Experimental investigation on heat transfer characteristics of partially premixed round methane-air impinging flame jet using Mach-Zehnder interferometry

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ABSTRACT

In this study, premixed and partially premixed laminar co-flow impinging flames are studied experimentally. Methane and air were used as the fuel and oxidizer in a co-annular burner with axisymmetric configuration. For the premixed setup, the mixture of fuel-air flowed from the inner tube, while for partial premixing, the secondary air was added from the region between the inner and the outer tubes. The intention is to investigate the temperature field of impinging flame using Mach-Zehnder interferometry and finding the Nusselt number and heat transferred to the impingement surface using this temperature field. The Mach-Zehnder method was used to find temperature field of a transparent fluid by the refraction index. The effects of operational factors such as Reynolds number, equivalence ratio, the distance of the impingement surface to the burner, burner diameter, and secondary air ratio have been studied. For validation of the experimental data processing, the temperature at different locations was measured by thermocouples, which confirmed the interferometry method. The results indicate that the maximum heat flux was transferred to the impingement surface while the impingement surface was near to the inner reaction zone. Partially premixed impinging flame has better heat transfer characteristics and 4.5–6% higher maximum flame temperature than premixed flame. The average total heat transfer enhancement by partial premixing is 10.2–13.5%.

1. Introduction

Partially premixed flames (PPF) are formed when a fuel stream is being mixed with a sub-stoichiometric amount of primary air, followed by adding excess air/oxidizer to the first mixture to achieve complete combustion. Partially premixed flames have several advantages in comparison with non-premixed (NPF) and premixed flames (PF); especially their high level of stability [1] can be mentioned which makes them a suitable option for various applications, e.g., in Bunsen burners, furnaces, staged combustion systems, and gas-turbine combustor. Also, PPFs have lower pollutant levels and provide a safer operation [2]. Because of their widespread applications, investigation of PPFs has been the subject of many classic studies within the field of combustion and flames. In a numerical and mathematical investigation, Claramunt et al. studied the methane-air laminar flame in a co-flow partially premixed burner [3]. They analyzed the effect of partial premixing level and the sufficiency of different mathematical sub-models on the modeling of flame with particular emphasis on the pollutant formation. Mishra et al. [4] measured the centerline concentration of some species by gas chromatography (GC) in the laminar partial premixed flame. They compared the flame structure and the height of the inner reaction zone for two kinds of fuels including methane and propane. According to their study, a double flame structure can be observed in PPFs when the equivalence ratio is kept within an optimum range; where this range depends on the fuel type and geometry of the burner. In an experimental and simulation study which has been conducted to investigate partially premixed methane-air counterflow flame by Luo et al. [5], flame structure and emissions of premixed and partially premixed flames were compared. According to their results, PF has a single flame structure, and the temperature is relatively high in the entire combustion zone, while the PPF has distinct double flame structure and two temperature peaks can be clearly detected in the premixed reaction zone and the non-premixed zone.

Impinging flame has been extensively used for heating or drying of materials in industrial and small scale processes in which most of the heat transfer to the target object is by convection rather than radiation [6]. It has several practical applications because of the greater heat flux which can be reached that consequently makes the process quick and
Abbreviations/acronyms

GC  Gas chromatography  
MZI  Mach-Zehnder interferometry  
NPF  Non-premixed flame  
PF  Premixed flame  
PL  Optical path length  
PPF  Partially premixed flame

List of Symbols

C  The Gladstone-Dale constant  
d  Inner diameter of the inner burner  
D  Inner diameter of the outer burner  
H  Distance between the burner and impingement surface (mm)  
k  Thermal conductivity  
W  Molecular weight (kg/mole)  
m  Mass flow rate  
n  Refractive index  
P  Pressure  
q″  Heat transfer from the flame to the surface (kW/m²)  
R  Gas constant (J/kg.K)

Re_in  Reynolds number of inner burner  
T  Temperature (K)  
V  Velocity of premixed gas (m/s)  
x  Mole fraction  
(X,Y,Z)  Cartesian axes coordinate

Greek Symbols

ε  Fringe displacement  
λ  Wavelength of the laser beam (m)  
μ  Viscosity (kg/m.s)  
ρ  Density (kg/m³)  
φ  Equivalence ratio  
φₒ  Overall equivalence ratio

Subscripts

exit  At exit position  
i  Mixture component comprising fuel and air  
mix  Mixture of fuel and air  
ref  Reference fringe in the undisturbed region  
w  Wall

Fig. 1. Schematic of the a) co-annular burner, and b) impingement surface.
less costly and also the possibility of localized heating method that results in accurate heating control over a given area [7,8]. The main disadvantage of impinging flames is the non-uniform heat distribution on the surface [9]. To date, some studies have explored the thermal characteristics and temperature field of impinging premixed flames [10,11]. Data of the temperature field of the flame is crucial to determine the heat transfer rate to the receiving medium, i.e., the impingement surface. Thus, many experimental and numerical studies have focused on identifying and evaluating the heat transfer rate [12,13]. Operating parameters such as Reynolds number, equivalence ratio, and the distance between the impingement surface and the flame jet are recognized as the main influencing factors on the thermal performance of the impinging flame jet and the amount of heat transfer to the surface [9,14–16]. Recently, novel methods are proposed for determination of spatially varying heat transfer coefficient in flat plates [17]. Consequently, heat transfer enhancement in such physical applications with operational and geometrical modifications have been investigated extensively [18–20]. Thus far, previous studies have attempted to evaluate impinging flame heat transfer on the inclined impingement surface. Decreasing the oblique angle shifted the maximum temperature and heat flux zone toward the major flow region or downhill part of the plate [16,21,22]. According to a study of cylindrical impingement surface and its curvature effect, the cylindrical targets have higher stagnation region heat flux in comparison with the flat plates, and a reverse trend in the wall jet region can be observed [15]. Experimental studies on multiple-impinging flames with three circular burners arranged in a staggered pattern show the higher Nusselt number for smaller inter-tube spacing [23]. Different fuels, including butane [24], methane [14], biogas-hydrogen [25], and LPG [26] are used in the impinging flame studies.

The traditional method of finding the flame temperature profile is to insert thermocouples at different locations and construct the entire temperature field by the measured data. Thermocouple results are, however, affected by radiation, convection, and conduction. Moreover, they disturb the flow field due to their intrusiveness [27]. On the other side, optical methods, including interferometry [28], are generally preferred as they are fast, accurate, sensitive, full-field, and non-intrusive. All the interferometry methods such as Mach-Zehnder interferometry (MZI) [29,30], Talbot interferometry [31], lateral shearing interferometry [18], and holographic interferometry [32] are based on measuring the change in the refractive index of the flame. There is an inverse relationship between temperature and relative refractive index expressed by the Gladstone-Dale equation [33]. This relationship is defined for air while the air-fuel reactants and combustion products are present in the reaction zone. It has, however, been shown that the variation of gas composition in the luminous and homogeneous flame leads to errors less than 2% [34,35]. Thus, the temperature calculated by the Gladstone-Dale approximation, in which the refractive index of air is used instead of the refractive index of combustion products, is regarded as an accurate technique.

In a comprehensive study of using MZI method to analyze the heat flux of impinging flames, Morad et al. [36] investigated premixed impinging methane-air laminar flame experimentally and numerically. They found that for the shortest flame-to-plate distance and highest firing rate, the second peak in heat flux to the impinging surface can be observed, and these two parameters enable us to control the onset of the second stagnation point, which occurs in the highest firing rate (0.16 kW) and equivalence ratio of 1 according to a detailed numerical analysis. In the same vein, Iran doost et al. [37] studied the partially premixed flame characteristics of a laminar methane-air mixture using MZI technique at different equivalence ratios and Reynolds numbers. A co-annular burner produced the axisymmetric partially premixed flame. It was suggested that lowering the equivalence ratio (increasing the excess air ratio) increases the maximum flame temperature, whereas Reynolds number has a negligible effect on the maximum flame temperature.

As far as the authors are aware, despite the considerable amount of literature related to the heat transfer of impinging flame jets, the partially premixed axisymmetric co-flow impinging flames have not been explored. Considering all of these evidences, the need to understand the various perceptions of such flames is highlighted. The present paper seeks to document the heat transfer behavior of partially premixed round methane-air impinging flame jets, focusing on the most relevant factors, including equivalence ratio, Reynolds number, and burner-to-
plane distance using Mach-Zehnder interferometry method. Consequently, in the present study, for the first time the heat transfer characteristics of partially premixed flame have been compared with premixed flame in different flow conditions. Three axisymmetric co-annular burners with different diameter ratios were used to further investigate the effect of excess air ratio. This is the first study to undertake a detailed analysis of the local heat flux and temperature distribution of partially premixed impinging flames. Therefore, the findings should make a significant contribution to the field.

2. Materials and methods

2.1. Experimental setup

Fig. 1 shows the configuration of the axisymmetric co-annular stainless steel burner used to establish premixed and partially premixed flames in the current study. Three burners with the inner diameters of the inner tube equal to 4, 6, and 8 mm are selected, in which the fuel-air flows are mixed to have a premixed flame. Secondary air enters the combustion process from the annular region between the inner and outer tube to establish a controlled PPF. The inner diameter of the outer tube is 17 mm for all three burners studied. The thickness of all tubes is 1 mm, and their surfaces were polished to secure a smooth velocity profile at the exit. The length of the burners is 200 mm, which makes it reasonable to neglect the entrance effect. The tubes are filled with small stainless steel beads to homogenize the velocity profile and avoid flashback of the flame. A flow straightener is also installed on the tubes to provide uniform exit velocity and dissipate possible eddies of the flow.

A round copper plate with the diameter of 60 mm and thickness of 5 mm has been used as impingement plate. As shown in Fig. 1 (b), six thermocouples (type K), which have 6 mm distance from each other in the radial direction, are used to measure the surface temperature. The holes in which thermocouples were embedded have 1 mm in diameter and 4.5 mm in depth. There is only a distance of 0.5 mm between the thermocouple and the surface of the impingement plate. The maximum error occurs at the maximum rate of heat transfer for all experiments is below 0.3 K. The impingement plate was fixed on a positioner, which can move vertically to achieve different burner-to-plate distances. The burner was surrounded by a transparent Plexiglas enclosure with dimensions of 50 cm × 50 cm × 150 cm to prevent the flame from ambient disturbances.

Table 1

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Schematic of the experimental setup is depicted in Fig. 2. Methane and compressed air were measured and then they were premixed in a chamber, which was filled with tiny stainless steel beads. Methane gas with a purity of 99.9% was taken from a high-pressure cylinder, where a pressure regulating valve controlled its pressure. Two compressors were used to supply the primary and secondary air. The flow rates were measured by pre-calibrated flowmeters for both the fuel and air streams. To monitor the temperatures, “TESTO177” data logger was employed. For minimizing the potential error sources, relative humidity and ambient pressure were also measured by a pressure sensor and a humidity detector during experiments.
To cool the impinging plane, a water tank was used to supply the needed water which was heated to constant temperature of 38–40 °C. This temperature range was selected to avoid condensation of the combustion products on the impinging plane. The rate of cooling water is about 3 LPM for all experiment. Since the flowrate and temperature of the cooling water is kept constant during the experiments, the cooling procedure does not lead to any error in the results.

A schematic of the MZI setup is shown in Fig. 3. A 10 mW helium-neon (He–Ne) laser with 632.8 nm wavelength, two beam splitters (BS), two doublets (D), three mirrors (M), a CCD camera, and a PC are the main components of the interferometer.

The complete set of studied parameters, including flow rates of the fuel-air mixture in the inner tube and secondary air, Reynolds number of inner burner, equivalence ratio inner burner, and burner diameter are presented in Table 1. The flowrate of secondary air in the premixed flames is zero, and in the partially premixed flame, it is adjusted in such a way that the overall equivalence ratio equals one. All the flames, studied in the present work, were laminar.

2.2. Data reduction

In this part, the procedure of obtaining the impinging flame temperature field and heat flux from interferograms is described. The laser beam is parallel to the Z-axis, i.e., perpendicular to the XY-plane, as shown in Fig. 1. The refraction index in the radial direction changes because the flame is symmetric at each cross section. The He–Ne beam is divided into two parts by the first beam splitter; one passes through the hot medium, and the other passes through the ambient (reference beam). Accordingly, an optical path difference occurs, which is caused by the medium's refractive index variations [38]:

$$\varepsilon = \frac{\Delta \rho}{2\pi} = \frac{1}{\lambda_0} \int_{-\delta}^{\delta} (n(x, y, z) - n_{ref})dz$$

(1)

where $\varepsilon$ is the fringe displacement, in which $\varepsilon = 1, 2, 3, ...$ represent the light fringes and $\varepsilon = 0.5, 1.5, 2.5, ...$ indicates the dark fringes. $\lambda_0$ is the wavelength of the light source and has the value of 632.8 nm. $n_{ref}$ is the refractive index of the air at reference state and $n(x, y, z)$ is the local refractive index of the flame. According to Fig. 4 and transforming Eq. (1) to the polar coordinates gives:

$$\varepsilon = \frac{\Delta \rho}{2\pi} = \frac{1}{\lambda_0} \int_{0}^{2\pi} (n(r) - n_{ref}) \frac{2r}{\sqrt{r^2 - y^2}} dr$$

(2)

Eq. (2) has a form of Abel transform and its inversion is [39];

$$n(r) - n_{ref} = \frac{\lambda_0}{2\pi} \int_{0}^{\infty} \frac{\Delta \phi(y)}{\sqrt{y^2 - r^2}} dy$$

(3)

To calculate this integral and obtain the required parameter $(n(r)-n_{ref})$, a second order polynomial in form of $y = ax^2 + bx + c$ is used for fitting of discrete value of fringe number to obtain the gradient of fringe number [21,39]. When the distribution of refractive index is determined, the Gladstone-Dale equation is used, which links the density of gas mixture to the obtained refractive index:

$$n(r) - n_{ref} = k_i y_i$$

(4)

where $y_i$ is the mass fraction of each species and $k_i$ is the Gladstone-Dale constant of that species. The ideal gas law is then used to relate the temperature to the density of the gas mixture since the combustion occurs in rather a high temperature and at ambient pressure:

$$T = \frac{P}{\rho R} \bar{W}_{mix} \rho \bar{R}$$

(5)

where $\bar{R}$ is the universal gas constant, $P$ is the pressure and $W_{mix}$ is the molecular weight of the mixture which is expressed as:

$$W_{mix} = (\sum y_i W_i)^{-1}$$

(6)

where $W_i$ is the molecular weight of the $i$th species. The flame temperature distribution is then obtained by combining the above equations:

$$T(r) = (n(r) - 1)^{-1} \alpha (r)$$

(7)
in which $a(r) = \left(\frac{r}{L}\right) k_{mix} W_{mix}$, and $a(r)$ is assumed to be constant throughout the flame by assuming the local composition to correspond to that of air [40]. By this assumption, which will be further discussed in the uncertainty analysis section, the local temperature distribution is expressed as [40]:

$$T(r) = \frac{T_{ref} (n_{ref} - 1)}{[n(r) - 1]}$$

(8)

where $T_{ref}$ and $n_{ref}$ are the ambient temperature and refractive index, respectively. Therefore by calculation of $n(r)$ and having distance of fringe from the X-axis, the temperature distribution of flame can be obtained. The Reynolds number and equivalence ratio of inner jet fuel/air mixture are respectively defined as Eqs (9) and (10):

$$Re_{in} = \frac{\rho_{mix} Vd}{\mu_{mix}}$$

(9)

and

$$\varphi = \frac{m_{fuel,in}}{m_{in}}$$

(10)

where $\mu_{mix}$ is determined by Ref. [41]:

$$\mu_{mix} = \frac{\sum (\mu_i x_i \sqrt{MW_i})}{\sum x_i \sqrt{MW_i}}$$

(11)

The local convection heat transfer represented by:

$$h = -k \frac{dT}{dN} \frac{1}{T_{LF} - T_W}$$

(12)

Where $k$ is the thermal conductivity of combustion products at the surface temperature obtained from equation (13) [42]:

$$k = \frac{\sum x_i k_i}{\sum \frac{x_i}{\beta_{ij}}}$$

(13)

and $\beta_{ij}$ defined as:

$$\beta_{ij} = 1 + \left( \frac{n_i}{n_j} \right)^{0.5} \left( \frac{MW_j}{MW_i} \right)^{0.25}$$

(14)

$T_{LF}$ in equation (12) is the temperature of the last fringe near the wall that is used in the calculation of temperature gradient. In fact, $T_{LF}$ is the maximum temperature of each position of impingement plate that we calculated the Nusselt number. According to our previous work [43], for measurement of the temperature gradient, $T_{LF}$, we need to determine the temperature of the surface that is obtained by the thermocouple and the distance of nearest fringe to the wall. However, there are compressed fringes at this region, as shown in Fig. 5. For more accuracy, we used other fringes as well for measurement of the temperature gradient. To further improve accuracy and avoid any potential error, we used a MATLAB code in image processing to calculate the fringe distance from the wall.

Then the Nusselt number at different positions is determined from:

$$Nu = \frac{hd}{k}$$

(15)

Eq. (15) shows that $Nu$ number is not only dependent on the surface temperature but also on the temperature gradient near the surface, while the latter has a more significant effect than the other parameters in specific cases. Hence, in few cases where the maximum surface temperature and maximum $Nu$ number do not coincide at the same point, it can be inferred that the influence of temperature gradient has been dominating.

The heat transfer rate from the flame to the surface is finally obtained by:

$$q^* = -k \frac{dT}{dN} \bigg|_{N=0}$$

(16)
2.3. Uncertainty analysis

As a source of error in combustion experiments is due to uncertainties of equivalence ratio and Reynolds number. The maximum uncertainty was measured for equivalence ratio and for Reynolds number are 4.7% and 3.3%, respectively. The details of these uncertainty measurements are expressed in the literature [44,45]. The accuracy of temperature and pressure measurements are about 0.1 K and 100 Pa respectively, that are obtained from the measurement device precision. The accuracy of firing distance from the plate is about 0.0018 mm that is measured from the number of pixels per each millimeter in interferometry image. Another source of error is deflection of the laser beam. The laser beam is slightly deflected through the flame while passing the test section. Therefore, it does not move along its original path when reflected by the mirror. Aebischer and Rechsteiner [46] showed that the effect of this divergent angle is negligible in Mach–Zehnder interferometer results.

The application of air’s refractive index instead of combustion products in obtaining temperature field is another error source. In partially premixed flame the combustion product’s refractive index is identical to that of air [40]. The average error due to this assumption for the 2D axisymmetric flames is about 2.3% at \( \varphi = 2 \) and is below 2% at equivalence ratios less than 2 [47].

Also for a more detailed investigation about the effects of the refractive index of combustion products on the flame temperature, the flame temperature obtained from the interferometry method at \( \text{Re} = 600 \) and \( \varphi = 0.8 \) is compared with that of thermocouples (Fig. 6), where a good agreement could be observed (maximum error is 2.4%).

Fig. 9. Fringe pattern of the flame at different burner-to-plane distance \( \text{Re} = 600, \varphi = 1.5, \) and \( \varphi_o = 1 \).
The inner reaction zone where the fringe patterns are compressed has been observed in areas near the impinging plane and especially above the inner reaction zone. The maximum temperature gradient has been reduced and thereby a shorter axial distance is required for this penetration, so the flame height decreased.

The effects of Reynolds number, equivalence ratio, the burner-to-plane distance, and the diameter of the co-annular burner on heat transfer characteristics of the laminar flame jet for the premixed and partially premixed flames are discussed in the next sections. In all cases, except for the tests studying burner diameter's effect, the burner's inner tube diameter is 6 mm.

3. Results and discussion

Thermocouple measurements are first corrected to account for the effects of convection and radiation [48]. The maximum uncertainty of Nusselt number and heat flux has been calculated to be about 4.3% and 6.2% respectively. The detail of calculation procedure presented in the literature [49].

Also, it should be noted that each experimental case is repeated 8 times. To process the results, the outlying values are eliminated and finally, the average value of all reliable data is presented as the result of the specific case.

3.1. Effect of burner-to-plane distance

The effect of burner-to-plane distance is illustrated by fringe patterns in Fig. 9. By decreasing $H/d$, the fringe above the inner reaction zone become compressed near the impingement plate and increase the temperature gradient near stagnation point but by further decreasing $H/d$, the fringe above the reaction zone become separated two parts and goes away from the impingement plate that results in a significant decrease in the gradient temperature at the stagnation point. In partially premixed flame, secondary air not only leads to reduce the height of inner reaction zone but also pushes fringe pattern especially near impingement surface that is one of the effects of an increase in heat transfer, also added air extends fringes toward the wall-jet region. The dimensionless form of the burner-to-plane distance, $H/d$, was varied from 1.67 to 6.67, while the Reynolds number and the equivalence ratio were kept fixed at $Re = 600$ and $\varphi = 1.5$. The overall equivalence ratio ($\varphi_o$) was equal to 1 for the PPF case. In each test, the secondary air was added to the combustion process to investigate the effect of partially premixing while the other parameters were kept constant. When the flame impinges onto the target plane, hot combustion products spread from the stagnation region to the wall jet region of the plane. In the further distance in which the flame does not have contact with the impinging plane, the fringes congestion decreases and the surface temperature is lower. By getting closer to the plane, it intercepts the flame and the fringes density increases. Thus the temperature gradient is higher in the lower burner-to-plane distances. It is apparent from the graph that in the partially premixed flame, the airflow around the flame reduces the flame's height and width.

Fig. 10 illustrates the radial Nusselt number distribution of laminar premixed flame at various burner-to-plane distances. In the closest position of the impinging plane to the flame ($H/d = 1.67$), the low Nusselt number is relatively low in the stagnation point. In this position, the inner reaction zone is intercepted by the plane and thereby low-temperature unburned gases touch the plane. Then, the combustion process becomes complete in the radial distance from the stagnation point, and the maximum Nusselt number occurs at a position downstream from that point. The position of maximum Nusselt number moves toward the stagnation point by increasing the burner-to-plane distance. The maximum stagnation point Nusselt number was observed when the tip of the inner reaction zone just touched the plane ($H/d = 5$). Further increase of the burner to plane distance decreases the Nusselt number with the same trend in the radial direction.

Fig. 11 indicates the effect of adding secondary air to the combustion process in such a way that the equivalence ratio in the primary zone is 1.5, while the secondary air is injected to provide the stoichiometric conditions for the flame. From this graph, we can see that partial premixing process enhances the heat transfer to the impinging plane. The wall jet region gets a higher improvement of the heat flux compared to the stagnation region. This observation is more obvious on the burner-to-plane distances that the inner reaction zone is intercepted by the plane. For such conditions, the inner reaction zone spreads out on the plate surface, and the excess air permeates to the rich fuel-air mixture, so the heat flux is being enhanced in the wall jet region. The stagnation point Nusselt number at $H/d = 5$ decreases in the partially premixed flame. Since the flame height is less in the partially premixed case, the inner reaction zone moves further away from the plane-side.
compared to the premixed flame (see Fig. 12). Therefore, this lower Nusselt number is observed at the stagnation point of $H/d = 5$. The maximum flame temperatures at different conditions are presented in Table 1. It is apparent from the table that in the partially premixed state, the maximum flame temperature is ca. 4.5% higher than the premixed flame. The average Nusselt number increment by partial premixing is computed to be around 10.2% at different burner-to-plane distances.

3.2. Effect of Reynolds number

The effect of Reynolds number is illustrated by fringe patterns in Fig. 12 by increasing Reynolds number, the height of the inner reaction zone increases, the light fringe above the inner reaction zone becomes extended, and a new dark fringe appears at the center of this fringe. Also, the fringe above the inner reaction zone has been divided into two parts that decrease the temperature gradient at the stagnation point.
The effect of Reynolds number for a partially premixed flame on the Nusselt number is shown in Fig. 14. The other operating conditions (equivalence ratio and \( H/d \)) are constant, and the excess air is added to the reaction zone such that \( \phi_o = 1 \). The trend of Nusselt number of partially premixed flame is analogous to the premixed flame.

The radial Nusselt number distribution of premixed and partially premixed impinging flame of methane at two different Re numbers is shown in Fig. 15. It can be seen from the graph that adding secondary oxidizer to the combustion process enhances the heat flux, particularly in the wall jet region compared to the stagnation point. The enhancement of the total Nusselt number in the partially premixed state is about 12.3% at different Re numbers. This increase may be explained by the flow of the excess air around the flame, which makes the hot gases and combustion products compact in the region close to the plane (see Fig. 16). In addition, adding excess air such that the overall equivalence ratio is equal to one, approximates the rich fuel-air mixture to the stoichiometric mode in which the maximum flame temperature obtained. According to the results of interferometry images data reduction, partially premixing of the flame causes approximately 4.5% increase in the maximum flame temperature as presented in Table 1.

### 3.3. Effect of equivalence ratio

The effect of equivalence ratios is illustrated by fringe patterns in Fig. 16. There are oval fringes at the top of inner reaction zone at \( \phi = 1 \), and by decreasing or increasing the equivalence ratio these oval fringes are divided into parts and the temperature gradient decreases at the stagnation point and increase some away from stagnation point.

Fig. 17 provides the results obtained for the radial Nusselt number variation of premixed flame at five different equivalence ratios (\( \phi = 0.8, 1, 1.3, 2, \) and 3). The other operational parameters were kept constant (Re = 600 and \( H/d = 5 \)) during these set of experiments. From the lean to rich conditions, the height of inner reaction zone changes because of different flame speed at these five conditions. Although in the stoichiometric condition the flame has the minimum length, the maximum adiabatic temperature, because of the complete combustion in this condition, results in a high Nusselt number at \( \phi = 1 \). The stagnation point Nusselt number is 12.1 at \( \phi = 1 \), which is relatively greater than the peak Nusselt number at the other equivalence ratios. Further increase of \( \phi \) prolongs the flame height, so the inner reaction zone is intercepted by the plane, and the flame is being spread on the impinging plane. At \( \phi = 1.3 \), the peak Nusselt number is 11.6 at \( r/d = 1 \) and shifted toward the wall jet region. For higher equivalence ratios, the flame height gets greater and, therefore, spreads wider on the plane. As a result, the peak Nusselt number takes place at a further distance from the stagnation point. As the fuel-air mixture gets richer, the adiabatic flame temperature decreases, since the fuel does not find sufficient oxidizer to burn perfectly, leading to incomplete combustion. In this situation, it seems that adding excess air to the combustion, so that the overall equivalence ratio equals one (\( \phi_o = 1 \)), improves the heat transfer rate.

According to the maximum flame temperatures in Table 1, in the stoichiometric condition, the maximum flame temperature is 2095 K, which is apparently higher than the other premixed and partially premixed flame temperatures at different equivalence ratios.

Because of the fact that the secondary air is added to the combustion process to provide sufficient oxidizer and approach the stoichiometric condition, the tests of PPF flame were conducted only for rich fuel-air mixtures (\( \phi = 1.3, 2, \) and 3). The other parameters were kept constant (Re = 600 and \( H/d = 5 \)). The flowrate of the secondary air was adjusted so that \( \phi_o \) equals one. Fig. 18 shows the radial distribution of Nusselt number for the partially premixed flame at different

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**Fig. 13.** Radial Nusselt number of premixed flame at different Re numbers for \( H/d = 5 \) and \( \phi = 1.5 \).

**Fig. 14.** Radial Nusselt number of partially premixed flame at different Re numbers for \( H/d = 5, \phi = 1.5, \) and \( \phi_o = 1 \).

**Fig. 15.** Comparison of Nusselt number distribution in PF and PPF at different Re numbers for \( H/d = 5, \phi = 1.5, \) and \( \phi_o = 1 \).
equivalence ratios. Comparison of the two flame types in the graphs reveals a clear enhancement of the heat flux in the partially premixed flame. The flame's maximum temperature increases about 6.5% in the partially premixed flame compared to the premixed flame (Table 1).

For a more precise comparison, the Nusselt number distribution of premixed and partially premixed flames are plotted in Fig. 19 at two equivalence ratios of 1.3 and 3. From the graph, we can clearly see that the Nusselt number of the partially premixed flame to the impinging plane is higher than the premixed flame, especially in the stagnation region. The average total heat flux enhancement for all of the examined equivalence ratios is about 12%. There is a relatively great difference in the stagnation point heat flux at $\varphi = 1.3$. Since by partially premixing,
3.4. Effect of burner diameter

The effect of burner diameter is illustrated by fringe patterns in Fig. 20. The inner reaction zone of the flame has not been intercepted by the plane, therefore there is oval firings above the inner reaction zone that it cause the peak of heat transfer occurs at the stagnation point. Like other cases secondary air push fringe pattern toward the impingement surface that is one of the effects of secondary air on increase of heat transfer.

Fig. 21 shows the effect of burner diameter on the heat flux of the premixed impinging flame, while keeping the other operating conditions constant (Re = 200, H/d = 5, and φ = 2). To keep these parameters constant, the mixture’s axial velocity and absolute separation distance (H) varied for different burner diameters. A crossover at r = 8 mm shows the trend of heat transfer for r > 8 is opposite in comparison with r < 8. This can lead to the fact that for constant Re number, firing rate is higher for the burner which has larger diameter since this condition provides more mass flow rate. Therefore heat transfer for larger burner is higher in wall-jet region. Peak heat flux values were obtained in the stagnation point for all the three burners because the Re number is low and the tip of the flame does not have contact with the plane. The heat flux in the stagnation region for the smaller burner diameters is higher. This can be attributed to the higher convective heat transfer because of higher axial velocity in the smaller burner at the same Re number. Moreover, the secondary reason is less absolute separation distance (H) at constant H/d, thus for the smaller burner, the inner reaction zone is closer to the impinging plane.

The difference in the radial heat flux distribution in both the stagnation region and the wall jet region at different burner diameters can be explained by the special shape of each flame. The inner reaction zone is wider for larger burner diameter (d = 8 mm) and narrower for smaller diameter burner (d = 4 mm). Thus, thermal energy at the stagnation point and its vicinity will be more intensive for the burners with smaller diameters. According to these data, we can infer that the heat flux distribution is more uniform for the largest burner diameter (d = 8 mm).

Fig. 22 shows the radial heat flux of partially premixed impinging flame at three different diameter ratios of the co-annular burner (the inner tube’s diameter is variable, and the outer tube’s diameter is constant.). All the other operating conditions are identical to the premixed flame (Re = 200, H/d = 5, and φ = 2). What stands out in the graphs is that the trend of heat flux is similar in premixed and partially premixed flames.

The comparison of premixed and partially premixed flames is presented in Fig. 23. It can be seen from the data in the graph that the enhancement of heat transfer in the partially premixed condition is about 11.4%. Since the fuel-air mixture is in a rich state, the addition of excess air in such a way that the overall equivalence ratio is one, causes the fuel to burn completely and increase the flame temperature. The trend of partially premixed flame heat flux distribution is identical to that of premixed flame, as can be seen in Fig. 23. Furthermore, as already mentioned, the reduction of burner’s inner tube diameter causes the heat flux to increase in the stagnation region and decrease in the wall jet region.

Increasing the diameter of the burner’s inner tube decreases the area available for secondary air and hence increases its velocity. Since the overall equivalence ratio in all three cases is one, this velocity change does not make a significant difference in the heat flux distribution except for a more decrease in the flame height. The increase in maximum flame temperature resulting from partial premixing is about 5.5% according to the interferometry images (Table 1).

4. Conclusion

Experimental investigation of laminar impinging premixed and the partially premixed methane-air flame was conducted in this study. A co-annular burner is used in which the fuel-air mixture flows in the inner tube, and the excess air flows in the annular region between the inner and the outer tube. The effect of partially premixing the flame on the
heat transfer distribution was investigated and compared with the results of the corresponding premixed flame. The main conclusions may be summarized as:

- Partial premixing of the flame (up to the overall equivalence ratio of one) enhances the heat transfer to the impinging plane in all conditions and increases the maximum flame temperature.
- The enhancement of Nusselt number by partial premixing in different cases of burner-to-plane distance, Re number and equivalence ratio about 10.2%, 12.3%, 12% respectively and the enhancement of heat flux by partial premixing in different cases of burner diameter is about 13.5%.
- The heat transfer characteristic is dependent on the proximity of the inner reaction zone to the plane. Accordingly, various actions may be chosen to improve the heat transfer characteristics in such an impinging flame jet: Reynolds number increment (i.e., increasing the flame height), enhancing the convective heat transfer, and reduction of the burner-to-plane distance.
- In the case of intercepting the inner reaction zone by the plane, the peak heat flux shifted from stagnation point to the wall jet region.
- Reduction of the burner diameter increased the heat flux at the stagnation region, while opposite trend was observed for the heat flux to the wall jet region.

Fig. 20. Fringe pattern of the flame at different diameter of inner burner Re = 200, and $\varphi = 2$, and $\varphi_o = 1$. 
Fig. 21. Radial heat flux of premixed flame at different burner diameters for \(H/d = 5\), \(Re = 200\), and \(\varphi = 2\).

Fig. 22. Radial heat flux of partially premixed flame at different burner diameters for \(H/d = 5\), \(Re = 200\), \(\varphi = 2\), and \(\varphi_o = 1\).

Fig. 23. Heat flux distribution comparison for PF and PPF at different burner diameters for \(H/d = 5\), \(Re = 200\), \(\varphi = 2\), and \(\varphi_o = 1\).

Notes

The authors declare no competing financial interest.

References


