysis shows that the transport may be significant in the area close to the border of radiating and nonradiating segments. The extra pressure drop has its origin in the thin hot layer with thickness δ of the burner plate, and may rise up to approximately 50 percent of the total cold pressure drop. This causes a redistribution of the flow just before entering the hot layer, amplifying the velocity differences between hot and cold segments even further. This effect may cause a significant broadening of the transition zone, as the experiments clearly show.

5 Conclusions and Recommendations

A local 1-D model has been proposed to describe the behavior of the ceramic foam surface burner in the transition section between low-load and high-load operation. The pressure drop as well as the radiation fraction can be predicted reasonably, both for increasing and decreasing load. Also, on the basis of this model it is possible to estimate the local flame temperatures, Eq. (22), as well, which is a measure for the local thermal NO_x production rate.

The results indicate that there is no need to describe the burner plate on the scale of the pores, although the local resistance coefficients $k_{1,i,j}$ and $k_{2,i,j}$ depend on local geometrical parameters, such as the porosity ϕ , the characteristic length scales d_p and d_s , and the tortuosity T . Whenever it would be possible to calculate the statistical distributions of k_1 and k_2 starting from the distri-

Fig. 9 Hot pressure drop Δp_{hot} . Markers: measurement points. Lines: calculated curves, applying experimental values of $k_{1,i,j}$ and $k_{2,i,j}$

butions of ϕ , d_p , d_s , and T , the flame behavior of the burners may be improved by modification of the porous structure of the ceramic foam.

Further modeling improvement is expected when transport among the segments is included. In the low and high-load section, the exchange between the segments can be neglected; but in the transition section, the exchange may play a significant role. The latter phenomenon will be investigated in the near future.

Acknowledgment

The support of ECN and NOVEM, The Netherlands, is gratefully acknowledged.

References

- Beesteheerde, J., Beckers, G. J. J., Visser, P. R., Tjeng, S. L., and van der Drift, A., 1992, "Low NO_x Ceramic Foam Burner for Domestic and Industrial Utilisation," *Proceedings of the International Gas Research Conference*, Gas Research Institute, Vol. 5, Orlando, FL.
- Bird, R. B., Stewart, W. E., and Lightfoot, E. N., 1960, *Transport Phenomena*, Wiley (international edition).
- Coulaud, O., Morel, P., and Caltagirone, J. P., 1988, "Numerical Modelling of Nonlinear Effects in Laminar Flow Through a Porous Medium," *Journal of Fluid Mechanics*, Vol. 190, pp. 393-407.
- Ferguson, C. R., and Keck, J. C., 1979, "Stand-Off Distances on a Flat Flame Burner," *Combustion and Flame*, Vol. 34, pp. 85-98.
- Jänich, K., 1983, *Analysis für Physiker und Ingenieure*, Springer-Verlag.
- Kaskan, W. E., 1967, "The Dependence of Flame Temperature on Mass Burning Velocity," *Proceedings of the Sixth Symposium (International) on Combustion*, The Combustion Institute, pp. 134-143.
- Kendall, R. M., and Desjardin, S. T., 1989, *Basic Research on Radiant Burners*, Annual Report, Gas Research Institute, Alzeta, CA.
- de Lange, H. C., 1992, "Modelling of Premixed Laminar Flames," Ph.D. thesis, Eindhoven University of Technology, Eindhoven, The Netherlands.
- Law, C. K., 1993, "A Compilation of Experimental Data on Laminar Burning Velocities," *Reduced Kinetic Mechanisms for Applications in Combustion Systems*, eds., N. Peters and B. Rogg, pp. 19-34.
- Matsui, Y., and Yuuki, A., 1985, "Radical Concentration and Prompt NO Formation in Hydrocarbon-Air Premixed Flames," *Japan Journal of Applied Physics*, Vol. 24, pp. 598-603.
- Sathe, B., Peck, E., and Tong, W., 1990, *Combustion and Heat Transfer in Porous Media*, Final Report, Arizona State University.
- Williams, F. A., 1965, *Combustion Theory*, Addison-Wesley.
- Yuuki, A., and Matsui, Y., 1987, "A Study of Flat Flames on Porous Plug Burners: Structure, Standoff Distance, and Oscillation," *Combustion and Flame*, Vol. 69, pp. 37-48.