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Heat transfer distribution for impinging methane–air premixed flame jets

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HIGHLIGHTS

- IHCP technique to obtain local heat flux distribution using thermal camera.
- Adiabatic wall temperature and heat transfer coefficient are given.
- Circular, square and rectangular burners are used.
- Axis switching is observed in non-circular burners.
- Correlations for local Nusselt number and effectiveness are given.

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ABSTRACT

Heat transfer by flame jet impingement is extensively used in industrial and domestic heating applications. The present experimental study proposes the application of an inverse heat conduction (IHCP) technique to obtain the heat flux distribution for methane–air premixed flame jet impinging on a flat plate. The heat flux distribution is studied for burner tubes of circular shape ($d = 10$ mm and 8.75 mm), square shape (width = 10 mm and 7.65 mm) and rectangular shape (19 mm × 9 mm). Methane–air premixed flame jet of Reynolds number varying from 600 to 2200 and an equivalence ratio of 1 is considered. The nozzle to burner tip distance is varied from 2 to 6. Axis switching is observed for non-circular shaped burner flame jets. Correlations for local Nusselt number and effectiveness distribution are proposed for circular and square burners by direct measurement of the adiabatic wall temperature. The heat transfer coefficients and adiabatic wall temperatures are validated with the experimental heat flux data available in the literature. The non-dimensional flame premixed cone height (ratio of flame premixed cone height to the distance of the burner tip from the impingement wall) alone governs the Nusselt number and effectiveness.

1. Introduction

Heat transfer by flame jet impingement is extensively used in several industrial and domestic applications like melting of metal billets in a closed heating furnace, glass processing, domestic gas geysers and others. The phenomenon of flame jet impingement heating is dependent on four different mechanisms of heat transfer-forced convection, radiation, thermochemical heat release (TCHR) and condensation of water vapor in the burnt gas. Amongst the four mechanisms of heat transfer, forced convection is the most dominant and accounts for nearly 90% of the heat transferred in flames with air as oxidizer. Reviews by Viskanta [1], Baukal and Gebhart [2,3] and Chander and Ray [4] give substantial information of the flame jet impingement studies. These studies are mostly experimental in nature. Recently analytical expressions are reported by Remie et al. [5] for two-dimensional and axisymmetric cases of impinging flame jets. These analytical expressions are based on isothermal plug flow concept for the flame jet valid for a radial distance, $r < R$ and nozzle-tip distance, $z < 2R$. For two-dimensional case with non-viscous assumption, they have reported a simple expression given by Eq. (1). Remie et al. [5] are further able to arrive at an expression (Eq. (2)) for viscous flow case by adjusting the analytical expression result with numerical simulation result carried out in a commercial CFD package FLUENT.

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Nomenclature

A	area (m^2)
A/F	air to fuel ratio
C_p	specific heat (J/kg K)
d	hydraulic diameter (m)
I	current (A)
k	thermal conductivity (W/m K)
L	flame inner cone length (m)
M	molecular weight
Nu	Nusselt number
Pe	Peclet number
Pr	Prandtl number
q''	heat flux (W/m^2 K)
Q	volumetric heat (W/m^3)
r	arbitrary radius (m)
R	maximum radius of the burner tube (m)
Re	Reynolds number
Su	burning velocity (m/s)
t	time (s)
T	temperature (K)
u_m	average velocity of fuel–air mixture (m/s)
V	voltage (V)
w	square burner width (m)
x, y, r, z	coordinate directions
X	mole fraction
Y	mass fraction
z	burner tip to target plate distance (m)
Z	quartz plate thickness/depth (m)

Greek symbols

α	thermal diffusivity (m/s^2)
β	radial velocity gradient (1/s)
μ	absolute viscosity (Pa s)
ρ	density (kg/m^3)
ϕ	equivalence ratio
η	effectiveness

Subscripts/superscripts

aw	adiabatic wall
conv	convection
e	edge of boundary layer
f	flame
FJ	flame jet
i ,init	initial
j	component of the mixture
m	mixture
NC	natural convection
rad	radiation
0	stagnation point
w	wall

$$q''_0 = \frac{k\left(T_f - T_w\right)}{z}\sqrt{Pe} \tag{1}$$

where, $Pe = v_e \rho z Cp/k$

$$q''_0 = \frac{k\left(T_f - T_w\right)}{z}\sqrt{Pe}\cdot\exp\left(-0.28Pr^{0.4}\right) \tag{2}$$

The fluid properties for Pe are evaluated at an average temperature (maximum temperature within the boundary layer and target plate surface temperature). The fluid velocity (v_e) is the velocity at the edge of the boundary layer. In the subsequent work by Remie et al. [6], they matched the results obtained by analytical expression with the numerical solution from FLUENT. They found that the ratio of the two results decay in an exponential fashion for region away from the stagnation region ($r > R$) and therefore corrected their analytical expression by an exponential multiplication factor. However, in order to use the Eqs. (1) and (2), the flame temperature needs to be measured near the edge of the boundary layer. The measurement of flame at the edge of the boundary layer is difficult as this requires proper location of thermocouple. Furthermore, this intrusion of thermocouple would disturb the boundary layer.

Studies on numerical investigation of this problem are reported by Conolly and Davies [7], Som et al. [8], Chander and Ray [9] and Remie et al. [6]. Different burner geometries (single, multiple and annular) are used along with inverse diffusion flame (IDF) burners and swirl induced burners [10–18]). Almost all these reported studies are on the estimation of convective heat transfer characteristics while studies on radiation heat transfer characteristics [19–21] and emissions [21–24] are limited. Following conclusions can be drawn from the reported literature:

i) The heat transfer distribution is strongly dependent on the nozzle-plate spacing [1–24], burner shape [17], Reynolds number [1–25], equivalence ratio, oxygen enhancement [19], jet incidence angle [26], impingement plate material [27] and inter-jet spacing [14,16].
ii) The effect of surface characteristics of the impingement plate on the heat transfer characteristic is less significant [21].
iii) Thermochemical heat release is a dominant mechanism of heat transfer contributing by more than 40% for oxygenated methane flames [25]. This may be neglected for methane–air flames [25].

The heat flux reported in the literature is measured by a ring calorimeter [2,3] and heat flux sensors [4,9–18]. The size of the heat flux sensor is around 4–6 mm in diameter. Hence, the measured heat flux is averaged over this area. Furthermore, this heat flux sensor is to be traversed to different locations to measure the heat flux at different locations. Hence, the heat flux distribution reported in the literature lacks sufficient resolution, except the approach using inverse heat conduction (IHCP) method of Nortershaur and Millan [28] and Loubat et al. [29]. High spatial resolution enables one to understand important physical phenomenon like the axis-switching (Gutmark et al. [30] and Lou [31]). Use of inverse heat conduction method using liquid crystals are extensively used in the estimation of heat transfer coefficient for impinging air jets (Baughn [32], Talib et al. [33], Sagheby and Kowsary [34]) and film cooling problem (Ai et al. [35], Chen et al. [36]). In the present work, an IHCP method using analytical solution for semi-infinite medium is used to get high resolution heat flux using a thermal infrared camera.

Nusselt number distribution is of engineering importance for the flame jet impingement heat transfer process. Correlations for Nusselt number distributions are reported in the literature based on analytical and semi-analytical studies. A collection of these correlations is available in the review papers of Viskanta [1] and Baukal and Gebhart [2,3]. Van der Meer [37] compared the Nusselt number distribution of isothermal non-reacting jets with that of the flame jets. He showed that if the properties of the burnt gases are taken at the temperature corresponding to the average enthalpy