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Influences of material properties on thermal design of impinging flame jets

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Abstract

This paper presented a study of the development of an area-averaged heat flux model for the thermal design of impingement premixed gas-fired round flame jets in terms of the flame and surface conditions. The influencing parameters concerning flame condition include Reynolds number and equivalence ratio of the air–fuel jet, and nozzle-to-surface distance. In terms of the impinging plate surface condition, the two influencing parameters are thermal conductivity and surface emissivity of the plate material, which affect directly the thermal conduction and radiation, respectively. Experimental studies were performed to measure the heat flux and temperature distributions of the impingement plates fabricated with materials of different thermal conductivities and surface emissivities. The premixed air–butane flame was impinging vertically upwards upon the plate's flat bottom surface essentially at laminar flow conditions. In order to examine the influence of thermal conductivity on thermal conduction, there were three materials: brass ($k = 61$ W/mK), bronze ($k = 26$ W/mK) and stainless steel ($k = 14.9$ W/mK), used to fabricate the plate in the present study. Similarly, influence of surface emissivity on thermal radiation was examined by fabricating the brass impingement plate with surface emissivities of 0.1, 0.38 and 0.98, respectively. Based on the response surface methodology, one model was developed by taking into consideration the Reynolds number, equivalence ratio, nozzle-to-surface distance and material's thermal conductivity. Another model considering the Reynolds number, equivalence ratio, nozzle-to-surface distance and material's surface emissivity was also developed. The predicted area-averaged heat flux of an impingement plate has been found to agree closely with that of the experimentally measured data at a 95% confidence level. It is found that the influence on thermal characteristics of an impingement plate will be more significant due to its thermal conductivity rather than surface emissivity.

Keywords: Properties of materials; Area-averaged heat flux; Impingement flame jets; Factorial design; Response surface method

1. Introduction

Impinging premixed gas-fired flame jet heat transfer has been well established as a high-performance technique for heating, cooling and drying processes due to its very high heat and mass transfer for both industrial and domestic applications [1–5]. Investigations of the thermal performance of the impinging flame jets are usually conducted by impinging a premixed flame jet upon a horizontal impingement plate illustrated in Fig. 1. A premixed gas-fired flame emitted from a circular nozzle with a diameter of d, and a velocity of V impinges vertically upwards upon a horizontal impingement surface, which is held at a distance of H from the nozzle rim. After impinging on the plate, the flame and combustion products extend radially outwards along the surface with formation of the two regions: stagnation region and wall-jet region. Due to the complexity of the heat transfer characteristics of the impinging flame jets, which may involve thermal conduction, convection, radiation and thermo-chemical-heat-release (TCHR), reports on relevant previous studies in this area are rather rare [6–10]. Moreover, current design and application of impinging flame jets mainly rely on the practical experience, rather than scientific analysis.

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Nomenclature

A	area for calculating the area-averaged heat flux (m^2)	r	radial distance from the stagnation point (m)
d	diameter of the circular nozzle (m)	V	flame velocity (m/s)
H	distance between the nozzle and the impingement plate (m)	Y	molar fraction
E	numerical constant	*Greek symbols*	
(F/A)	fuel–air ratio	μ	dynamic viscosity (m^2/s)
G	numerical constant	ρ	density (kg/m^3)
M	molecular weight (kg/kmol)	λ	thermal conductivity (W/mK)
m	power index	ϕ	equivalence ratio
n	power index	*Superscript*	
$\dot{q}$	heat flux (W/m^2)	$\sim$	weighted mean
$\bar{\dot{q}}$	area-averaged heat flux (W/m^2)	exit	at the exit position
R	diameter of the area for calculating the area-averaged heat flux (m)	i	mixture component including fuel and gas
		mix	air–fuel mixture
Re	Reynolds number	stoic	stoichiometric condition

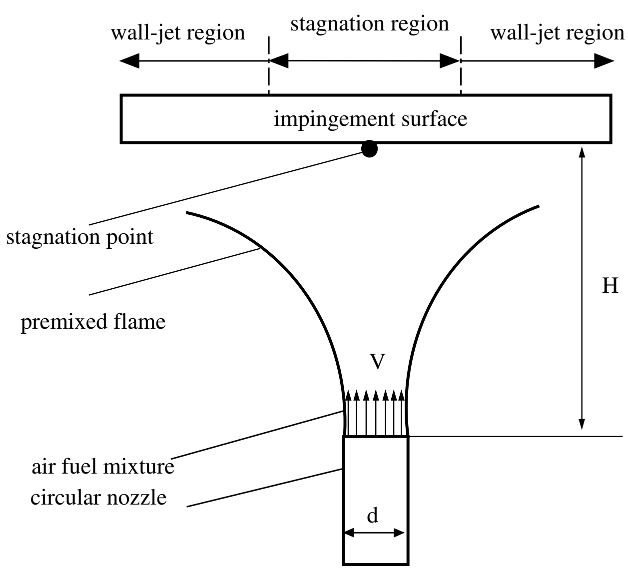

Fig. 1. Impingement flame jet.

According to the studies of Dong et al. [11–13], the premixed butane–air laminar flame jet impingement heat transfer was dependent on Reynolds number and equivalence ratio of the air–fuel jet, and configurations of the air–fuel nozzle and impingement plate. Effects of nozzle shape (circular or slot) and inclination of the impingement plate on heat transfer characteristics of the system under different flame conditions had been studied. The slot flame jet was found to produce more uniform and higher average heat flux than the circular counterpart. In a similar study, Kwok et al. [14] suggested the importance in matching the flame length with the nozzle-to-plate distance to achieve the best heat transfer performance. Baukal and Gebhart [15] found that the heating process of premixed impingement flame jets usually comprised multiple heat transfer mechanisms. In their studies, polished, untreated and blackened surfaces were used to investigate the effect of surface emissivity on thermal radiation. The largest difference in heat flux between the polished and blackened surfaces was only 9.8%. It was therefore suggested that both non-luminous radiation and thermo-chemical-heat-release (TCHR) were small fractions of the total heat flux.

Although some experimental studies have been performed, very few prediction models for evaluation of the thermal performance of impinging premixed flame jets are developed. The aim of the present study is therefore to develop a prediction model for the area-averaged heat flux, with the aid of the factorial design method and response surface model (RSM) [16–18]. By using the response surface methodology and 3-level factorial design of experiments, the prediction models for the evaluations of the effects of both flame and impingement surface conditions have been developed with a 95% confidence level.

2. Methodology and procedures

The area-averaged heat flux of an impingement plate impinged upon by a flame jet is influenced by several independent variables (hereafter defined as factors), and the design methodology of experimental work to study these factors is normally used to analyze their reciprocal influences. In the present investigation, a full factorial design of experiments has been employed: an output variable is correlated to the input factors expected to have influence. Therefore it is possible to identify those factors deserve particular attention. For example, when further analysis on a particular factor deems necessary then more experimental measurements are needed to perform. On the contrary, a factor having a negligible effect will even not be considered in the subsequent analysis. Most essentially, the combined effects among the factors can also be assessed via this approach.

In the full factorial design method, it is necessary to determine the range of variation (the experimental domain) of each of the chosen input factors. In the present experimental investigation, there are four factors chosen and each of them consisting of three levels (low, intermediate and high), therefore, a total of $3^4 = 81$ experiments are required to perform in obtaining all the possible effects and their interactions. The influences of the input factors are identified as main effects $(A, B\ldots)$ and interaction effects $(AB, AC\ldots)$.

2.1. Area-averaged heat flux models

The flame jet's Reynolds number was evaluated based on the air–fuel mixture as

$$Re = \frac{\mu_{\text{exit}} d \rho_{\text{mix}}}{\mu_{\text{mix}}} \tag{1}$$

where μ_{mix} was calculated according to Ikoku [19] as

$$\mu_{\text{mix}} = \frac{\sum(\mu_i Y_i \sqrt{M_i})}{\sum(Y_i \sqrt{M_i})} \tag{2}$$

The equivalence ratio, ϕ, is commonly used to indicate quantitatively whether a fuel–oxidizer mixture is rich, lean, or stoichiometric, and is defined as

$$\phi = \frac{(F/A)}{(F/A)_{\text{stoic}}} \tag{3}$$

To evaluate the thermal performance of the experimental system, the equation for calculating the area-averaged heat flux is provided as shown below:

$$\bar{\dot{q}} = \frac{\int\int_A \dot{q}\, \mathrm{d}A}{A} \tag{4}$$

where $\dot{q}$ is the local heat flux on the impingement plate, which is measured experimentally. The polar coordinates are applied, because a circular nozzle has been used and the local heat flux is essentially a function of the radius from the stagnation point. Calculations of the area-averaged heat flux by using Eq. (4) can therefore be simplified as

$$\bar{\dot{q}} = \frac{2\int_0^R \dot{q} r\, \mathrm{d}r}{R^2} \tag{5}$$

The curve of $\dot{q}$ is first fitted by a polynomial function in terms of r, then the area-averaged heat flux can be calculated according to Eq. (5). To investigate the effects of different thermal conductivities and surface emissivities of the impingement plate, and their interactions with Reynolds number, equivalence ratio and nozzle-to-plate distance, the proposed relationships between the area-averaged heat flux and the independent variables under different flame conditions can be given as

$$\bar{\dot{q}} = ERe^{m1} \phi^{m2} (H/d)^{m3} \lambda^{m4} \tag{6}$$

$$\bar{\dot{q}} = GRe^{n1} \phi^{n2} (H/d)^{n3} \varepsilon^{n4} \tag{7}$$

where E and G are numerical constants, and m (1–4) and n (1–4) are power index to be determined by using the factorial design method.

Eq. (6) represents a relationship between the area-averaged heat flux and the Reynolds number, equivalence ratio, nozzle-to-plate distance and thermal conductivity; whereas Eq. (7) provides similar relationship except the surface emissivity is considered instead of thermal conductivity. To facilitate the determination of all the numerical constants and power index, Eqs. (6) and (7) are linearized by applying the logarithmic transformations:

$$\ln \bar{\dot{q}} = \ln E + m1 \ln Re + m2 \ln \phi + m3 \ln(H/d) + m4 \ln \lambda \tag{8}$$

$$\ln \bar{\dot{q}} = \ln G + n1 \ln Re + n2 \ln \phi + n3 \ln(H/d) + n4 \ln \varepsilon \tag{9}$$

Table 1
Properties of impingement plates with different thermal conductivities (Group 1)

Materials	Brass	Bronze	Stainless steel
Thermal conductivity (W/mK)	61	26	14.9
Surface emissivity	0.38	0.40	0.55
Surface roughness (μm)	1.03	3.33	5.18

Table 2
Properties of brass impingement plates with different surface emissivities (Group 2)

Materials	Brass	Bronze	Stainless steel
Thermal conductivity (W/mK)	61	61	61
Surface emissivity	0.10	0.38	0.98
Surface roughness (μm)	1.03	1.03	1.03

Table 3
Three-levels of all independent variables

Levels	Low	Intermediate	High
Re	500	1000	1500
ϕ	0.9	1.0	1.1
H/d	3	5	7
λ (W/mK)	14.9	26	61
ε	0.10	0.38	0.98

2.2. Factorial design of experiments

In order to develop the models for the investigation of the influence of material thermal conductivity and surface emissivity on the heat transfer to an impingement plate under different flame conditions, two groups of impingement plates have been used, which are summarized in Tables 1 and 2, respectively:

In Group 1, surface emissivities of the three plates are rather similar (0.38, 0.4 and 0.55), and therefore variation of thermal radiation can be assumed negligible. In addition, the three impingement plate surfaces have very small surface roughness (1.03, 3.33 and 5.18 μm) and they can be considered as very smooth, such that effect of surface roughness on convection enhancement can also be assumed negligible as suggested by Kang et al. [20]. In Group 2, only surface emissivity of the three brass impingement plates has been varied and the other surface properties are maintained constant.

Table 3 shows the three levels of the other independent variables. Since there are four independent variables considered in each of these two models, as a result, there are 81 sets of experimental results needed in developing each model.

3. Results and discussions

3.1. Prediction model for influence of thermal conductivity

According to the results obtained from the Group 1 experiments, with the factorial design in response surface methodology [21], Eq. (8) can then be determined as shown:

$$\ln \bar{\dot{q}} = -1.48 + 0.582 \ln Re + 0.453 \ln \phi - 0.248 \ln(H/d) + 0.797 \ln \lambda \tag{10}$$

Eq. (10) can be transformed into the following form:

$$\bar{\dot{q}} = 0.228 Re^{0.582} \phi^{0.453} (H/d)^{-0.248} \lambda^{0.797} \begin{cases} 500 \leqslant Re \leqslant 1500 \\ 0.9 \leqslant \phi \leqslant 1.1 \\ 3 \leqslant H/d \leqslant 7 \\ 10 \leqslant \lambda \leqslant 110\ \text{W/mK} \end{cases} \tag{11}$$

The R^2 and adjusted R^2 values for this design model are 0.8987 and 0.9052, respectively, which indicate a relatively high reliability for this developed model. The maximum deviation of the prediction from the measured values is 5.4%.

Table 4a shows a test for significance of the individual variables for the design model as shown in Eq. (10), in which A, B, C, D represent (H/d), ϕ, Re and λ, respectively. It can be found that although all four independent variables affect the area-averaged heat flux, the Reynolds number and thermal conductivity show more significant effects than the other two variables. Tests have been conducted with the quadratic model to determine the significance of the combination of individual variables, as shown in Table 4b. It can been found that since variables C and D have higher influences on the area-averaged heat flux, the squares of C and D and the combination of C and D are also significant. However, those of A and B are less insignificant reflecting their less importance.

It is known that Reynolds number indicating the amounts of fuel and air taking part in the combustion. Larger Reynolds number indicates that more fuel is burnt, providing higher heat flux when calorific values of the fuel and equivalence ratio of the air–fuel mixture are assumed to maintain. Equivalence ratio is referred to the proportion between fuel and air in combustion, however, the changing of equivalence ratio is not found to influence the thermal performance of the impingement flame jets as much as the Reynolds number. H/d ratio affects directly the relative position between the flame and impingement plate surface. According to Hargrave et al. [9], a maximum heat transfer can be achieved by having a large concentration of active species existing at the tip of the flame's luminous inner reaction zone (the flame front), which enhances the convection by the diffusion and exothermic recombination at the impingement surface. Such activities are extremely significant when the separation between the flame front and impingement surface is not excessively large (at $H/d = 3$ for brass plate and $5 \leqslant H/d \leqslant 6$ for stainless steel plate). When this separation becomes excessively large (at $H/d > 4$ for brass plate and $H/d > 6$ for stainless steel plate), effect of such activities on the heat flux received by the impingement plate becomes less important [22]. As a result, H/d ratio can affect the heat transfer characteristics only in a very small domain for impingement flame jets. A lower thermal conductivity leads to a larger suppression of the heat flux at its stagnation point [22], the thermal performance of impingement flame jet system can be significantly affected by the thermal conductivity of the plate materials.

3.2. Prediction model for influence of surface emissivities

With reference to the results obtained from the Group 2 experiments, similarly, all the unknowns of Eq. (9) can be determined, such that:

$$\ln \bar{\bar{q}} = 1.22 + 0.642 \ln Re + 0.341 \ln \phi - 0.234 \ln(H/d) + 0.025 \ln \varepsilon \tag{12}$$

Eq. (12) can further be developed into the following form:

$$\bar{\bar{q}} = 3.387 Re^{0.642} \phi^{0.314} (H/d)^{-0.234} \varepsilon^{0.025} \begin{cases} 500 \leqslant Re \leqslant 1500 \\ 0.9 \leqslant \phi \leqslant 1.1 \\ 3 \leqslant H/d \leqslant 7 \\ 0.1 \leqslant \varepsilon \leqslant 0.98 \end{cases} \tag{13}$$

The R^2 and adjusted R^2 values for the developed model are 0.9684 and 0.9669, respectively, indicating a high reliability. The maximum deviation between the prediction and actual values is 3.4%.

Table 5a shows the tests of significance for the individual variables as considered in Eq. (12), in which A, B, C

Table 4a
Test for significance of individual variables

Sources	Sum of squares	Degree of freedom	Mean squares	F value	Prob > F	Significance
A	0.14	1	0.14	1.60	0.209	Less
B	0.11	1	0.11	1.31	0.256	Less
C	5.55	1	5.67	66.35	<0.0001	More
D	22.16	1	17.31	202.52	<0.0001	More

Table 4b
Test for significance of combination of individual variables

Sources	Sum of squares	Degree of freedom	Mean squares	F value	Prob > F
A^2	0.0035236	1	0.0035236	1.091226838	0.2997
B^2	0.0001281	1	0.0001281	0.039671808	0.8427
C^2	0.071537	1	0.071537	22.15406332	< 0.0001
D^2	1.6516336	1	1.6516336	511.4889795	< 0.0001
AB	0.000593	1	0.000593	0.18364097	0.6695
AC	0.0061628	1	0.0061628	1.90854519	0.1714
AD	0.0012371	1	0.0012371	0.383125634	0.5379
BC	0.0224434	1	0.0224434	6.950435136	0.0103
BD	0.0004496	1	0.0004496	0.139242788	0.7101
CD	0.1944381	1	0.1944381	60.21489852	< 0.0001

Table 5a
Test for significance of individual variables

Sources	Sum of squares	Degree of freedom	Mean squares	F value	Prob > F	Significance
A	0.12	1	0.12	43.20	<0.0001	More
B	0.063	1	0.063	22.33	<0.0001	More
C	6.9	1	6.9	2432.28	<0.0001	More
D	0.044	1	0.044	15.62	0.0002	Less

Table 5b
Test for significance of combination of the variables

Sources	Sum of squares	Degree of freedom	Mean squares	F value	Prob > F
A^2	0.004738	1	0.004738	2.701687843	0.1046
B^2	0.003048	1	0.003048	1.737818379	0.1916
C^2	0.019146	1	0.019146	10.91598607	0.0015
D^2	0.000206	1	0.000206	0.117205066	0.7331
AB	4.09E−05	1	4.09E−05	0.023322879	0.0295
AC	0.003056	1	0.003056	1.742519943	0.1910
AD	0.008655	1	0.008655	4.934854525	0.8790
BC	0.059916	1	0.059916	34.16173352	<0.0001
BD	0.003891	1	0.003891	2.21843664	0.1407
CD	0.002499	1	0.002499	1.424687882	0.2366

and D represent (H/d), ϕ, Re and ε, respectively. It is found that surface emissivity has rather low influence on the heat transfer performance when it is compared to the other variables, which is well agreed with the study of Baukal and Gebhart [15]. The quadratic model tests have also been done to determine the significance of the combined variables as shown in Table 5b. It can been found that the combination of B and C (ϕ and Re) has very significant effect on the area-averaged heat flux for such model, effects of both surface emissivity and H/d ratio are much lower than those of the other two variables.

Impingement flame jet system is used in both domestic and industrial applications, such as stove, cooktop burner and small scale furnace. The prediction models developed in the present study provide designers of the impingement flame jet systems, which can be used to determine the area-averaged heat flux on an impingement surface promptly and more exactly.

4. Conclusions

Factorial design for the experimental work conducting with the impinging gas-fired premixed round flame jets has been successfully employed to develop the area-averaged heat flux models in terms of flame and surface conditions. The following conclusions have been drawn:

1. First-order model equations for the predictions of area-averaged heat flux of an impingement plate impinged upon by the flame jet have been developed using the response surface methodology with 3-levels assigning to each factor. The independent variables considered in the model are Re, ϕ, (H/d), thermal conductivity and surface emissivity. The prediction and measured values agree well with each other indicating that the developed area-averaged heat flux prediction models are reliable. In fact, a 95% confidence level has been obtained for both models.
2. By carrying out a variance analysis for the developed models shows that the effect on thermal performance of thermal conductivity under different flame conditions is as statistically significant as that of Reynolds number. The influence of combination of different factors can also be predicted with the variance analysis. The effect of surface emissivity under different flame conditions when compared to those of the other independent variables seems to be less significant. In addition, assessing the effects of different factor combinations by using the developed models shows that the influence of ϕ, Re and thermal conductivity are more significant than that of the other two other factors, H/d ratio and surface emissivity.
3. Two equations are developed in this study, concerning the effects of material thermal conductivity and surface emissivity, respectively:

$$\bar{q} = 0.228 Re^{0.582} \phi^{0.453} (H/d)^{-0.248} \lambda^{0.797} \begin{cases} 500 \leqslant Re \leqslant 1500 \\ 0.9 \leqslant \phi \leqslant 1.1 \\ 3 \leqslant H/d \leqslant 7 \\ 10 \leqslant \lambda \leqslant 110\ \mathrm{W/mK} \end{cases}$$

$$\bar{q} = 3.387 Re^{0.642} \phi^{0.314} (H/d)^{-0.234} \varepsilon^{0.025} \begin{cases} 500 \leqslant Re \leqslant 1500 \\ 0.9 \leqslant \phi \leqslant 1.1 \\ 3 \leqslant H/d \leqslant 7 \\ 0.1 \leqslant \varepsilon \leqslant 0.98 \end{cases}$$

The developed equations provide a quick but reliable method to predict accurately the area-averaged heat flux of a horizontal plate impinging upon by a premixed butane–

air flame jet. The present work has facilitated the thermal design of domestic and small-scale industrial combustion systems utilizing impingement flame jets.

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