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Combustion and Heat Transfer in Porous Media

by

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ABSTRACT OF THESIS

STUDY OF A POROUS RADIANT BURNER

The objective of the present study is to generate fundamental knowledge about heat transfer and combustion in porous radiant burners (PRBs) in order to improve their performance. A theoretical heat transfer and combustion model is developed to study the characteristics of PRBs. The model accounts for non-local thermal equilibrium between the solid and gas phases. The solid is assumed to absorb, emit and scatter radiant energy. Combustion is modeled as a one-step global reaction. It is revealed that the flame speed inside the porous medium is enhanced compared to the adiabatic flame speeds due to the higher conductivity of the solid compared to the gas as well as due to radiative preheating of the reactants. The flame speeds near the edges of the porous medium are controlled by conduction alone. Two spatial domains of stable flame speeds are predicted. One spanned the upstream half of the porous region and the other was a narrow region near the downstream edge. The effects of the properties of the porous material on the flame speeds, radiative outputs and efficiencies were investigated. To improve the radiative output from the burner, it is desirable that the porous layer has an optical thickness of about ten. The radiative output and the efficiency is higher for lower scattering albedo. The heat transfer coupling between the solid and gas phases should be high enough to ensure local thermal equilibrium, by choosing a fine porous matrix. Higher solid phase conduction enhances the flame speed and the radiative output. Experiments are performed on a ceramic foam to verify the theoretical findings. The existence of the two stability regions was verified experimentally. The experimental data on flame speeds and radiant outputs for flames stabilized inside the porous medium agreed reasonably well with the theoretical predictions.

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Forward

This report documents the results of a study of combustion and heat transfer from premixed, gaseous fuel-air mixtures in inert porous media. The objective was to generate fundamental information about flame propagation and heat release behavior in porous materials.

A theoretical model of the combustion and multimode heat transfer processes in planar porous media as a function of the relevant factors has been developed. The model includes finite-rate reaction kinetics and the equation of transfer for describing radiative heat transfer in porous media. Calculations revealed a unique spatial variation in elevated, stable flame speeds that was linked to the relative importance of radiative and conductive heat fluxes. Parametric studies demonstrated the influence of porous material properties (optical depth, scattering albedo, and thermal conductivity), stoichiometry, convective coupling and heat sink temperature and reflectivity on the burning rate and external radiant output when the porous matrix was used as a porous radiant burner.

Experiments were performed to obtain flame speed, product species concentration and temperature profiles, and energy output data under various operating conditions for comparison to theoretical predictions. Results for ceramic foam of low optical depth have confirmed the existence of two stable combustion zones, one in the upstream half of the porous material and the other at the downstream edge. Measured flame speeds and radiant output are also in close agreement with predicted values.

The findings will be useful for applying the relatively unexploited technology of porous radiant burners to industrial heating systems, yielding potentially large savings in energy consumption and operating costs. The results will also provide a scientific basis for understanding combustion and heat transfer in other energy technologies including catalytic combustors and combustion in packed beds and geologic systems.

This report is based on information contained in the doctoral dissertation of S.B. Sathe. Further documentation may be found in the following list of publications. In addition, the work has led to the award of a U.S. patent for an improved porous burner concept.

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NOMENCLATURE

a	surface area per volume of solid [1/m]
A	pre-exponential factor [1/s]
A _d	detector area [m ²]
A _p	pinhole area [m ²]
b ₀ -b ₃	constants in rotameter calibration
c	specific heat capacity at constant pressure [J/kg-K]
C _a	cross-sectional area [m ²]
d	particle diameter [m]
D _o	detector output
E	activation energy [J/mole-K]
f	float reading
G	non-dimensional emissive power, $\sigma T^4/Qx_3$
h	heat transfer coefficient [W/m ² -K]
h _k	species specific enthalpy [J/kg]
H	heat of reaction [J/mole-K]
H ₁	dimensionless heat transfer coefficient, $hL/\sigma T_i^3$
H ₂	dimensionless thermal conductivity, $k_s/\sigma T_i^3 L$
H ₃	Biot number, hL/k_s
i	radiant intensity [W/m ² -sr]
I	non-dimensional radiant intensity, i/Qx_3
k	thermal conductivity [W/m-K]

L	porous layer length [m]
m	mass flow rate [kg/s]
n	number of moles
p	pressure [N/m ²]
P ₁ -P ₅	non-dimensional parameters in Table 2.1
q ^r	radiant heat flux [W/m ²]
q _c	reaction enthalpy flux [W/m ²]
q ^c	conductive flux [W/m ²]
Q	volumetric heat generation rate [W/m ³]
Q _c	dimensionless enthalpy flux, $q_c/\sigma T_i^4$
Q _e	extinction efficiency
Q ^c	dimensionless conduction flux, $q^c/\sigma T_i^4$
Q ^r	dimensionless radiant heat flux, $q^r/\sigma T_i^4$
Q [*]	dimensionless radiant heat flux, q^r/Qx_3
Q _c ^r	dimensionless radiant output, $Q^*(\xi_3)$
r	reflectivity
R	universal gas constant [J/mole-K]
S	dimensionless flame speed, $u(-x_1)/u_{ad}$
T	temperature [K]
u	axial velocity [m/s]
V	diffusion velocity [m/s]

W	molecular weight [kg/mole]
$\bar{W}$	average molecular weight [kg/mole]
x	axial coordinate [m]
x_1-x_3	lengths in Fig. 2.1 [m]
Y	mass fraction

Greek Symbols

δ	flame thickness, unit step function
ϵ	emissivity
η_o	output radiative efficiency, $q^r(L)/q_c$
η_t	total radiative efficiency, $\{q^r(L)-q^r(0)\}/q_c$
θ	dimensionless temperature, $(T-T_i)/T_i$
μ	direction cosine
ξ	dimensionless axial coordinate, x/L
ξ_{f1}	non-dimensional flame location in the heat transfer model $(\xi_1+\xi_2)/2$
ξ_f	non-dimensional flame location, x_f/L
ρ	gas density [kg/m ³]
σ	Stefan Boltzmann constant [W/m ² -K ⁴]
$\sigma_a, \sigma_s, \sigma_e$	absorption, scattering and extinction coefficients [1/m]
τ	optical depth, $\sigma_e L$

ϕ	porosity
Φ	equivalence ratio
ω'	molar generation rate [moles/m ³ -s]
ω	scattering albedo, σ_s/σ_e

Superscripts

+	positive direction
-	negative direction
c	conduction
r	radiation

Subscripts

a	air
ad	adiabatic
b	blackbody
c	combustion
d	diffuse, downstream
e	exit plane
f	flame, fuel
g	gas
i	inlet plane
ig	ignition
j	interface
k	index for species
o	negative infinity

s solid
u upstream

CHAPTER 1

INTRODUCTION

Forous radiant burners (PRBs) are emerging as a promising new gas-burner technology. The operation of PRBs involve stabilizing a premixed flame inside or near a non-combustible porous material. Combustion in the gas phase heats the solid matrix which emits thermal radiation that is directed towards a heat load. Shown in Fig. 1.1 is one such burner in which the porous body is planar. Premixed fuel enters into a distribution chamber and the flame is stabilized within a porous slab. The radiation emanating from the slab is directed towards a heat load.

In small-scale residential and commercial installations such as water heaters, warm air heaters and cooking appliances, demonstrated advantages compared to conventional open-flame burners include (GRI Report, 1984):

- (1) improved system efficiency due to lower convective loss in the flue gas,
- (2) lower burner operating temperatures due to radiation release, thus lower NO_x emissions,
- (3) uniform heat distribution over the entire burner surface, preventing hot spots due to flame impingement and increasing equipment life and
- (4) burner can be shaped to match the heat load geometry to provide more effective heating.

Fig. 1.1 Schematic diagram of a planar porous radiant burner

In addition, existing conventional open-flame burners can easily be retrofitted with a porous segment at the fuel-air nozzle and operated as a porous radiant burner. The extension of the PRB technology to large-scale industrial applications would undoubtedly yield significant gains in energy conservation, productivity and air quality. However, since the required heating rates in most industrial applications are higher, improvements in the thermal performance of PRBs are needed before they can be adopted in widespread industrial usage. To accomplish this, a better understanding of the heat transfer and combustion phenomena in porous media is required. The goal of the present work is to develop a body of knowledge on heat transfer and combustion in PRBs and establish the influence of the different controlling parameters on burner performance.

1.1 Background

As mentioned earlier, PRBs operate by stabilizing a flame in or near a non-combustible porous medium. The energy released in the gas phase heats the solid which emits thermal radiation to a heat load. It is clear that for a burner to operate steadily and continuously, a flame that is fixed to the laboratory frame of reference must be obtained. For such a condition to be achieved in a premixed system, the flame propagation rate or simply the flame speed must be equal and opposite to the speed of the incoming reactants. If the flame speed is higher than the incoming flow speed, the flame

would move in the upstream direction and may eventually flash back. Likewise, if the flame speed is lower than the reactant speed, the flame would be swept away in the downstream direction and blowoff would occur. In the event of either flashback or blowoff, the burner would cease to operate. In principle a steady operating flame may be obtained by appropriately adjusting the flowrate of the reactants to match the flame speed. Thus, for an efficacious operation of PRBs, a detailed knowledge of the factors controlling the flame speeds is required.

1.1.1 Laminar Flame Propagation in Premixed Gaseous Systems

In order to understand the complex problem of flame propagation in porous media, it is informative to first review elements of laminar flame propagation in premixed gaseous systems without the presence of a porous medium. A one-dimensional laminar flame structure is shown in Fig. 1.2. The reactants flow from left to right with velocity u . The gas temperature T increases from an initial value T_i to a final flame temperature T_f . As the gas flows through the preheat zone, it is heated conductively by the flame until its ignition point T_{ig} is reached. Then the gas is heated further to T_f in the reaction zone due to exothermic reactions. The mass fractions of the reactants Y_r decrease continuously as the reaction progresses. Similarly, mass fractions of the products Y_p increase.

Fig. 1.2 Laminar flame structure in premixed gaseous systems

Many theories on the prediction of laminar flame speeds have been put forward (Glassman, 1977). A simple thermal model that agrees favorably with experimental trends is found to be insightful in showing the functional dependency of the flame speed. By applying an energy balance on the preheat and reaction zones, for constant properties and no downstream heat loss, it can be shown (Glassman, 1977) that:

$$u_l = \left(\frac{2k\omega'H}{\rho c(T_f - T_i)} \right)^{1/2} \quad (1.1)$$

where u_l is the laminar adiabatic flame speed, k is the thermal conductivity, ρ is the density, c is the specific heat, H is the enthalpy of reaction and ω' is the reaction rate. The results indicate that the flame speed is directly proportional to the square root of the thermal diffusivity and the reaction rate. Such a dependency is expected qualitatively. Greater thermal diffusivity enhances diffusion of energy upstream of the flame, and thus increases the flame speed. An increase in the reaction rate results in a faster consumption of the reactants and hence a larger flame speed. The temperature dependency is reflected in the reaction rate given by $\omega' = A \exp(-E/RT)$ where A is the preexponential factor, E is the activation energy and R is the universal gas constant. The pressure dependency is reflected through ρ . It is clear that the thermophysical

properties are important factors that control the flame speed.

1.1.2 Laminar Flame Propagation in Porous Media

Due to the fact that the solids have different properties than gases, the flame speed is altered by the presence of a solid porous matrix. Indeed, De Soete (1967) experimentally observed different flame velocities in packed sand columns and purely gaseous systems. Moving flames were studied for premixed $\text{CH}_4/\text{O}_2/\text{N}_2$ mixtures. Solid and gas phase temperatures were found to be quite different due to gas phase heat release and solid phase conduction loss.

Korzhavin et al. (1982) indicated the possibility of obtaining stationary flames in packed beds. The flame speeds were observed to increase inside the porous bed compared to those outside the bed. Takeno and Sato (1979) analytically proposed an excess enthalpy burner which used an energy feedback mechanism to increase the flame speed, thereby extending the lean combustion limit. The energy feedback was achieved by externally recirculating part of the flue gas to increase the incoming reactant temperatures as well as by enhanced conduction upstream of the flame through the porous slab in which combustion was sustained.

Several analytical studies on the expected range of stable combustion in porous media with applications to excess enthalpy burners have also appeared (Buckmaster and Takeno, 1981; Takeno et al., 1981; Takeno and Hase, 1983; Takeno and

Murayama, 1986), but radiation and the temperature gradients within the porous medium were neglected. The temperature of the porous material was an eigenvalue to be solved for. Experimental studies on excess enthalpy burners were conducted subsequently (Kotani and Takeno, 1982; Kotani et al., 1984). The burners consisted of multiple porous slabs in which combustion was sustained. It was demonstrated that the lean limit of combustion could be extended considerably. Though all the above mentioned studies provide some insight into the problem of flame propagation in porous media, they do not address the critical issues where radiative heat transfer is important.

Arpaci and Tabaczynski (1982) reported the effect of radiation on laminar flame speed in a semi-infinite medium. Absorption, emission and scattering of radiation were incorporated in the analysis. Using a rather simplistic and qualitative approach it was shown that radiation increases the flame speed. The need to analyse radiation affected flame propagation using an exact model of radiative transfer was stressed. Joulin (1987) employed asymptotic techniques in a semi-infinite medium to show the potentially important effects of radiation on flame propagation. Radiative transfer was assumed to give a small contribution to the energy equation and that it could be computed from the unperturbed radiation-free temperature profile. A one step combustion reaction was employed by Yoshizawa et al. (1988)

to reveal the structure of radiation-controlled flames in inert porous media. Absorption and emission of radiation was accounted for but the important effect of radiation scattering was neglected in their analysis. The flame was assumed to be situated at or near the middle of the porous medium. Their results confirmed the significance of radiative heat transfer in promoting energy feedback towards the upstream side of the reaction zone yielding higher than adiabatic flame speeds. Chen et al. (1987) investigated the effect of radiation on the structure of methane-air premixed laminar flames in porous media using detailed multi-step chemistry to model combustion. However, the effect of radiation scattering and non-local thermal equilibrium between the gas and solid phases was neglected in their model, and the flame was assumed to be located at the center of the porous segment. They reported an increase in the flame speeds when compared to the adiabatic gaseous premixed flame speed. They attributed the increase to preheating of the incoming gases by radiation. In a contrasting effect, excessive radiative heat loss from the flame zone has been shown to lead to flame extinction (Sohrab and Law, 1984; Arpacı and Tabaczynski, 1984; Gerstein and Stine, 1973).

1.1.3 Heat Transfer in Porous Radiant Burners

Since the flame speed is affected by the temperature of the porous medium, it is expected that heat transfer in PRBs will have a significant effect on the flame speed and hence

on the radiant output. It is thus important to gain insight into the heat transfer processes inside PRBs.

Not much literature exists on the topic of porous radiant burners, since this is still an evolving technology. Some related heat transfer literature on the usage of porous media in converting fluid enthalpy to thermal radiation exists. In one such investigation, Echigo (1982) studied the problem of recovering energy from hot exhaust gases by placing a porous segment in exhaust ducts. His analysis revealed that a substantial amount of the exhaust gas energy might be recovered by converting it to radiative energy traveling upstream from the porous segment. The solid was assumed not to scatter radiant energy. Wang and Tien (1984) extended Echigo's analysis by including the effect of radiation scattering in the analysis. They demonstrated the importance radiation scattering had on the effectiveness of recovering the energy that would otherwise be lost. By considering a heat generation zone inside a porous medium, Echigo et al. (1986) extended their earlier work (Echigo et al., 1982) to investigate the use of porous media for combustion augmentation. It was found that energy recirculation by radiation preheated the combustion mixture, resulting in significant combustion augmentation. Tong et al. (1987) studied the effects of the position of the heat generation zone location, the optical thickness and radiative scattering on the radiant efflux from a porous slab. Their

heat transfer model was a purely radiative model, but it accounted for anisotropic scattering of radiation. It was demonstrated that the optical depth, the scattering of radiation and the position of the heat generation zone had a substantial effect on the radiant efflux.

1.2 Scope of this Work

Though some research has been done in the related areas of flame propagation and heat transfer in porous media, there has been no prior study on PRBs. Also fundamental knowledge on the effects of the various heat transfer modes on the flame propagation in porous media has not been established. Moreover, the effects of multi-mode heat transfer and combustion on the stability of premixed flames in finite porous segments, such as that encountered in PRBs have not been studied.

The aim of this investigation is to generate a body of knowledge on combustion and heat transfer in inert porous media in order to obtain a basic understanding of the operation of PRBs. Such knowledge is important for devising methods to improve their performance. This dissertation is organized as follows:

A numerical analysis of the heat transfer characteristics of PRBs is performed in Chapter 2. The combustion phenomenon is modeled using a heat generation zone. The results identify the role of the heat transfer parameters such as the properties of the porous

material and flow conditions on radiant output. In Chapter 3, a numerical model is developed that incorporates combustion in the heat transfer model of Chapter 2. The behavior of flame speed, radiant output and flame stability in PRBs are studied. Chapter 4 describes the experimental program conducted to validate the theoretical predictions in the prior chapters. Finally, conclusions and recommendations for future studies are given in Chapter 5.

CHAPTER 2

HEAT TRANSFER CHARACTERISTICS OF POROUS RADIANT BURNERS

A one-dimensional numerical model is developed to study the heat transfer characteristics of PRBs. The model incorporates multimode heat transfer, namely, conduction, convection and radiation. Combustion is modeled as a heat generation source. Radiant output is determined as a function of the porous material properties, environmental conditions and flow rates. Results from such a model will help to gain insight into a rather complex heat transfer problem. These results will also provide the initial guess for numerically solving the more complicated problem with combustion included.

The analysis is based on one-dimensional fluid flow through a porous segment in which combustion takes place. One dimensional flow may be justified in many PRBs configurations due to the thinness of the porous layer that results in negligible pressure drop. Non-local thermal equilibrium between the gas and solid phases are accounted for. Conduction through the solid phase and scattering of radiation are also incorporated in the model. Combustion is modeled as a constant heat generation zone inside the porous segment. The inlet boundary condition for the gas is specified far upstream from the porous segment so that the

effect of preheating the combustion mixture before it enters the porous segment is allowed for.

2.1 Mathematical Formulation

The situation considered is schematically shown in Fig. 2.1. A gaseous fuel-air mixture enters an adiabatic duct that may be either cylindrical or rectangular. The flame is situated in the porous region, whose thickness is much smaller than the duct hydraulic diameter. All thermophysical properties are constant. Slug flow is assumed, and it is steady and laminar. The solid is assumed to be gray and to emit, absorb and scatter radiant energy isotropically. The gas is treated as transparent. This is a reasonable assumption since the solid emission is expected to be substantially higher than the gas emission (see Appendix A). Since the solid and gas are not in local thermal equilibrium, separate energy equations are needed to describe energy transfer in these two phases. The properties and variables represent quantities averaged over the particular phase in question.

Energy equation for the gas

$$\rho c_g u \phi \frac{dT_g}{dx} + (1-\phi) h a (T_g - T_s) = Q \phi \delta(x) + k_g \phi \frac{d^2 T_g}{dx^2} \quad (2.1)$$

Fig. 2.1 Schematic diagram of the porous radiant burner

where ρ is the gas density, u is the velocity, c_g is the gas specific heat capacity, T_g is the gas temperature, ϕ is the porosity defined such that $\phi = 1$ for $x < 0$ and $0 \leq \phi \leq 1$ for $0 \leq x \leq x_3$, h is the heat transfer coefficient for the convective interaction between the solid and the gas in the porous segment, a is surface area of the solid per unit volume of the solid, T_s is the solid temperature, Q is the heat generation rate per unit volume of the gas and $\delta(x)$ is a delta function defined as unity for $x_1 \leq x \leq x_2$ and zero elsewhere.

A known temperature at the inlet and an adiabatic condition at the outlet constitute the gas phase boundary conditions. Adiabatic conditions at the outflow position are justified if local one-way behavior exists (Patankar, 1980), which is the case in the present work. Thus, we have

$$T_g = T_i \text{ at } x = -x_1 \quad (2.2a)$$

$$\frac{dT_g}{dx} = 0 \text{ at } x = x_3 \quad (2.2b)$$

Energy equation for the solid phase in the porous segment

$$(1-\phi)k_s \frac{d^2 T_s}{dx^2} + (1-\phi)ha(T_g - T_s) = \frac{dq^r}{dx} \quad (2.3)$$

where k_s is the solid thermal conductivity and q^r is the net radiative heat flux due to the solid. In this analysis three different types of boundary conditions are used in order to investigate the effect of the boundary conditions. The first type equates the convective heat flux between the gas and solid phases to the solid conductive heat flux. It is

$$-k_s \frac{dT_s}{dx} = h(T_g - T_s) \text{ at } x = 0 \quad (2.4a)$$

$$k_s \frac{dT_s}{dx} = h(T_g - T_s) \text{ at } x = x_3 \quad (2.4b)$$

The second type is the adiabatic condition written as

$$\frac{dT_s}{dx} = 0 \text{ at } x = 0 \quad (2.5a)$$

$$\frac{dT_s}{dx} = 0 \text{ at } x = x_3 \quad (2.5b)$$

The third and last type of boundary condition considered matches the gas and solid conductive heat fluxes

$$k_g \left. \frac{dT_g}{dx} \right|_{x=0} - k_s \left. \frac{dT_s}{dx} \right|_{x=0} = 0 \quad \text{at } x = 0 \quad (2.6a)$$

$$k_s \left. \frac{dT_s}{dx} \right|_{x=x_3} - k_g \left. \frac{dT_g}{dx} \right|_{x=x_3} = 0 \quad \text{at } x = x_3 \quad (2.6b)$$

Equations (2.1-2.6) are non-dimensionalized as

$$P_1 \frac{d\theta_g}{d\xi} + (1-\phi) P_2 (\theta_g - \theta_s) = \phi \delta(\xi) + P_3 \frac{d^2 \theta_g}{d\xi^2} \quad (2.7)$$

$$\theta_g = 0 \quad \text{at } \xi = -\xi_i \quad (2.8a)$$

$$\frac{d\theta_g}{d\xi} = 0 \quad \text{at } \xi = \xi_3 \quad (2.8b)$$

$$(1-\phi) P_4 \frac{d^2 \phi_s}{d\xi^2} + (1-\phi) P_2 (\theta_g - \theta_s) = \frac{dQ^*}{d\xi} \quad (2.9)$$

$$-\frac{d\theta_s}{d\xi} = P_5 (\theta_g - \theta_s) \quad \text{at } \xi = 0 \quad (2.10a)$$

$$\frac{d\theta_s}{d\xi} = P_5 (\theta_g - \theta_s) \text{ at } \xi = \xi_3 \quad (2.10b)$$

$$\frac{d\theta_s}{d\xi} = 0 \text{ at } \xi = 0 \quad (2.11a)$$

$$\frac{d\theta_s}{d\xi} = 0 \text{ at } \xi = \xi_3 \quad (2.11b)$$

$$P_3 \left. \frac{d\theta_g}{d\xi} \right|_{\xi=0} - P_4 \left. \frac{d\theta_s}{d\xi} \right|_{\xi=0} = 0 \text{ at } \xi = 0 \quad (2.12a)$$

$$P_4 \left. \frac{d\theta_s}{d\xi} \right|_{\xi=\xi_3} - P_3 \left. \frac{d\theta_g}{d\xi} \right|_{\xi=\xi_3} = 0 \text{ at } \xi = \xi_3 \quad (2.12b)$$

where $\xi = x/x_3$, $\theta_g = (T_g - T_i)/T_i$, $\theta_s = (T_s - T_i)/T_i$, $Q^* = q^r/Qx_3$ and the definitions of P_1 through P_5 and their physical significance are given in Table 2.1.

Radiative heat transfer in the porous segment is modeled using the radiative transfer equation (RTE). In one-dimensional form the RTE can be written as (Ozisik, 1973)

Table 2.1 Summary of the non-dimensional parameters

Parameter	Physical Significance	Typical Value*	Range Studied
$P_1 = \rho u c_g T_i / Q x_3$	flow enthalpy	0.014	0.01-0.05
$P_2 = h a T_i / Q$	convective energy coupling between gas and solid	4 (100 μm fiber) 400 (10 μm fiber)	1-500
$P_3 = k_g T_i / Q x_3^2$	gas conduction	2.5×10^{-4}	10^{-5} - 10^{-3}
$P_4 = k_s T_i / Q x_3^2$	solid conduction	0.02 (non-metals) 0.20 (metals)	0.002-0.2
$P_5 = h x_3 / k_s$	Blot number	5	0.05-500

* Based on $\rho = 0.19 \text{ kg/m}^3$, $u = 2.8 \text{ m/s}$, $c_g = 1.28 \text{ kJ/kg K}$, $T_i = 298 \text{ K}$, $Q = 1.5 \times 10^9 \text{ W/m}^3$, $x_3 = 0.01 \text{ m}$, $h a = 2 \times 10^9$ and $2 \times 10^7 \text{ W/m}^3 \text{ K}$ for 10 and 100 μm fibers, respectively, $k_g = 0.125 \text{ W/m K}$, $k_s = 10$ and 100 W/m K for non-metals and metals, respectively.

$$\mu \frac{\partial I(x, \mu)}{\partial x} + (\sigma_a + \sigma_s) I(x, \mu) = \sigma_a I_b(T_s) + \frac{\sigma_s}{2} \int_{-1}^1 I(x, \mu') d\mu' \quad (2.13)$$

where I is the radiant intensity, μ is the cosine of the angle between the direction of the intensity and the x direction, σ_a and σ_s are the absorption and scattering coefficients, respectively, and I_b is the blackbody radiant intensity.

The upstream boundary condition for the intensity is written by assuming the upstream condition can be characterized by a gray and diffusely emitting and reflecting surface at T_1 . Therefore,

$$I^+(0) = \varepsilon I_b(T_1) + 2r_d \int_0^1 I^-(0, -\mu') \mu' d\mu' \quad (2.14)$$

where I^+ and I^- are the radiant intensities in the positive and negative x directions, respectively, ε and r_d are the emissivity and diffuse reflectivity of the upstream boundary, respectively. The burner is assumed to be seeing a black environment at T_e . Thus, the boundary condition for the intensity at x_3 can be expressed as

$$I^-(x_3) = I_b(T_e) \quad (2.15)$$

Equations (2.13 - 2.15) are non-dimensionalized to yield the following equations, respectively

$$\frac{\mu}{\tau} \frac{\partial i(\xi, \mu)}{\partial \xi} + i(\xi, \mu) = (1-\omega) i_b(T_s) + \frac{\omega}{2} \int_{-1}^1 i(\xi, \mu') d\mu' \quad (2.16)$$

$$i^+(0) = \frac{\epsilon G_i(T_i)}{\pi} + 2(1-\epsilon) \int_0^1 i(0, -\mu') \mu' d\mu' \quad (2.17a)$$

$$i^-(\xi_3) = \frac{G_e(T_e)}{\pi} \quad (2.17b)$$

where $i = I/Qx_3$, $i_b = I_b/Qx_3$, $G_i = \sigma T_i^4/Qx_3$, $G_e = \sigma T_e^4/Qx_3$, $\tau = (\sigma_a + \sigma_s)x_3$ is the optical thickness of the porous segment and $\omega = \sigma_s/(\sigma_a + \sigma_s)$ is the scattering albedo. Note that Kirchhoff's law has been invoked in arriving at eq. (2.17a).

The net radiative heat flux $q^{\pm}$ appearing in eq. (2.3) is obtained by integrating the first moment of the radiant intensity over all directions. That is

$$q^{\pm} = 2\pi \int_{-1}^1 I(\xi, \mu') \mu' d\mu' \quad (2.18)$$

In terms of the dimensionless radiant intensity i^* is

$$Q^* = 2\pi \int_{-1}^1 i(\xi, \mu') \mu' d\mu' \quad (2.19)$$

where $Q^* = q^r / Qx_3$.

2.2 Method of Solution

Equation (2.16) is transformed into a set of four first order ordinary differential equations in terms of the moments of the radiant intensity using the P-3 spherical harmonics approximation (Swathi et al., 1987; Tong et al., 1987). Equations (2.17a and 2.17b) are replaced by the Marshak's boundary conditions which require the intensity moments to be conserved. The spherical harmonics method and the Marshak's boundary conditions for solving the equation of transfer have been well documented elsewhere (Davison, 1957; Ozisik, 1973; Menguc and Viskanta, 1985; Swathi et al., 1987; Tong et al., 1987). The readers are referred to these references for the mathematical details. As indicated by eq. (2.18), the first moment of the radiant intensity is the net radiative heat flux.

The four equations governing the moments of the radiant intensity and eqs. (2.7 and 2.9) are solved according to the following procedure:

1. Guess $d\theta_g/d\xi$ at $\xi = 0$;

2. Solve eq. (2.7) analytically for the region $-\xi_1 \leq \xi \leq 0$ using eq. (2.8a) and the boundary condition from step 1;
3. Determine θ_g at $\xi = 0$ using the solution from step 2;
4. Numerically solve eqs. (2.7 and 2.9) and the four equations resulting from the P-3 spherical harmonics approximation for the region $0 \leq \xi \leq \xi_3$ using eqs. (2.8b, 2.10a and 2.10b), the boundary condition from step 3 and the Marshak's boundary conditions;
5. Determine $d\theta_g/d\xi$ at $\xi = 0$ using the solution from step 4; and
6. Repeat steps 2 to 5 until convergence is achieved.

The subroutine DVCPR from the IMSL library was used to perform the computation in step 4. Convergence for step 6 was assumed when all the following conditions were met:

1. Fractional change in $d\theta_g/d\xi$ at $\xi = 0$ in consecutive iterations was less than 10^{-4} ;
2. Fractional change in θ_g at $\xi = 0$ in consecutive iterations was less than 10^{-4} ; and
3. An overall energy balance of less than 0.1 percent was achieved.

A relative error tolerance of 10^{-4} was specified for the iterations internal to subroutine DVCPR.

2.3 Computations and Test Cases

Computations were performed for a wide range of conditions, including those that were typical for actual PRB

operations. The following values were used as a guide to set the parameters P_1 through P_5 : $u = 2.8$ m/s, $T_i = 298$ K, $Q = 1.5 \times 10^9$ W/m³, $\rho = 0.19$ kg/m³, $c_g = 1.28$ kJ/kg K, $k_g = 0.125$ W/m K, $k_s = 10$ and 100 W/m K, $h_a = 2 \times 10^9$ and 2×10^7 W/m³ K. The values of P_1 through P_5 resulting from these properties are summarized in Table 2.1. The heat generation rate (Q) used was based upon the lower heating value for stoichiometric methane-air combustion. Gas properties were for air at 1800 K. The lower and higher values for k_s were intended to be representative of non-metallic and metallic fibers, respectively. Values of h_a were estimated from the experimental data for h for cylinders in cross-flow (Incropera and DeWitt, 1985) and a for infinitely long fibers that were either 10 or 100 μ m in diameter. In addition, the porous segment was assumed to be composed of monodispersed cylindrical fibers with a thickness (x_3) of 0.01 m and a porosity (ϕ) of 0.95. The flame zone ($x_2 - x_1$) was assumed to be one millimeter thick. This was found to be a reasonable estimate of the flame thickness when the calculations were performed by including combustion in Chapter 3. The length x_1 was considered to be large (1 m) so that the gas temperature at $x = 0$ was not affected by preheating.

All the calculations were performed on an IBM 3090 computer. To determine the effect of the relative error tolerance in DVCPR, test calculations were conducted for $P_1 = 0.01$, $P_2 = 100$, $P_3 = 2.5 \times 10^{-4}$, $P_4 = 0.02$, $P_5 = 5$, $\omega = 0.5$, r_d

$= 0$, $\xi_{f1} = 0.5$, $\tau = 2$ and $G_i = G_e = 2.98 \times 10^{-5}$. It was found that for a relative error tolerance of 1×10^{-4} , the calculations took 240 cpu seconds and required 350 grid points. An overall energy balance of 0.022 percent for the porous layer was obtained. When the relative error tolerance was 1×10^{-5} , the computations took 515 cpu seconds and 520 grid points, with an energy balance of 0.014 percent. The corresponding change in radiative output was less than 0.1 percent. For all subsequent calculations, 1×10^{-4} was chosen as the relative error tolerance so as to achieve accurate results without incurring prohibitively high computational times.

2.4 Results and Discussion

From the computation performed, it was observed that the three different boundary conditions specified by eqs. (2.4, 2.5 and 2.6) for the solid phase produced almost identical radiant output. Hence, only the results obtained from using the convection boundary condition are presented. Also, only results for $P_3 = 2.5 \times 10^{-4}$ and $P_5 = 5$ are presented since a variation in these parameters over the ranges indicated in Table 2.1 resulted in negligible changes in the radiant output. Calculations were performed by making variations in the following baseline values of the parameters: $P_1 = 0.01$, $P_2 = 500$, $P_3 = 2.5 \times 10^{-4}$, $P_4 = 0.02$, $P_5 = 5$, $\omega = 0.5$, $r_d = 0$, $\xi_{f1} = 0.5$, $\tau = 10$, $\phi = 0.95$ and $G_i = G_e = 2.98 \times 10^{-5}$.

Effect of τ

Figure 2.2 shows the effect of the optical thickness (τ) on the temperature distribution inside the porous medium. The flame zone position represented by ξ_{f1} [$= (\xi_1 + \xi_2)/2$] is 0.5. That is, the flame is situated in the middle of the porous segment. Because P_2 is large, meaning there is strong convective heat transfer between the gas and solid, the temperatures of the two phases are almost identical and exhibit no observable differences. The curve for $\tau = 0$ corresponds to the situation when there is no heat transfer by radiation. In comparison to the other curves for $\tau > 0$, it is seen that the peak temperature is highest when there is no radiation. This indicates that radiation serves as an additional heat removal mechanism from the flame zone. The temperature at $\xi = 0$ is higher for larger τ due to preheating by radiation. However, the peak temperature and the post-flame ($\xi > \xi_2$) temperature at first decreases with increasing τ ; but when $\tau > 2$ it increases as τ is increased further. This is because the post-flame region is optically thick enough when τ reaches 2 and it begins to prevent radiative heat loss from the flame zone.

Figure 2.3 shows the effect of τ on the non-dimensional radiative output ($Q_0^F = Q^*$ at $\xi = \xi_3$) for different locations of the flame zone. The quantity $Q_0^F x_3 / (x_2 - x_1)$ represents the ratio of the radiative output to the heat generated. When ξ_{f1} is closer to 0, the post-flame region acts like a shield

Fig. 2.2 Effect of the optical thickness on the temperature distribution

Fig. 2.3 Effect of the optical thickness on the radiative output

and results in lower radiant output. On the other hand, if ξ_{f1} is close to ξ_3 , there is only a small region for the gas to transfer heat to the solid before the gas leaves the porous region. Thus, only a small region of the solid is at a high temperature and the radiant output tends to be low. As a result of these counteracting phenomena, there exists an optimal position of ξ_{f1} for which the radiant output is maximized. When ξ_{f1} is not close to 0 the radiant output increases substantially with an increase in τ . Because of the shielding effect the converse is true when ξ_{f1} is close to 0.

Effect of P_1

The results in Fig. 2.4 show that the radiant output can be increased by decreasing P_1 . Smaller P_1 can be obtained by slowing the gas flow. It takes longer for the gas to travel through the porous segment and more energy can be transferred from the gas to the solid, resulting in higher radiant output. Based on the results, one would attempt to continue to decrease P_1 to enhance the radiant output. In reality there is a lower limit for P_1 . When the gas flow is too low flash-back will occur and the PRB will cease to function. However, such a limit cannot be established through a purely heat transfer analysis. The optimal ξ_{f1} for which the radiant output is maximized depends on the gas flow. In general, the optimal position is more towards the downstream direction when the gas flow is lower. This trend is consistent with the findings of Tong et al. (1987). They

Fig. 2.4 Effect of P_1 and P_2 on the radiative output

illustrated that when there was no convection, the heat generation zone should be positioned right at the front surface to maximize radiant efflux from that surface

Effect of P_2

Depicted in Fig. 2.5 are the temperature distributions for different values of P_2 . For large fibers, P_2 is small. Heat transfer between the gas phase and the solid phase is poor, resulting in gas temperatures significantly higher than the solid temperatures. This is undesirable from the standpoint of maximizing the radiant output and minimizing NO_x production. It is seen that when $P_2 \geq 100$ the two phases are quite close to local thermal equilibrium.

Figure 2.6 shows the distributions of the convective, conductive and radiative heat fluxes. Gas conduction is almost negligible compared to the other modes of heat transfer. This supports the use of an adiabatic outflow boundary condition as discussed earlier. The results also reveal that the solid conductive flux is comparable in magnitude to the radiative flux. Since the value of P_4 (0.02) is representative of non-metallic fibers, the results indicate that solid conduction should not be neglected in such an analysis. The convective flux is most predominant. This indicates that there is high potential for significant increase of the radiant output by converting more of the convective energy to radiative energy. This can be achieved by increasing P_2 . For instance, the ratio of the radiative

Fig. 2.5 Temperature distributions for different P_2

Fig. 2.6 Comparison of the convective, radiative and conductive energy fluxes for different P_2

heat flux to the convective heat flux at $\xi = 1$ is seen to increase with an increase in P_2 . However, the increase appears to be minimal once P_2 has reached 100. Therefore, further increase in the radiant output must be achieved by adjusting other parameters.

Combined effect of P_4 and preheating the gas before it enters the porous segment

The effect of P_4 or solid conduction is shown in Fig. 2.7. Smaller P_4 results in higher radiant output because less energy is removed from the flame zone via solid conduction. The dashed curves correspond to the case when $x_1 = 0$. That is, the effect of preheating the incoming gas before it enters the porous segment is not taken into account. The radiant output is grossly underpredicted in many instances. The underprediction is greater for higher solid conduction and when the flame zone is closer to $\xi = 0$.

Effect of ω

Shown in Fig. 2.8 is the effect of radiation scattering. A lower albedo results in a higher radiant output due to stronger emission by the solid. Thus a strong emitting-absorbing medium is preferred for obtaining high radiant output. The results indicate that scattering of radiation is an important effect.

Effect of r_d

Reflection of radiant energy from the inlet environment has negligible effects if the flame is located at the

Fig. 2.7 Effect of P_4 on the radiative output

Fig. 2.8 Effect of the scattering albedo on the radiative output

position that maximizes radiant output, as seen in Fig. 2.9. However, if the flame zone is at a position far upstream from the optimal position, reflection affects the radiant output significantly. A higher reflectivity increases the radiant output from the burner since less radiative energy is lost in the upstream direction.

2.5 Conclusions

The heat transfer characteristics of PRB have been studied using a multi-mode heat transfer model. The model accounted for non-local thermal equilibrium between the gas and solid phases of the porous layer, and radiation scattering by the solid. Preheating of the gas before it entered the porous layer was also allowed for. The spherical harmonics method was used to transform the equation of transfer to a set of ordinary differential equations, which were numerically solved simultaneously with the energy equations.

It was found that gas conduction had a negligible effect on the radiant output. On the other hand, solid conduction and radiation scattering both exhibited considerable influence. Low solid conduction and low radiation scattering resulted in higher radiant output. The effect of preheating the gas before it entered the porous segment was significant for solids that had a high thermal conductivity. To maximize the output: P_1 (flow enthalpy) should be as small as possible without causing flashback to occur; P_2 (convection between

Fig. 2.9 Radiative output for different r_d

the gas and solid) should be at least 100; τ should be large enough for the porous segment to be optically thick; and as a result of the convective effect, the flame zone should be at some distance behind the outlet surface of the porous layer.

CHAPTER 3

COMBUSTION AND HEAT TRANSFER IN POROUS RADIANT BURNERS

The heat transfer characteristics of porous radiant burners (PRBs) were determined in Chapter 2. The combustion was modeled by fixing a heat generation zone inside the porous medium. The analysis, though simple, provided considerable insight into the heat transfer processes inside PRBs. To incorporate the dependence of flame speed on the flame location, the combustion and heat transfer must be modeled simultaneously. Since the problem is more difficult to solve with combustion effects included, results from the heat transfer model will be used as starting guesses.

An efficacious use of combustion devices such as the PRBs has been limited due to the lack of understanding of the fundamental mechanisms of flame propagation in porous media. A key to determining the stable range of combustion in porous materials is to evaluate the burning velocity and its link to the heat transfer in the media. In this chapter a theoretical model is developed for quantifying the effect of the relevant parameters on flame propagation in inert porous materials. A one-dimensional, laminar flame model is employed, wherein the flame thickness is much less than the length of the porous medium. Combustion is modeled using a single-step, irreversible reaction. The solid and gas phases are assumed not to be in local thermal equilibrium. The solid is assumed to emit, absorb, and scatter radiant energy.

The computational domain is extended beyond either side of the porous region to accurately model flames close to the edges of the porous region. The results will clarify the coupling between flame propagation and heat exchange in porous materials. The influence of the combustion and heat transfer processes on PRB performance will be established.

3.1 Mathematical Analysis

The analysis is based on a model somewhat similar to the heat transfer model developed in Chapter 2, but in addition combustion effects are included. The schematic diagram of the problem under consideration is shown in Fig. 3.1. A premixed methane-air gas mixture enters an adiabatic duct containing an inert porous layer of length L . The flame may be located either inside or outside the porous layer and the flame thickness is much less than L . The flow is assumed to be one-dimensional, steady, and laminar. The viscous effects are neglected so that the momentum equation can be integrated and replaced by the condition that the thermodynamic pressure is a constant. The solid matrix is assumed to be gray and to emit, absorb, and scatter radiant energy. Gaseous radiation is neglected compared to solid radiation. Non-local thermal equilibrium between the gas and the solid phase is accounted for by considering separate energy equations for the two phases. All the properties of the solid are assumed to be homogeneous and constant.

Fig. 3.1 Schematic diagram of the porous radiant burner
(combustion model)

The methane-air combustion is modeled using the one-step global reaction (Reitz and Bracco, 1982) with an equivalence ratio of one or less so that the only species involved are CH_4 , O_2 , N_2 , CO_2 , and H_2O . The reaction rate constant is given by $k_c = 1.8 \times 10^8 \exp(-125600/RT_g) [\text{CH}_4] \text{ moles/m}^3\text{-s}$. The thermodynamic and transport properties of the gases are evaluated using the CHEMKIN (Kee et al., 1980) and TRANFIT (Kee et al., 1986) codes, respectively, from Sandia National Laboratory. Thus the mixture density and thermal conductivity variations with temperature are accurately taken into account

The governing equations are written by incorporating convection, conduction, radiation, and combustion processes.

Continuity equation

$$\frac{d}{dx}(\rho u \phi) = 0 \quad (3.1)$$

where ρ is the variable gas density, u is the velocity and ϕ is the porosity so that $\phi < 1$ for $0 < x < L$ and $\phi = 1$ elsewhere. Note that the continuity equation was not required in the heat transfer analysis because the velocity and density were held constant in that analysis.

Species conservation equation

$$\rho u \phi \frac{dY_k}{dx} + \frac{d}{dx}(\rho \phi Y_k V_k) = \phi \omega'_k W_k \quad (k = 1, K) \quad (3.2)$$

where Y_k is the species mole fraction, ω'_k is the species production rate, W_k is the species molecular weight and the diffusion velocity V_k is the sum of the diffusion velocities due to mole fraction gradients and the thermal gradients (Kee et al., 1986). The index k refers to the k^{th} species. The boundary conditions constitute specifying inlet mole fractions and vanishing gradients at the exit:

$$Y_k = Y_{k,i}, \quad \text{at } x = -x_i \quad (3.3a)$$

$$\frac{dY_k}{dx} = 0 \quad \text{at } x = x_e \quad (3.3b)$$

Energy equation for the gas

$$\rho u \phi c_g \frac{dT_g}{dx} + (1-\phi) h a (T_g - T_s) + \phi \sum_{k=1}^K \rho Y_k V_k c_k \frac{dT_g}{dx}$$

$$= -\phi \sum_{k=1}^K \omega'_k h_k W_k + \frac{d}{dx} \left(\phi k_g \frac{dT_g}{dx} \right) \quad (3.4)$$

where c_g is the gas specific heat, T_g is the gas phase temperature, h is the heat transfer coefficient between the solid and gas phases, a is the surface area of the solid per unit volume of the solid, T_s is the solid phase temperature, c_k is the species specific heat, h_k is the species specific enthalpy and k_g is the gas phase conductivity. The boundary conditions for the gas energy equation are similar to those employed in the heat transfer model (see section 2.1):

$$T_g = T_i \quad \text{at } x = -x_i \quad (3.5a)$$

$$\frac{dT_g}{dx} = 0 \quad \text{at } x = x_e \quad (3.5b)$$

Equation of state

$$\rho = \frac{Wp}{RT_g} \quad (3.6)$$

where W is the average molecular weight of the gas phase, p is the pressure and R is the universal gas constant.

Energy conservation for the solid phase in the porous segment

$$-\frac{d}{dx}[(1-\phi)k_s \frac{dT_s}{dx}] + (1-\phi)ha(T_g - T_s) = \frac{dq^r}{dx} \quad (3.7)$$

where k_s is the solid phase thermal conductivity and q^r is the radiative flux. Convective boundary conditions are employed for the solid phase for the reasons mentioned in Chapter 2:

$$-k_s \frac{dT_s}{dx} = h(T_g - T_s) \text{ at } x = 0 \quad (3.8a)$$

$$k_s \frac{dT_s}{dx} = h(T_g - T_s) \text{ at } x = L \quad (3.8b)$$

Radiative Transfer Equation (RTE)

$$\begin{aligned} \mu \frac{\partial i(x, \mu)}{\partial x} + (\sigma_a + \sigma_s) i(x, \mu) &= \sigma_a i_b(T_s) \\ &+ \frac{\sigma_s}{2} \int_{-1}^1 i(x, \mu') d\mu' \end{aligned} \quad (3.9)$$

where μ is the direction cosine, i is the radiative intensity, σ_a is the absorption coefficient, σ_s is the scattering coefficient, i_b is the blackbody intensity at temperature T_s and μ' is the dummy variable of integration.

The upstream boundary condition for the radiant intensity is written assuming that the upstream environment can be characterized by a gray, diffusely emitting and reflecting surface with an emissivity ϵ and a reflectivity r_d . On the downstream side, the burner is assumed to be seeing a black environment at $T_e = 298$ K. Thus, we have

$$i^+(0) = \epsilon i_b(T_i) + 2r_d \int_0^1 i^-(0, -\mu') \mu' d\mu' \text{ at } x = 0 \quad (3.10a)$$

$$i^-(L) = i(T_b) \text{ at } x_e = L \quad (3.10b)$$

where the + and - superscripts refer to the positive and negative x directions. The radiative heat flux appearing in the solid energy equation is given by

$$q^r = 2\pi \int_{-1}^1 i(x, \mu') \mu' d\mu' \quad (3.11)$$

Since combustion effects are incorporated in this model, it is not possible to fix the flow rate and the heat generation rate independently as was done in the heat

transfer model. The heat generation will in fact depend on the flow rate and thus a non-dimensionalization scheme different from that used in the heat transfer model is followed. The governing equations are non-dimensionalized by letting $\xi = x/L$, $\theta = (T-T_1)/T_1$ and $I^* = i/\sigma T_1^4$. Upon non-dimensionalizing, the parameters relevant to this study are: $H_1 = haL/\sigma T_1^3$, $H_2 = k_s/\sigma T_1^3 L$, $H_3 = hL/k_s$, $\omega = \sigma_s/\sigma_e$, and $\tau = \sigma_e L$ where $\sigma_e = \sigma_a + \sigma_s$. H_1 , H_2 , H_3 , ω and τ are the non-dimensional heat transfer coupling between the solid and gas, non-dimensional solid thermal conductivity, Biot number, scattering albedo and the optical depth respectively.

3.2 Solution Procedure

The radiative transfer equation (RTE) was transformed into a set of ordinary differential equations using the spherical harmonics approximation, and the Marshak's formulation was used to model the boundary conditions (see section 2.2). An efficient numerical scheme was devised to solve the governing equations (3.1-3.11) by modifying PREMIX (Kee et al., 1985). The PREMIX code has been applied successfully to solve the governing equations pertaining to flame propagation in premixed gaseous systems. The code employs finite-difference techniques with adaptive gridding and solves the set of nonlinear governing equations by the damped Newton-Raphson method. The PREMIX code was modified to include the solid energy equation and the four differential equations resulting from the spherical harmonic

approximation for the RTE. As noted earlier, the computational domain was extended on both the upstream and downstream sides of the porous layer in order to accurately model flames situated close to the edge of the porous medium.

It must be emphasized that due to combustion effects, the flame location will depend upon the flow rate. If the mass flow rate is fixed, then the flame location will be an outcome of the numerical solution. However, in such an approach there is no control over the flame position. Moreover, in such an approach, it is not clear if non-convergence means that a physically plausible flame does not exist or that it is just a difficult problem to solve numerically. Since one of the aims of this work is to study the effect of flame location on burner performance, and to avoid the above difficulty, the flame location was fixed and the corresponding flame speed was computed as an eigenvalue. The flame location was fixed numerically by setting the gas temperature at a particular point in the domain at 1100 K.

3.3 Test Cases and Computations

The governing equations were solved using the IBM 3090 computer. The time required to solve for one set of parameters ranged widely from approximately 100-500 cpu seconds, depending on the initial guess provided. Results generated from the heat transfer model were used as initial guesses for the gas and solid phase temperatures and the four dependent variables in the coupled ODE's resulting from the

spherical harmonics approximation. The initial guesses for the species profiles were assumed to be gaussian curves (Kee et al., 1985). A relative convergence criterion of 10^{-4} was specified for the Newton's iteration. Computations were continued on successively finer grids until the flame speed changed by no more than 0.1 percent. An overall energy balance of approximately 0.5 percent was achieved for most of the cases, with an energy balance of less than 1 percent in the worst case. Approximately 100 grid points were required. Reducing the grid refinement criterion had a negligible effect on the results but required more computer time. The computational domain was large enough so that flame speed was not affected by the length of the computational domain (Smooke et al., 1983). The accuracy of the numerical code was checked by comparing the results with laminar flame speed and adiabatic flame temperature data of Reitz and Bracco (1982) by letting the porosity be close to one (0.999). For stoichiometric mixtures, an adiabatic flame speed of 39.2 cm/s and an adiabatic flame temperature of 2307 K was obtained which compared well with 38 cm/s and 2280 K, respectively, obtained by Reitz and Bracco (1982). Increasing the porosity beyond 0.999 had a negligible effect on the flame speed and temperature.

The model is used to calculate results for a wide range of porous material properties and operating conditions that may be encountered in PRBs. In particular, the effect of the

extinction coefficient, scattering coefficient, solid thermal conductivity, heat transfer coupling, upstream environment reflectivity, flame location, and stoichiometry on flame speeds and heat transport is investigated. The porous layer is assumed to be 5 centimeters long with a porosity (ϕ) of 0.95. A methane-air mixture is allowed to enter the duct with a uniform velocity and inlet temperature, $T_i = 298$ K. The burner is assumed to see a black surface at $T_e = 298$ K on the downstream side. Variations were made in the following baseline property values to produce the results presented below: $\sigma_e = 20 \text{ m}^{-1}$, $\sigma_s = 10 \text{ m}^{-1}$, $k_s = 10 \text{ W/m-K}$, $h_a = 2.0 \times 10^9 \text{ W/m}^3\text{-K}$, $r_d = 0$ and $\Phi = 1.0$. The corresponding relevant non-dimensional parameters are: $\tau = 1$, $\omega = 0.5$, $H_1 = 6.67 \times 10^7$, $H_2 = 133.3$, $H_3 = 1$. The range of the parameters that were varied in the study is given in Table 3.1.

3.4 Results and Discussion

Flame structure

The structure of a flame located at the center of the porous medium ($\xi_f = x_f/L = 0.5$) is presented in Fig. 3.2. Species concentration profiles for CH_4 , O_2 , N_2 , CO_2 , and H_2O are shown along with the non-dimensionalized gas density and velocity. The gas density is non-dimensionalized with respect to $\rho_1 = 1.13 \text{ kg/m}^3$ which is the density of the incoming reactant mixture. The velocity is normalized with respect to the adiabatic laminar flame speed $u_{ad} = 39.2 \text{ cm/s}$. The flame speed is considerably higher than in the adiabatic

Table 3.1 Summary of the non-dimensional parameters
in the combustion model

Parameter	Baseline Value*	Range Studied
$\tau = \sigma_e L$	1	1-40
$\omega = \sigma_s / \sigma_e$	0.5	0-1
$H_1 = haL / \sigma T_i^3$	6.67×10^7	$3.33 - 3.33 \times 10^8$
$H_2 = k_s / \sigma T_i^3 L$	133.3	1.33-666.7
$H_3 = hL / k_s$	5	none
r_d	0	0-1

* Based on $\sigma_e = 20 \text{ m}^{-1}$, $\sigma_s = 10 \text{ m}^{-1}$, $L = 0.01 \text{ m}$, $T_i = 298 \text{ K}$,
 $ha = 2 \times 10^9 \text{ W/m}^3 \text{ K}$ and $k_s = 10 \text{ W/m K}$.

Fig. 3.2 Flame structure

case, shown with a dashed curve. The flame thickness is reduced due to the presence of the porous medium.

Effect of position on flame speed: the flame speed curve

It must be recalled that in the heat transfer model, the flame zone and the flow rate were fixed independently. When combustion is accounted for, the flame location is expected to have a strong effect on the flame speed. Thus, foremost, it is desirable to study the flame location on the flame speed.

Shown in Fig. 3.3 is a typical flame speed curve, or the plot of flame speed vs. flame location, calculated for the baseline case. The flame location is defined as the position where the reaction rate is maximum. The upstream (inlet) and downstream (outlet) edges of the porous medium are denoted by vertical dashed lines at $\xi_f = 0$ and $\xi_f = 1$, respectively. The flame speeds are normalized by the adiabatic flame speed ($u_{ad} = 39.2$ cm/s). The flame speed approaches the adiabatic flame speed value outside the porous medium, as expected. The curve is characterized by a sharp minimum near the upstream edge, a broader local maximum at the middle of the porous medium and a sharp local minimum and maximum near the downstream edge. As will be seen later, the maximum may be at the middle or at the downstream edge of the porous medium, depending on its properties. In order to understand the complicated nature of the curve, it is necessary to study the

Fig. 3.3 Flame speed curve for the baseline case

local fluxes inside the porous medium that will clarify the relevant mechanisms of flame propagation.

Figure 3.4 shows the detailed heat fluxes in the porous medium as a function of the flame location for the baseline case. For laminar adiabatic gaseous flames with no radiation effects, it has been shown (Von Elbe, 1987) that the flame speed is proportional to the conduction of heat from the flame to the incoming reactants. It is thus illustrative to plot the conductive flux at the upstream edge at the upstream edge of the reaction zone (Q_u^C). The edge of the flame zone is taken to be the location where the reaction rate is 1/1000 of its peak value. It is seen that Q_u^C is almost constant inside the porous region away from the edges, so that the variation in flame speed in that region is due solely to the changing preheat by radiation. The increased conductive flux near the downstream face of the porous medium leads to higher flame speeds there; whereas, at the upstream face of the porous medium, the conductive flux and, hence, the flame speed diminishes. These observations are consistent with the analytical results leading to eq. (B.20) in Appendix B. Note that the conductive flux at the downstream edge of the reaction zone (Q_d^C) is almost zero and does not affect the flame speed. The preheating contribution due to radiation (proportional to $Q^{\tilde{r}}(0) - Q_u^{\tilde{r}}$) reaches a maximum at the center of the porous medium and reduces to zero near the edges. The quantity $Q_u^{\tilde{r}}$ is the dimensionless radiation flux at the

Fig. 3.4 Dimensionless energy fluxes: explanation of the flame speed curve

upstream edge of the flame, and $Q^r(0)$ is the radiative loss in the upstream direction so that $Q^r(0) - Q_u^r$ represents the net radiative flux absorbed by the porous medium in the preheat zone. The radiation preheating of the reactants is zero when the flame is near the upstream edge because there is no porous material upstream of the flame to absorb the radiation from the flame and transfer the energy to the gas. On the other hand, it is zero when the flame is near the downstream edge since there is not enough material downstream of the flame at an elevated temperature to radiate and cause preheating. Near the edge of the medium, it is the thermal conductivity that mainly affects the flame speed, whereas inside the porous matrix, both radiation and conduction are determining factors. A plot of $Q^r(0) - Q_u^r - Q_u^c$ is seen to be very similar to the flame speed curve, and thus the mechanism of flame propagation is amply revealed. It shows that the flame speed is directly proportional to the preheating of the reactants and the conduction upstream of the flame.

Effect of τ

Figure 3.5 presents the computed burning velocities for different flame locations, ξ_f , obtained by varying the extinction coefficient ($\sigma_e = \sigma_a + \sigma_s$) from 20 m^{-1} to 200 m^{-1} so that the total optical thickness of the layer ($\tau = \sigma_e L$) ranges from 1.0 to 10.0 while the scattering albedo ($\omega = \sigma_s / \sigma_e$) is kept constant at 0.5. The optical depth does not affect the flame speeds near the edges. Thus it is confirmed

Fig. 3.5 Effect of burner optical thickness on the flame speed for different flame locations

that the conduction alone governs the flame speeds near the edge of the porous medium while radiation contributes to the variation of flame speeds only in the porous region away from the edges. An increase in optical depth from one to ten increases the flame speeds at all flame locations away from the edge due to increased radiative preheating.

The spatial variation of the gas temperature for a flame situated at the center of the porous medium is shown in Fig. 3.6 for different optical thicknesses of the medium. When the optical thickness is increased from 1 to 5, the temperature in the preheat zone rises due to increased radiation feedback from the post-flame region to the preheat zone yielding higher flame speeds. The enhanced shielding effect accompanying any further increase in τ causes the temperatures in the preheat zone to decrease. The peak temperature is a maximum at $\tau = 10$ and decreases when τ is either increased or decreased by changing the absorption coefficient or the flame location. The solid and gas temperatures coincide here due to the high heat transfer coefficient imposed.

The temperatures in the post-flame region increase with increasing τ . For large τ (≥ 20) a sharp drop in temperature is observed near the downstream edge of the porous medium. This behavior is due to the fact that deep inside the post-flame zone, the media is optically thick thereby reducing the mean path for radiation substantially; whereas near the edge

Fig. 3.6 Effect of optical thickness on gas temperature when $x_f/L = 0.5$

radiation emanates in the downstream direction from a layer that is optically thin.

The effect of the optical thickness on combustion-heat transfer behavior deep in the porous region is illustrated in Fig. 3.7. It is evident that the reactant enthalpy flux Q_c (proportional to the flame speed) increases as τ is increased from 1 to about 10 due to enhanced radiation feedback across the flame in the upstream direction. This explanation is substantiated by the radiative energy fluxes at the upstream and downstream edge of the flame zone. The quantities Q_u^r and Q_d^r in the figure are the radiant fluxes on the upstream and downstream edge of the flame zone. However, as τ is increased further beyond 10, the diminishing radiative energy feedback across the flame zone lowers the reactant enthalpy flux. The output radiative efficiency, η_o , defined as the ratio of the radiant output (radiant heat flux at $\xi = x/L = 1$) and the reactant enthalpy flux, reaches a maximum at $\tau \sim 5$ and then decreases very gradually.

Shown in Fig. 3.8 are the upstream and downstream radiative fluxes (at $\xi = x/L = 0$ and $\xi = x/L = 1$, respectively) resulting from different flame locations and optical thicknesses. The upstream and downstream radiative fluxes depend on the heat generation rate, which is proportional to the flame speed, and the optical depths between the flame and the edges of the porous medium. For a fixed τ the upstream radiative flux increases as the flame

Fig. 3.7 Effect of optical depth on the radiation feedback, flame speed and radiative output efficiency when $x_f/L = 0.5$

Fig. 3.8 Radiant output (downstream radiative flux) and upstream radiative flux for varying optical thicknesses and flame locations

moves upstream from the outlet plane due to an increase in flame speed and because there is less shielding between the flame and the upstream edge. The heat flux continues to increase due to the reduced shielding until the sharp drop in flame speed near the inlet occurs. When the flame is near the downstream edge, the shielding effect dominates, so that the upstream radiative flux is higher for lower τ . However, when the flame is outside the porous layer and near the upstream edge, there is no shielding effect involved and the upstream radiant flux is largest for the largest τ due to greater emission. Similarly, the downstream radiative flux increases when the flame is moved upstream due to an increase in the flame speed and due to an increase in the extent of porous material at elevated temperatures. As the flame moves further upstream, the radiative output decreases eventually because the reduction in flame speed and shielding on the downstream side outweighs the increase in the amount of high-temperature porous material. The shielding effect is more important for larger optical thicknesses; hence, the peak in the radiant output occurs at more downstream positions. For the flame located upstream of the porous region, due to the shielding effect, the radiant output is larger for a smaller τ . These results are consistent with those obtained from the heat transfer analysis where the peak in the radiant output shifted downstream with increasing τ even when the flow speed was held constant.

The radiative output efficiencies for different flame locations and optical depths are shown in Fig. 3.9. The efficiencies are nearly zero when the flames are at the outlet plane and increase quickly at higher τ as the flame is moved upstream. It is interesting to note that the highest efficiencies are achieved when the flame is located outside the porous medium near the upstream edge and that the efficiency at this position increases with decreasing optical depth.

Effect of ω

Shown in Fig. 3.10 are the distribution of flame speeds for different albedos. When $\omega = 1$, the solid does not emit or absorb radiation, while $\omega = 0$ corresponds to the case when the solid strongly emits and absorbs radiation. As expected from the results in Fig. 3.4, the flame speed deep inside the porous medium for $\omega = 1$ does not change with the flame location. As ω is decreased a stronger radiation feedback from the post-flame region to the preheat region increases the flame speed inside the porous medium away from the edges. It must be noted that when there is no emission or absorption by the solid, i.e. when $\omega = 1$, the flame speed exceeds the adiabatic value because the thermal conductivity of the solid matrix is much greater than the gas.

Temperature profiles for different scattering albedos are shown in Fig. 3.11. A decrease in the albedo causes a stronger feedback of radiation, increasing the temperatures

Fig. 3.9 Variation of radiative output efficiencies for different optical thicknesses and flame locations

Fig. 3.10 Effect of albedo on flame speeds for different flame locations

Fig. 3.11 Effect of albedo and solid thermal conductivity on the gas phase temperature for $x_f/L = 0.5$

in the preheat and reaction zone. The exhaust temperatures drop as albedo decreases due to a greater amount of radiative heat loss from the downstream side of the porous material.

The detailed energy fluxes in the vicinity of the flame are shown in Fig. 3.12 for different scattering albedos. A decrease in albedo results in higher convective, reactive, and radiative energies. It is interesting to note that the gas and solid conductive fluxes near the upstream edge of the reaction zone do not change appreciably with a change in the albedo. Again it can be seen that as the albedo is decreased a stronger radiation feedback from the post-flame zone to the preheat zone occurs which results in higher flame speeds.

Figure 3.13 shows the effect of the scattering albedo on the burner characteristics for a flame located at $x_f/L = 0.5$. A reduction in albedo increases the flame speed due to stronger emission from the solid. It must be noted that when there is no emission or absorption by the solid, i.e. when $\omega = 1$, the flame speed exceeds the adiabatic value because the thermal conductivity of the solid matrix is much greater than the gas. Both the radiant output and the efficiency also increase with a decrease in the albedo; hence, a strongly emitting-absorbing medium is preferred to enhance the burner performance. The results demonstrate that scattering by the solid is an important mechanism in the radiative transport and cannot be neglected.

Fig. 3.12 Energy fluxes in the flame vicinity for different albedos

Fig. 3.13 Burner characteristics for different albedos and upstream reflectivities when $x_e/L = 0.5$

Effect of H_1 , heat transfer coupling

The effect of the heat transfer coupling between the solid and the gas is reported in Fig. 3.14. A stronger heat transfer coupling results in a better pre-heating effect when the solid conductivity is higher than that of the gas.

Hence, the flame speed increases as the heat transfer coupling is increased. Once the coupling is large enough so that the gas and solid are in local thermal equilibrium, the rise in flame speed becomes negligible. On the other hand, when the heat transfer coupling is weak, very little energy is transferred from the gas to the solid so that the flame speed approaches the adiabatic flame speed and the radiant output approaches zero. This asymptotic approach to the adiabatic flame speed has served as an additional check of the code. The radiative output increases as the heat transfer coupling is enhanced because of higher solid temperatures, and no appreciable increase occurs once the solid and gas have reached thermal equilibrium. The radiant efficiency is observed to pass through a maximum as the heat transfer coupling is enhanced.

Effect of H_2 , solid thermal conductivity

Temperature profiles for different conductivities are shown in Fig. 3.11. The effect of the variation in conductivity is revealed for the case when the solid does not emit or absorb radiant energy (i.e., $\omega = 1$). The temperature gradient in the flame zone does not change with a variation

Fig. 3.14 Effect of heat transfer coupling on burner performance for $x_f/L = 0.5$

in conductivity, but the peak temperature rises slightly with an increase in the conductivity. This plot also displays the significant difference in the mechanism by which the flame speed is augmented by either radiation or conduction. A comparison of the curves for the same solid conductivity but different albedoes reveal that the radiation feedback affects the whole length of the porous region upstream of the flame, and this length changes with the scattering albedo. Whereas the solid-phase conduction heating is confined very near the reaction zone for most k_s 's of interest.

Shown in Fig. 3.15 are the burner characteristics for different thermal conductivities of the solid. An increase in the thermal conductivity increases the flame speed due to an enhancement in the preheating of the incoming reactants. The trend is consistent with the analytical expression in eq. (B.20) when $k_u = k_d$. However, the increase in the solid conductivity does not increase the radiant output to the same extent as the flame speed so that the radiative efficiency actually declines. This result is due to the fact that for the same flowrate and heat release rate, an increase in the solid conductivity produces lower post-flame temperatures and higher pre-flame temperatures, thereby reducing the radiative output as was demonstrated in the heat transfer model. Thus, the radiant output can be increased by increasing the conductivity and concurrently sacrificing on the radiant efficiency.

Fig. 3.15 Effect of the solid thermal conductivity on the burner characteristics when $x_f/L = 0.5$

Effect of r_d

Figure 3.13 shows the effect of the upstream environment (at $\xi = -x_1/L = -\xi_1$) reflectivity on the burner characteristics for a flame located at $x_f/L = 0.5$. An increase in the upstream reflectivity reduces the radiation loss from the flame in the upstream direction thereby raising the flame temperature and increasing the flame speed. The radiant output increases when the reflectivity is increased because the flame speeds are higher and less radiation escapes in the upstream direction. This latter effect also improves the output efficiency and suggests that a large reflectivity is desired for the best burner performance. These results are consistent with the findings from the heat transfer model.

Figure 3.16 shows the effect of the stoichiometry on the flame speeds that have been normalized by the flame speed prevailing when the equivalence ratio, Φ , is one. Stable combustion can evidently be maintained for very lean mixtures ($\Phi = 0.4$), which is consistent with the experimental findings of Kotani and Takeno (1982). The flame thickness increases with decreasing Φ to allow longer preheat distances. This result is consistent with the observation of wider flames near the extinction limit. It is interesting to note that the total radiant efficiency is enhanced as the reactant mixture becomes leaner. This result stems from the fact that

Fig. 3.16 Flame and burner characteristics for different equivalence ratios

the temperatures in the post-flame zone do not decrease in proportion to the flame speed as the mixture gets leaner.

Flame stability

The stability of the flame is of primary importance in the operation of practical systems such as porous burners and in establishing safety limitations in gas-solid two-phase systems. In theory, a stable flame may be obtained by matching the flow speed to the flame speed, as mentioned earlier. However, no practical device can provide a constant flow rate and flow fluctuations, however small, are inevitable. If during steady-state operation a flame is located in the region of the flame speed curve where the slope is negative, a sudden increase in the flow velocity will cause the flame to drift in a downstream location where the flame speed is lower, causing a further drifting of the flame in the downstream direction. This would prevent the flame to operate steadily in a fixed location. Similarly, a sudden decrease in the flow velocity would tend to induce a flashback. If the flame is however located in the positive slope region, a small sudden increase in the flowrate would lock the flame in a closeby downstream location. It is thus speculated that the regions where the flame speed curve has a positive slope are the stable operating regions.

In order to check the consistency of the model with these trends the solution of the corresponding transient flame propagation problem was attempted. This was

accomplished by imposing a sudden increase in flowrate on the steady-state flame speed eigenvalue solution and then observing the location of the flame with time. In this case, the flame speed was kept fixed. The calculations, carried out over a few short time steps of 10^{-7} seconds, indicate that for all initial flame positions the flame was shifted downstream when the mass flowrate increased. Thus, the steady-state flames predicted to lie in the region where the slope of the flame speed vs. location curve is negative are inherently unstable with respect to changes in flow conditions. These findings suggest that the flame can be stabilized in the upstream half or near the downstream edge of the porous medium. The experimental results obtained by Kotani and Takeno (1982) seem to indicate such spatial domains of stable flame operation. This phenomenon will be tested in the experimental program described in the following chapter.

3.5 Conclusions

The effect of combustion and multimode heat transfer on the performance characteristics of PRBs has been studied numerically. The model incorporated emission, absorption, and scattering of radiant energy by the porous matrix and accounted for non-local thermal equilibrium between the gas and solid phases. The computational domain was extended on either side of the porous medium to accurately model the flames located near the edge of the porous medium. The flame

location had a strong effect on the flame speeds. Stable flames were found to occur in two different spatial domains. It was shown that the flame speed variation near the edge of the porous medium is controlled by conduction; whereas, radiation heat transfer becomes a contributing factor for flame propagation in the interior. It was revealed that for maximizing the radiant output, the optical depth should be approximately ten with the flame located near the center of the porous medium, and the heat transfer coupling should be high enough to ensure local thermal equilibrium between the gas and the solid. The solid-phase conduction and scattering had a significant effect on the burner performance. Low solid conductivities and scattering albedos result in higher radiant efficiency.

CHAPTER 4

EXPERIMENTAL VERIFICATION

In the previous chapter, a numerical model was developed to study the combustion and heat transfer characteristics of PRBs. The theoretical model predicted the existence of two spatial domains of stable combustion. The burner performance was studied as a function of the various parameters. The aim of the experimental study is to verify the theoretical predictions. In particular, the existence of the stability regions and some trends in flame speeds and radiant outputs will be verified.

4.1 Experimental Apparatus

The schematic diagram of the experimental setup is shown in Fig. 4.1. The setup consists of an inlet section, a burner section and an exhaust section. The accompanying apparatus includes the fuel/air sources, flow rate measuring devices, and an optical arrangement to measure the radiation from the burner. The description of the optical setup can be found in Appendix D.

The inlet section consists of a 4.2 in. diameter x 12 in. long water-cooled vertical steel pipe through which premixed methane-air is introduced. A honeycomb piece is inserted in the pipe to straighten the flow. A steel screen is affixed to arrest the flame in case of a flashback. A relief valve is affixed to the inlet section to release the

Fig. 4.1 Schematic diagram of the apparatus

pressure in the event of a flashback. The burner section consists of the cylindrical porous materials, insulated from the environment by 2-in. thick high-temperature ceramic insulation (Zircar Industries, Zirconia felt type ZYF-100). The insulation was used to maintain one-dimensional heat transfer as assumed in the model. The exhaust section consists of a steel manifold, connected to four horizontal 3-in. diameter steel ducts (Flexhaust Ind., Bendway type S) to carry away the combustion products. The exhaust manifold has a 1-in. diameter aperture on the top side to enable collection of radiant energy from the burner. Cold water/ethylene glycol mixture from a constant temperature bath (Forma Scientific, model 2095) is circulated through 1/4 in. stainless-steel tubes attached to the exhaust manifold to control the temperature of the downstream environment of the burner. The burner is separated from both the inlet and exhaust sections by 1/8 in. thick high-temperature gaskets made of the zirconia felt. All leaks near hot surfaces were sealed with high-temperature zirconia cement (Zircar Industries). All interior sections were coated diffusely with black paint so the upstream and downstream environment reflectivities were zero. Pressure taps were drilled into the inlet and exhaust sections to monitor the pressure drop across the porous samples.

4.2 Porous Samples

Due to the fact that not much research has been done on the subject of porous radiant burners, limited knowledge exists on the type of burner materials that are readily available. A practical burner material would be one that can withstand high temperatures, resist thermal shock and have minimum pressure drop. From the standpoint of withstanding high temperatures, ceramic or a few metallic porous materials would be suitable. Two broad categories of ceramic porous materials were available in the market. One category consisted of fiber-mat/board materials (Duraboard by Thermal Ceramics Inc., Zirconia/Alumina boards by Zircar Industries, Fiberfrax felts by Carborundum Co.). These are made of ceramic fibers, a few microns in diameter, held together by organic binding agents. The other category consisted of ceramic foams or sponges with a reticulated dodecahedral structure (ceramic foam from Hi-Tech. Ceramics Inc. and Refractory Composites Inc.). The foams are available in a large variety of pore sizes ranging from a tenth of a millimeter to a few millimeters. Manufacturers of metallic porous materials that could withstand flame temperatures could not be located.

4.3 Material Testing.

After some initial testing, it was determined that the ceramic foams with larger pore sizes (10 pores per inch) were well suited for validation of the model. These had

negligible pressure drop compared to the same thickness of the fiber mat/board type materials or the smaller pore size foams. Two types of foam materials, Lithium Aluminum Silicate (LAS) and Silicon Carbide were considered based on the superior thermal shock resistance. The composition was 96-97 percent fused aluminum silicate ($2\text{Al}_2\text{O}_3 \cdot 3\text{SiO}_2$) and silicon carbide (SiC), respectively. Lithium is added to aluminum silicate in small quantities to impart better thermal shock resistance. A high emissivity coating containing lithium and other substances raises the surface emittance of the LAS foam above 0.9.

The SiC foams cracked very soon, even after initial heating and slow cooling, so that it was extremely difficult to prevent non-uniformities in the heat transfer and flame speeds across the flow cross section. Thus all validation was done with the Lithium Aluminum Silicate material. The properties of the LAS material are given in Table 4.1. The extinction and scattering coefficients are estimated in Appendix C. The thermal conductivity of the solid is estimated from data in the literature (Touloukian and Dewitt, 1972a).

4.4 Experimental Procedure

Experiments were performed on the LAS material 2-in. long and 4.8 in. diameter. The porous disk had a 0.1-in. thick wall made of the same material. Experiments were run for different equivalence ratios and flame locations. Both

Table 4.1 Properties of the LAS foam

Property	Value
Optical depth, $\tau = \sigma_e L$	10.4*
Albedo, $\omega = \sigma_s / \sigma_e$	0.1
Solid thermal conductivity, k_s	1 W/m K
Porosity, ϕ	0.85

* Based on $L = 2$ in.

flame speeds and the radiant outputs were compared with the theoretical model. In preliminary runs it was observed that either lab grade methane (99 percent purity) or the less expensive commercial grade methane (93 percent purity) produced identical flame speeds. Changing the flow area from 4.2 in. inlet section dia. to 4.8 in. porous material dia. either gradually via a tapered flange or suddenly did not change the flame speed or radiant output results for all locations of the flame.

The flame was ignited by a propane torch at the downstream edge of the porous medium by adjusting the mass flow rate and equivalence ratio favorable for ignition. It was necessary to maintain a stable flame at the downstream edge for some time (15-30 minutes) so as to heat the block slowly and prevent cracking. The stable flame at the downstream edge is characterized by a thin blue sheet with negligible radiant output. The equivalence ratio was then increased slightly by increasing the fuel flow rate, thus keeping the mass flow rate approximately the same. This increases the flame speed and thus the flame starts traveling upstream and stabilizes in the upstream half of the porous material or flashes back, depending upon the fuel rate adjusted. The flame position then could be adjusted by changing the air flow rate and the fuel flow rate, keeping the same equivalence ratio. It took about 20 minutes to attain steady state thereafter.

Four one-millimeter wide slits placed vertically across the insulation surrounding the porous material facilitated visual observation of the flame. The flame could be seen as a luminous ring (bright orange glow) about 0.1 in. thick on the foam wall. The portion upstream of the ring was darker in a sharp contrast with the ring, while the downstream portion maintained a dull orange glow. Both visual observation and temperature measurements indicated the same position of the flame, taken to be the position where the luminosity and temperature were the highest. The exact location was recorded with a pointed scribe with respect to a reference base and measured with vernier calipers of least count 0.01 in. It was observed that flames could be stabilized only in the upstream half and at the downstream edge of the porous material, thus confirming the existence of the two spatial domains of flame stability.

It is important to note that the transient phenomena play a big role in the way a steady flame is obtained. It was observed that it was extremely difficult to obtain a uniformly flat flame, perpendicular to the flow axis in almost seventy-five percent of the attempts to stabilize the flame in the upstream half. Thus, transient flame propagation must be studied in the future, as it has important implications on the manner in which a flame can be stabilized.

4.5 Data Reduction

Steady-state flames were obtained in the upstream half of the porous medium. The rotameter readings and the radiation detector reading were recorded in order to compute the flame speed and radiation output, respectively. The experiments were run for different equivalence ratios and flame locations. The raw data for all the observations are presented in Appendix E. The flame speed was calculated from the cross-sectional area of the porous material and the measured mass flow rate of the fuel and air (see section E.2). The detector is calibrated against a blackbody source. The radiant output is computed by using this calibration. A complete description of the optical setup, calibration of the detector and measurement procedure can be found in Appendix D. In order to check the overall accuracy of the experiment, an energy balance was conducted on the system and the energy balance could be closed within 5 percent (Appendix E).

4.6 Results

Flame speeds and radiant output

Shown in Figs. 4.2 and 4.3 are the flame speeds and radiative outputs, respectively, for equivalence ratios of 0.55 and 0.60. Shown in solid curves and symbols are the corresponding values obtained from the theoretical predictions. Note that the results are for stable flames in the upstream half of the porous medium. The flame speeds and

Fig. 4.2 Experimental data for flame speed

Fig. 4.3 Experimental data for radiative output

the radiant output increase as the flame is shifted towards the center of the material for a fixed Φ . The flame speed and the radiant output also increase with increasing fuel content for a fixed flame location. The radiative efficiency varied from 26-40 percent in all the runs in Figs. 4.2 and 4.3. The efficiency was highest for leaner mixtures, which is consistent with the theoretical predictions. The results shown are for two different porous sample pieces and show good consistency.

The discrepancy in the experimental and numerical results ranges from 3.4 to 16.7 percent in the flame speeds. The corresponding figures for the radiative heat flux were 3.9 to 16.8 percent. The uncertainties in the flame speeds and radiant outputs were about five and nine percent, respectively (see appendix E) and are shown as a hatched band around the data points. Also, there is some uncertainty in the numerical results, as indicated by error bars, because of the uncertainty in the values of the various parameters (see Appendix C). Accounting for the various uncertainties, the theoretical predictions are seen to match very well with the experimental results. The numerical results were relatively more sensitive to the heat transfer coupling (h_a) than other parameters. It is worth mentioning that the uncertainty in the h_a value arising from the fact that the fibers are not in cross flow but at random directions to the flow has not been included. Thus there is a need to conduct further research

on developing an in-situ method for estimating this parameter.

Difficulties were observed in stabilizing the flame very close to the upstream edge, and such an attempt always resulted in a flashback or flame oscillations. Flame oscillations have been observed in flames close to extinction. The very low values of steady-state flame speeds at the upstream edge obtained numerically also suggest physically implausible flames or flames close to extinction. Flame speeds ranging from 15.2 to 17.3 cm/s could be stabilized at the downstream edge for $\Phi = 0.5$. Numerical solutions for this case could not be obtained due to difficulty in obtaining convergence.

Wall temperature profiles

The wall temperature profiles for equivalence ratios of 0.5 and 0.6 are shown in Fig. 4.4 for the non-dimensional flame locations of 0.4 and 0.2, respectively. The wall temperatures were measured with twenty five gage 30 chromel alumel thermocouples (Omega Ind.) cemented to the wall by alumina cement. The wall temperatures are comparable with the solid phase temperature predicted from the combustion model. Based on a radial heat loss estimate of one percent, the radial temperature drop at the hottest section is estimated to be about 60 K. This could account for some difference in the predicted solid temperature and the wall temperature at the hottest section. The difference could

Fig. 4.4 Wall temperature profiles

also be possibly explained by the uncertainties in the heat transfer coupling, as pointed out earlier.

4.7 Conclusions

An experimental investigation was performed to study laminar premixed combustion in an inert porous medium. As predicted by the numerical study, it was found that stable combustion could be maintained in two spatial domains, one in the upstream half of the porous segment and another at the downstream edge of the porous segment. Tests were conducted with methane-air mixtures in Lithium Aluminum Silicate foams. The flame speed and radiative output were highest when the flame was at the center of the porous material. The radiant efficiency was higher when the mixture was leaner. The experimental trends and values for the flame speeds, radiant output and solid temperatures agreed favorably with the theoretical predictions.

CHAPTER 5

CONCLUSIONS AND RECOMMENDATIONS

5.1 Conclusions

A numerical and experimental investigation was conducted to study combustion and heat transfer characteristics of porous radiant burners (PRBs). A body of knowledge relevant to the fundamental mechanisms of flame propagation in PRBs was generated. The effects of combustion and heat transfer on PRB performance were established.

The theoretical model considered one dimensional steady-state heat transfer and combustion in a porous segment. The combustion was modeled using one-step global chemistry for methane-air mixtures. Results obtained from the heat transfer model served to provide initial guesses for the more complicated model when combustion was accounted for. It was revealed that both upstream conduction and radiation from the flame affected the flame propagation when the flame was situated away from the edges of the porous medium. At the edges, the conduction from the flame in the upstream direction affected the flame speeds and the radiative transfer had a much smaller effect. The results suggested the existence of two spatial domains for stable flame operation, one in the upstream half of the porous layer, and the other in a very narrow region at the downstream edge of the porous medium. To enhance the radiative output from PRBs, it is necessary to have an optically thickness of about

ten. The fibers constituting the porous material should be small enough to ensure local thermal equilibrium between the solid and the gas. The radiative output and efficiency are highest when the scattering albedo is zero. Higher solid thermal conductivities enhance the flame speeds, but reduce the radiative efficiency.

Experiments were conducted using premixed methane-air mixtures and alumina silicate foam to verify the theoretical model. Flame speeds, radiative outputs and temperature profiles were measured for different equivalence ratios and flame locations. Results confirmed the existence of the two spatial stability domains. The trends and values for the radiant outputs, flame speeds and temperature profiles were in good agreement with the theoretical findings.

5.2 Recommendations for Future Studies

Based on the findings in this work, the following are recommended for future research:

- (1) Flashback and blowoff phenomena should be studied numerically using a transient flame propagation model. These phenomena have significant implications in the useful and safe operation of PRBs.
- (2) An in-situ experimental method should be developed to accurately estimate the heat transfer coefficient between the gas and solid phase in porous foams.
- (3) Rugged ceramic materials that are less prone to thermal shock must be developed in the future.

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APPENDIX A

Estimation of Gas Emissivity

Assuming a methane-air flame at atmospheric pressure and 1627 K with an equivalence ratio of 0.91, the equilibrium distribution for the major product species in terms of mole fraction is (Gordon and McBride, 1971):

CO₂ : 0.087, H₂O : 0.174, N₂ : 0.713 and O₂ : 0.017.

Nitrogen and oxygen have symmetric diatomic molecules and hence are transparent to thermal radiation. Thus the major species contributing to the thermal radiation are CO₂ and H₂O. The gas emittance for a CO₂-H₂O mixture is given by the mean beam length theory (Seigel and Howell, 1981) as

$$\epsilon_g = C_{\text{CO}_2} \epsilon_{\text{CO}_2} + C_{\text{H}_2\text{O}} \epsilon_{\text{H}_2\text{O}} - \Delta\epsilon \quad (\text{A-1})$$

where C is a correction factor for pressure and $\Delta\epsilon$ is a correction factor for band overlap. Using the appropriate charts in the above reference in conjunction with the mole fraction data obtained above, we have the following results

mean beam length (in.)	1.0	6.0	12.0
gas emittance ϵ_g	0.010	0.057	0.098

The above figures for ϵ_g are approximate due to errors caused by reading the charts, but nevertheless give a good idea about the magnitude of ϵ_g . Since the solid emittance is of the order of 0.1, the radiation contribution from the gas can be neglected for small enough mean beam lengths.

APPENDIX B

Analytical Solution for Flame Speeds Near the Edge of the
Porous Medium

It is possible to obtain an approximate expression for the flame speed near an interface when the material is not radiatively participative. The analysis is based on Fig. B.1 wherein the temperature profile in a typical flame is considered. Reactants flow in at temperature T_i at $x = -\infty$ and are ignited at an ignition temperature of T_{ig} at $x = 0$. The burnt gas flows out at temperature T_f at $x = x_f$. The effective thermal conductivity of the medium upstream of the interface ($x < x_j$) is k_u and that of the downstream medium is k_d . The reaction zone extends from $x = 0$ to $x = x_f$.

Writing an energy balance in the different regions we have

$$k_u T'' - \rho u c T' = 0 \text{ for } x < 0; T(-\infty) = T_i, T(0) = T_{ig} \quad (B.1)$$

$$k_u T'' + \omega' H = 0 \text{ for } 0 < x < x_j; T(0) = T_{ig}, T(x_j) = T_j \quad (B.2)$$

$$k_d T'' + \omega' H = 0 \text{ for } x_j < x < x_f; T(x_j) = T_j, T(x_f) = T_f \quad (B.3)$$

where the prime denotes derivative w.r.t. x . The convective flux has been assumed to be constant in the reaction zone in obtaining Eqns. (B.2) and (B.3). The reaction rate is given by $\omega' = A \exp(-E/RT)$ and H is the heat of reaction.

Equating the heat flux at $x = 0$ we have

$$T'(0^-) = T'(0^+) \quad (B.4)$$

Fig. B.1 Schematic diagram for analytical flame speed solution near an interface

Integrating Eqn. (B.1) and imposing the condition that $T'(-\infty) = 0$, we obtain $T' = \rho u c (T - T_i) / k_u$ for $x < 0$. Thus

$$T'(0^-) = \rho u c (T_{ig} - T_i) / k_u \approx \rho u c (T_f - T_i) / k_u \quad (B.5)$$

Equation (B.2) is integrated w.r.t. T by first multiplying it by $2T'$ so that

$$-[T'(0^+)]^2 = - \int_{T_{ig}}^{T_j} (2H\omega' / k_u) dT - \int_{T_j}^{T_f} (2H\omega' / k_d) dT \quad (B.6)$$

The above equation has been derived assuming that $T'(x_f) = 0$. To perform the integrations in Eqn. (B.6), we let $\gamma = T_f - T$ so that $T = T_f - \gamma = T_f(1 - \gamma/T_f) \approx T_f / (1 + \gamma/T_f)$ since γ/T_f is small. Using this approximation for T in the reaction rate expression, the above integrations are easy to carry out and the following expression is obtained

$$[T'(0^+)]^2 = \frac{2HA \exp(-E/RT_f) RT_f^2}{E} \left[\frac{\exp(-\beta_j) - \exp(-\beta_{ig})}{k_u} + \frac{1 - \exp(-\beta_j)}{k_d} \right] \quad (B.7)$$

where $\beta = E\gamma/RT_f$ and the subscripts on β refer to the locations in Fig. (B.1).

Since the sum of the thermal and chemical energies per unit mass mixture is constant throughout the combustion zone, we have

$$cT + mH/\rho = cT_i + m_i H/\rho_i = cT_f \quad (\text{B.8})$$

Substituting Eqns. (B.5) and (B.7) in eqn. (B.4)

$$[\rho u c (T_f - T_i) / k_u]^2 = \frac{2HA \exp(-E/RT_f) RT_f^2}{E} \left[\frac{\exp(-\beta_j) - \exp(-\beta_{ig})}{k_u} + \frac{1 - \exp(-\beta_j)}{k_d} \right] \quad (\text{B.9})$$

Using continuity of mass and eqn. (B.8) the following expression is obtained

$$u_i = \left\{ \frac{2A \exp(-E/RT_f) RT_f^2}{Em_i c (T_f - T_i)} \left([\exp(-\beta_j) - \exp(-\beta_{ig})] k_u + \frac{[1 - \exp(-\beta_j)] k_u^2}{k_d} \right)^{1/2} \right\} \quad (\text{B.10})$$

The result clearly shows that the flame speed increases by increasing k_u and decreases by increasing k_d . Thus it is expected that at the downstream edge of the porous medium (where $k_u > k_d$) the flame speed will be higher than the case when the flame is situated in the interior of the porous medium (where $k_u = k_d$). The opposite effect is expected at the upstream edge of the porous medium (where $k_u < k_d$).

APPENDIX C

Estimation of Parameters

In order to validate the theoretical model developed, it is necessary to estimate the property values of the porous material. These are used as input in the numerical code. The heat transfer coefficient, optical depth, scattering albedo, solid thermal conductivity and porosity are estimated.

Heat transfer coefficient

The heat transfer coefficient is estimated from the experimental correlation for flow over circular cylinders (Incropera and DeWitt, 1981). The Nusselt number is given by

$$Nu = hd/k_g = 0.989(Re)^{0.33} \quad (C.1)$$

where h is the heat transfer coefficient, d is the equivalent particle diameter, k_g is the gas conductivity and Re is the Reynolds number based on the particle diameter.

The average fiber diameter of the foam was estimated to be approximately 0.047 in. with an uncertainty of ± 10 percent. This uncertainty comes about because of the complicated dodecahedral structure of the foam wherein the knots at the vertices of the dodecahedrons may result in a different diameter. Such a change in diameter changed the flame speed by almost ± 10 percent for the experimental conditions. Note that the uncertainty due to the fact that the foam fibers are at random directions to the flow and not in cross flow has not been taken into account. The density, viscosity and thermal conductivity values were taken as those for air at 1000 K, needed for evaluation of the Reynolds

for air at 1000 K, needed for evaluation of the Reynolds number and the heat transfer coefficient. Assuming different temperatures other than 1000 K did no change the final value of the heat transfer coefficient by more than 2 percent. It was assumed that the fibers constituting the foam were infinitely long cylinders so that $a = 4/D$. The h_a values ranged from 2.08 - 2.62 W/m³ K for all the experimental runs.

Radiative properties

The optical thickness is estimated from (Wang and Tien, 1983)

$$\tau = 4Q_e(1-\phi)L/\pi d \quad (C.2)$$

where Q_e is the extinction efficiency, ϕ is the porosity, L is the length of the porous medium and d is the particle diameter. For large cylindrical cylinders, such as encountered in the present case, Q_e approaches 1.28 (Wang and Tien, 1983). With $\phi = 0.85$, as determined by the manufacturer and $L = 2$ in., we have $\tau = 10.4$. A change in the optical depth from 8 to 15 changed the theoretical results by less than two percent for conditions close to the experiment.

The radiative scattering from large particles, such as in the present case, is governed by geometrical optics, and can be separated into diffracted and reflected components. The diffracted radiation is concentrated in a small solid angle about the direction of propagation, with the angle of peak intensity inversely proportional to the particle size. On this basis, the scatter properties for large particles are

determined entirely by the reflected component (Hottel and Sarofim, 1967). Thus for large opaque particles, the scattering albedo is given by $\omega = 1 - \epsilon$, where ϵ is the emittance of the particle surface (Hottel and Sarofim, 1967). Such an estimation of the albedos for packed beds has been used as a reasonable approximation (Tien, 1988). The surface emittance of LAS foam is increased beyond 0.9 due to the high emissivity coating and thus the albedo is estimated to be less than 0.1. Numerical runs for typical experimental conditions showed that the numerical results changed by only 4 percent when albedo ranged between 0.2 and 0.0. Thus the present estimate of the albedo is considered to be adequate for comparison with the experiment. The numerical predictions in Figs. 4.2-4.4 are for an albedo of 0.1.

Solid thermal conductivity

The solid thermal conductivity as estimated by the manufacturer was 1 W/m K. Data in literature showed values ranging from 1-3 W/m K for aluminum silicate (Touloukian and DeWitt, 1972a). Changing the thermal conductivity from 1-5 W/m K changed the numerical results by 2.6 percent for the experimental conditions.

APPENDIX D

Optical Arrangement for Radiation Measurement

D.1 Description of the Apparatus

The schematic diagram of the optical setup for measurement of the radiative output from the burner is shown in Figs. D.1 and D.2. The apparatus consists of a blackbody radiation source (Infrared Industries, model IR-464) to serve as a reference for the pyroelectric detector (Oriel Corp. model 7086). The detector response is increased by using a parabolic mirror (Melles Griot model 02POA015) to focus the energy onto the detector. The detector generates an electrical signal read out from a radiometer (Oriel Corp. model 7081). A chopper (Oriel Corp. model 7509) is used to produce an alternating signal to operate the detector and to eliminate background noise. Two pinholes, 1/8 in. dia. were used to restrict the field of view of the detector to encompass only the source. The system is arranged as per Figs. D.1 and D.2. The distances were determined based on the discussion to follow. The sliding flat mirror (M1) was used to allow calibration of the detector against the blackbody radiation, while the flat mirror (M2) was used to direct the radiation from the burner onto the detector.

It can be shown (Harrison, 1960) that the output of the detector depends on the pinhole size, distance of the pinhole from the detector, the focussing mirror and the source intensity. Hence by using the same pinhole size for both the burner and the blackbody and keeping the same pathlength from the pinhole to the detector, the radiation from the blackbody

Fig. D.1 Plan of the optical arrangement

VIEW A

Fig. D.2 Elevation of the optical arrangement

and the burner can be directly compared. It is necessary for the detector to see only the source for such a direct comparison. Assuming the source to be diffuse it can be shown that the detector output (D_o) under such conditions is given by

$$D_o = ME A_p A_d / \pi l^2 \quad (D.1)$$

where M is the constant dependent on the focussing mirror, E is the radiative flux emanating from the source, A_p is the pinhole area, A_d is the detector area and l is the distance between the pinhole and detector. Thus, as long as the detector sees the source alone, the distance of the source from the pinhole is immaterial. These conditions were achieved by maintaining the distances shown in Figs. D.1 and D.2, and thus the burner radiation could be directly measured using the detector. The calibration curve for the detector in the present arrangement is shown in Fig. D.3. The calibration was performed by increasing the blackbody temperature from 600 - 1200 °C in steps of 5 °C. As expected from eq. (D.1) the detector reading is a linear function of the radiant flux emanating from the blackbody. This calibration is then used in computing the burner radiation output. It is implicit that the burner is assumed to radiate diffusely, which is a reasonable assumption because of the porous structure (DeWerth, 1962).

Fig. D.3 Calibration curve for the radiation detector

D.2 Alignment

To obtain accurate readings, the optical components must be aligned well. The alignment should be such that slight movements in the optical components would not affect the detector readings. As mentioned earlier, if the detector sees only the source, then the source position does not affect the detector output. The area of the source seen by the detector is determined by the envelope of rays joining the opposite ends of the diameters of the focussing mirror and the pinhole (Harrison, 1960). The dimensions chosen assured that the detector saw only part of the source. The positions and dimensions of the chopper and plane mirror were such that they did not interfere with this envelope of sight.

The axes of the two pinholes were first marked on the table to set the dove-tail rails. The plane mirror M1 was set at 45° to the pinhole P1 using an accurate machinist's angle and then slid on the rail to the marked position where the axes crossed. The parabolic mirror was mounted in the shown position with the base perpendicular to the pinhole P1 using a large right angle. The blackbody aperture was then placed parallel to the pinhole P1. All the components were raised to 6 in. above the reference table by using machined gages to match the height of the blackbody aperture. Pinhole P2 was then set perpendicular to pinhole P1 in the marked position. The position of the plane mirror M1 was fine adjusted so that the beam from the blackbody fell at the

center of the parabolic mirror. The detector was then placed such that the beam was focussed at the detector so that the detector gave the maximum output at the radiometer. Mirror M2 was set at 45° from the horizontal using an accurate 45° machinist's angle and then the height of the center of the mirror was adjusted to 6 in. above the table. It was then swiveled into the desired position by adjusting the position of its mounting post. The exhaust section was then positioned such that the exhaust port was aligned with the axis of the pinhole P2 and vertically below the center of the mirror M2. The mirror M2 was then fine adjusted to ensure that the beam from the burner fell at the center of the parabolic mirror. A maximum output was obtained in this position of the mirror M2.

D.3 Alignment Check

After alignment was completed the sensitivity of the output to slight movements of all pinholes and the mirrors were checked. It was observed that slight rotary and translatory movements in both the mirrors and the pinholes did not affect the detector output, ensuring insensitivity to vibrations and accidental movements. Moreover, the detector reading did not change when the blackbody was moved along its axis over 1-8 in. from the pinhole P1. This is due to the fact that the source area does not affect the detector output in the present setup, according to eq. (D.1). Similar effect was observed with the burner. Also, tilting the blackbody or

the burner did not affect the output from the detector (Harrison, 1960).

D.4 Measurement

The exhaust port on the burner was kept covered during the burner operation. When the radiation output was to be measured, the exhaust port was uncovered briefly and a high pressure air jet (60 psi.) was blown across the exhaust port to prevent exhaust gases from reaching the mirror M2. After taking the detector reading, the exhaust port was covered and the jet shut off.

APPENDIX E

Data Sheets, Uncertainty Analysis and Energy Balance

E.1 Raw Data

The raw experimental data is given in Table E.1. The quantities given in the table are the fuel and rotameter readings, the detector reading and the flame location. The computed flame speeds and the equivalence ratios are also given.

Table E.1 Raw data

Obs.	Rotameter Reading Φ		u_f (cm/s)	x_f (in.)	Detector Reading $\times 10^3$
	Air	Methane			
1	115	45	0.60	21.9	0.158
2	128	50	0.60	24.3	0.166
3	134	52	0.60	25.3	0.175
4	137	53	0.60	25.9	0.175
5	141	55	0.60	26.6	0.197
6	136	53	0.60	25.7	0.202
7	145	57	0.60	27.3	0.218
8	146	57	0.60	27.5	0.213
9	147	57	0.60	27.7	0.227
10	91	34	0.55	17.6	0.125
11	93	34	0.55	18.0	0.125
12	95	35	0.55	19.4	0.133
13	101	37	0.55	20.3	0.140
14	106	39	0.55	22.7	0.150
15	120	43	0.55	23.1	0.164
16	122	44	0.55	23.1	0.175
17	133	48	0.55	25.1	0.184
18	91	31	0.50	17.7	0.088

Air and fuel were monitored by using Brooks Instruments rotameters with tube/float combination R-8M-25-4/8-RV-14 and R-6-25-B/glass, respectively. The pressures at the fuel and

air rotameters were adjusted to 30 psig for all the data. The rotameter settings were predetermined to produce the equivalence ratio desired.

E.2 Flame Speed Computation

The flame speeds are estimated by computing the mass flow rates of the methane air mixture. The volume flow rate in the rotameter is given by

$$V = \frac{[(\rho_f - \rho)/p]^{1/2}}{[(\rho_f - \rho)/p]_{STP}^{1/2}} V_{STP} \quad (E.1)$$

where ρ_f is the float density, ρ is the fluid density and p is the pressure. The subscript STP refers to standard conditions. With $\rho_f = 8.04$ g/cc, the mass flow rate of air is given by

$$m_a = 1.6413 \times 10^{-8} p (9.974 \times 10^4/p - 1)^{1/2} V_{STP} \text{ (kg/s)} \quad (E.2)$$

Similarly, for the fuel, with a float density of 2.54 g/cc, the mass flow rate of the fuel is given by

$$m_f = 1.6125 \times 10^{-8} p (5.708 \times 10^4/p - 1)^{1/2} V_{STP} \text{ (kg/s)} \quad (E.3)$$

where the pressure p at the rotameter is in psia and V_{STP} is in l/min as provided by the manufacturer. The correlation for V_{STP} in terms of the rotameter reading f is given by

$$V_{STP} = b_0 + b_1 f + b_2 f^2 + b_3 f^3 \quad (E.4)$$

Given in Table E.2 are the coefficients for the flow rates for the air and fuel rotameter float combinations.

Table E.2 Coefficients for volume flow rates

	b_0	b_1	b_2	b_3
Air	6.9092	0.6059	-0.4835×10^{-4}	0.7315×10^{-6}
Methane	0.1175	0.0733	0.8186×10^{-4}	-0.4451×10^{-7}

The flame speed u_f is then given by

$$u_f = (m_a + m_f) / \rho_i C_a \quad (E.5)$$

where ρ_i is the density of the inlet mixture and C_a is the total flow area of the mixture inside the porous medium as determined in the model.

Sample calculation

For run #6 in Table E.1, using the correlation in eq. (E.4), one obtains the volumetric flow rates at STP for air and fuel to be 90.3 l/min. and 4.27 l/min. respectively. With the pressure at the rotameters being 30 psig (44.7

psia.), the mass flow rates of air and fuel are computed using eqs. (E.2) and (E.3) to be 3.13×10^{-3} kg/s and 1.1×10^{-4} kg/s respectively. With a flow diameter of 4.8 in., the flow area C_a is computed to be 1.2×10^{-2} m². With the density of mixture $\rho_i = 1.1$ kg/m³, the flame speed is calculated for eq. (E.5) as 0.257 m/s.

E.3 Radiant Output Computations

The radiant output is directly determined from the calibration of the detector described in Appendix D. A straight line fit was obtained to give the following correlation for the detector reading, D_0

$$D_0 = (E + 713.09) / 7.0654 \times 10^5 \quad (E.6)$$

where E is the radiant flux in W/m².

E.4 Uncertainty Analysis

Flame Speed

The uncertainty in flame speed is given by

$$\frac{du_i}{u_i} = \frac{d(m_a + m_f)}{m_a + m_f} + \frac{dp_i}{\rho_i} + \frac{dA}{A} \quad (E.7)$$

$$\text{Also, } \frac{dm_a}{m_a} = \frac{dp}{p} + \frac{dv_{STP}}{V_{STP}} + \frac{1}{2} \left[\frac{adp/p^3}{a/p-1} \right] \quad (E.8)$$

where $a = 9.974 \times 10^4$, the constant appearing in eq. (E.2).

Similarly,

$$\frac{dm_f}{m_f} = \frac{dp}{p} + \frac{dv_{STP}}{V_{STP}} + \frac{1}{2} \left[\frac{bdp/p^3}{b/p-1} \right] \quad (E.9)$$

where $b = 5.708 \times 10^4$, the constant in eq. (E.3).

Also,

$$\frac{d(m_a + m_f)}{m_a + m_f} = \frac{dm_a}{m_a} \left[\frac{m_a}{m_a + m_f} \right] + \frac{dm_f}{m_f} \left[\frac{m_f}{m_a + m_f} \right] \quad (E.10)$$

$$\frac{dp_i}{\rho_i} = \frac{dT_i}{T_i} \quad \text{where } T_i \text{ is the inlet temperature and}$$

$$\frac{dA}{A} = 2 \frac{dD}{D} \quad \text{where } D \text{ is flow diameter.}$$

The quantity dV_{STP}/V_{STP} is given by the following expression after differentiating eq. (E.4) and adding the magnitudes of the differentials to give

$dV_{STP}/V_{STP} = (|a_1| + |2a_2x| + |3a_3x^2|)dx/V_{STP}$. Thus now all the expressions required to estimate the uncertainties in the flame speed are available. With $dp = 0.5$ psig, $dx = 0.5$ rotameter division, $dD = 1/16$ in. and $dT_i = 0.1$ °C the uncertainties in flame speed u_i is estimated to range from 4.13 to 4.34 percent in all the runs.

Radiant Output

From eq. (D.1) the following expression is obtained for the uncertainty in the radiative flux E

$$\frac{dE}{E} = \frac{dA_p}{A_p} + \frac{dA_d}{A_d} + 2\frac{dl}{l} + \frac{dD_0}{D_0} \quad (E.11)$$

With $dA_p = dA_d = 0.01$ in., $dl = 1/16$ in. and $dD_0 = 0.001$ the uncertainties in the radiant outputs range from 9.0 to 9.5 percent.

E.5 Energy Balance

To check for the overall accuracy of the flame speed and radiative output computations, an energy balance was performed on one of the experimental runs (#6 in Table E.1). The temperature of the exhaust gas as it emerges from the burner was measured with a shielded chromel alumel thermocouple to be 1170 K. The product enthalpy at the 1170 K and the reactant enthalpy at 298 K were computed using the following chemical reaction:

where Φ is 0.60. From thermodynamic tables, the reactant enthalpy was computed to be $h_R = -74850$ kJ/kg-mole, whereas the product enthalpy was $h_P = -388807.0$ kJ/kg-mole. The difference in the enthalpy fluxes must be then equal to the sum of the radiative fluxes in the upstream and downstream direction. The difference in enthalpy fluxes is given by

$(h_p - h_R) \rho_1 u_1 / \{ [2 / (137.28 \Phi) + 16] 10^{-3} \}$. The quantity in the curly bracket is the factor for converting the enthalpy on the mixture mass basis. Substituting the appropriate values, the difference is computed to be $1.66 \times 10^5 \text{ W/m}^2$. The radiative output is $1.42 \times 10^5 \text{ W/m}^2$. The heat of combustion on a flux basis is $4.24 \times 10^5 \text{ W/m}^2$. The percentage discrepancy is thus $(1.66 - 1.42) / 4.24 \times 100 = 5.6$ percent. Also, the figures are in the right direction since the radiant flux in the upstream direction is not accounted for and the difference in enthalpy is larger than the radiative output.

END

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