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Heat transfer distribution for three interacting methane–air premixed impinging flame jets

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Multiple impinging flame jets are widely used in industrial heating (Ex. ladle preheating) and melting applications (Ex. glass forming). In the present method, high resolution heat flux distribution is obtained by applying the inverse heat conduction technique using thermal infrared camera. The high resolution thermal imaging enables to capture the steep gradients of spatial heat flux distribution. Nusselt number and effectiveness distributions are obtained by analytical–numerical method of estimation of the adiabatic wall temperature. Three circular tube burners arranged in a staggered pattern with inter tube spacing (S/d) of 2 to 6 is considered. The ratio of distance from tube burner tip to the impingement plate (z/d) is varied from 2 to 6 while the Reynolds number is varied from 400 to 1000. For the smallest $S/d = 2$, the interaction amongst the flames is significant and leads to non-circular hot spot distribution for heat flux. The average Nusselt number and average effectiveness are higher for higher Re for all z/d unless the inner premixed cone touches the impingement plate. The average Nusselt number and average effectiveness are marginally higher for smaller S/d for the same z/d and Re. The coefficient of variance increases with the increase in S/d.

1. Introduction

Heat transfer by flame jet impingement is extensively used in many industrial and domestic applications like melting of scrap metals, glass processing, ladle preheating, gas geyser and others. Extensive information on the flame jet impingement heat transfer characteristics is reported in the literature. Reviews by Viskanta [1], Baukal and Gebhart [2,3] and Chander and Ray [4] give substantial information of the flame jet impingement studies reported in literature. These reported studies are mostly experimental in nature. However, some analytical expressions are also reported [5,6] for two-dimensional and axisymmetric cases of impinging flame jets. Studies on numerical investigation are reported in the work of Conolly and Davies [7], Som et al. [8], Chander and Ray [9] and Remie et al. [6]. Different burner geometries (employing single and multiple jets) have been used, the inverse diffusion flame (IDF) burners have been employed by Dong et al. [10–14] and burners with induced swirl by Huang et al. [15], Chander and Ray [16,17] and Singh et al. [18]. Following conclusions are drawn from the literature.

(i) The heat transfer distribution depends strongly on the nozzle-plate spacing, Reynolds number [1–24], shape of the burner, equivalence ratio, oxygen enhancement [19], jet incidence angle [25] and inter-jet spacing [14,16].
(ii) The surface characteristic of the impingement plate does not have significant effect on the impingement heat transfer characteristics [21].
(iii) Thermochemical heat release contributes by more than 50% for highly oxygenated methane–air flames [23], this is negligible for methane–air flames [23].

A general observation in most of the reported work in literature is that the peak heat flux and surface temperature location is not always at the stagnation point even when the inner premixed cone has not touched the impingement plate. The tube burner when compared with the orifice and nozzle has the maximum heat flux and surface temperature near the stagnation region, for the nozzle and orifice burners it is shifted away from the stagnation point [17]. The reason attributed to this is the velocity profile of the unburnt mixture exiting from these burners is different. A slightly rich fuel mixture of $\phi = 1.1$ yields maximum heat transfer characteristics at the stagnation point [17]. At low separation distances and near stagnation point, the platinum-coated surface has a distinctively higher heat flux (12% more) than the alumina-coated surface, the catalytic *TCHR* is reported to be insignificant mechanism for the flame

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Nomenclature

Symbol	Meaning
A	area (m^2)
Cp	specific heat (J/kg K)
COV	coefficient of variance
d	tube inside diameter (m)
E, F	correction factors for analytical–numerical method
k	thermal conductivity (W/m K)
L_f	flame cone height (m)
Nu	Nusselt number
q''	heat flux (W/m^2)
r	arbitrary radius (m)
Re	Reynolds number
t	time (s)
T	temperature (K)
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)
η	effectiveness
δ_t	thermal boundary layer thickness (m)
μ	absolute viscosity (Pa s)
ρ	density (kg/m^3)
σ	standard deviation
ϕ	equivalence ratio
η	effectiveness
θ	angle (°)

Subscripts/superscripts

aw	adiabatic wall
e	edge of boundary layer
f	flame
i, $init$	initial
j	component of the mixture
m	mixture
p	plate
w	wall

impingement conditions [19]. Flame impinging normal to plate has the highest efficiency and lowest heating time [25]. For multiple jets with equilateral triangle arrangement, the heat flux at stagnation point decreases with increase in the inter-jet spacing and the peak heat flux shifts farther away from the stagnation point [16]. For pure methane–oxygen flames, the equilibrium thermochemical heat release parameter (TCHR) is as high as 1.6 to 3.0 for different surface temperatures which implies that more than 50% of the heat transfer is by TCHR [23].

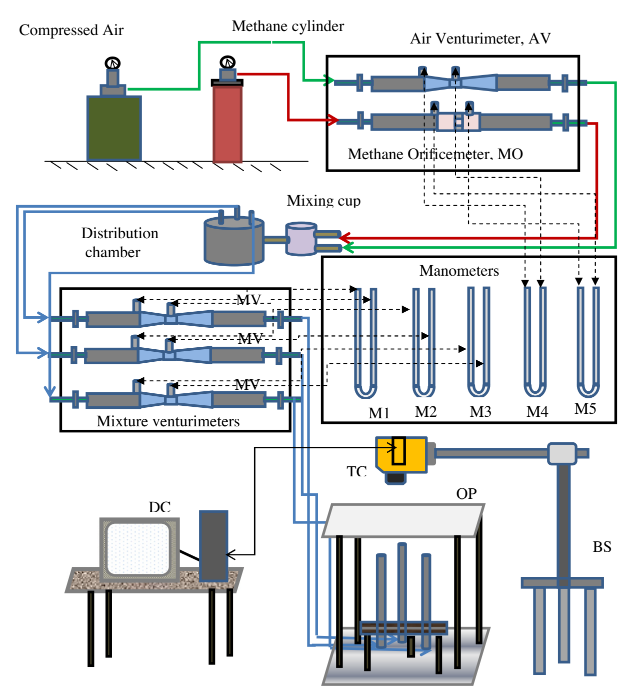
The heat flux reported in literature is mostly either measured by ring calorimeter [2,3] or heat flux sensors of size 4 to 6 mm in diameter [4,9,10–18], the measured heat flux is averaged over this corresponding area. Furthermore, traversing this heat flux sensor to different locations implies need to collect a large number of data. Hence, the heat flux distribution reported in the literature lacks sufficient resolution, except the approaches using inverse heat conduction (IHCP) method of Nortershauser and Millan [26] and Loubat et al. [27] and use of Phosphor Thermometry by Remie et al. [6] and Atakan et al. [28]. In the present study, an IHCP method using analytical solution for semi-infinite medium is used to obtain high resolution heat flux using a thermal infrared camera [28]. The Nusselt number distribution is of engineering importance for the flame jet impingement heat transfer process. The heat flux and Nusselt number are defined as given in Eqs. (1) and (2):

$$q'' = h(T_{aw} - T_w) = -k_f (dT/dz)_{z=0} = -k_f \left(\frac{T_e - T_w}{\delta_t} \right) \quad (1)$$

$$Nu = \frac{hd}{k_f} \quad (2)$$

The steady state heat flux in a practical application has sole dependence on the impingement side wall temperature (T_w) for a specified jet Reynolds number, equivalence ratio and nozzle-plate spacing. Furthermore, this wall temperature is decided by the material thermal properties and this causes difficulty in using the heat flux data available in literature. The heat transfer coefficient, h given in Eq. (1), is of engineering interest. The heat transfer coefficient can be evaluated from the knowledge of adiabatic wall temperature (T_{aw}). The adiabatic wall temperature can be estimated by the analytical–numerical method proposed in our previous work [29].

In the present work, the heat flux is estimated by analytical IHCP method with high spatial resolution and this heat flux data is presented in non-dimensional form as Nusselt number. Three circular tube burners of diameter (d) = 7.5 mm with varying spacing (S/d) of 2 to 6 arranged in staggered pattern are used. The distance between the nozzle tip to the impingement plate (z/d) is varied from 2 to 6 and Reynolds number is varied from 400 to 1000. The objectives of the present work are as follows.



TC	Thermal Camera	M	Manometer
QP	Quartz plate	BS	Boom stand
MV	Mixture venturimeter	DC	Desktop computer

Fig. 1. Schematic of the experimental setup.