International Journal of Engineering Research & Technology (IJERT)
ISSN: 2278-0181
Vol. 2 Issue 8, August - 2013

"Analysis Of Heat Transfer Characteristics Of Flame Impinging To A Plane Surface Perpendicular To Flame Jet Axis"

Sushant Sahu [1], Mahesh Dewangan [2]

[1]Department of Mechanical Engineering (Thermal Engineering), Shri Shankracharya Group of Institution, Junwani (SSGI, Bhilai), Chhattisgarh, India.

[2]Department of Mechanical Engineering, Shri Shankracharya Group of Institution, Junwani (SSGI, Bhilai), Chhattisgarh, India.

ABSTRACT

This thesis work consists a Flame jet impingement which is used in industrial heating and melting, aerospace applications also in safety research etc. to obtain high heat transfer rates. Jet impingement is generally used where high convective heat transfer rates are required. A theoretical analysis of laminar flame impinging perpendicularly on a flat surface has been developed to predict the influences of jet Reynolds number, ratio of plate separation distance to nozzle diameter and equivalence ratio on the Nusselt number. The analysis is based on numerical solution of the governing differential equations for conservation of mass, momentum and energy. Ethane, Propane, Acetylene, Hydrogen, Nitrogen and Oxygen have been considered as fuel and oxidizer respectively. It has been observed that the heat flux distribution for perpendicular plate is increasing with an enhancement on velocity gradient. The heat flux is higher for Nitrogen as compared to other fuels. The local heat flux decreases with an increase in heating height. A fuel rich mixture increases the plate heat flux. The average Nusselt number, Nu increases with an increase in jet Reynolds number, Re and separation distance. The average Nusselt number, Nu decreases with equivalence ratio. The increase in Nu is profound at higher values of Re. The conclusions and future scope has been manifested at the end of the work.

Nomenclatures

Cp	specific heat
D	sphere or cylinder diameter
d	nozzle diameter
G	mass of gas flowing per second per unit cross-sectional area of gas stream
h	enthalpy per unit mass of mixture

h_d	atomic dissociation energy
H_t	total enthalpy
K	thermal conductivity
Nu	Nusselt number
Pr	Prandtl number
Ra	Rayleigh Number
q	heat flux
q_f	firing rate
R	radial distance from stagnation point along impingement surface
St	Stanton number
H/d	dimensionless distance between nozzle and target surface
T	temperature
U	gas stream velocity or time averaged longitudinal velocity
Y	mass fraction of different species
h	inclination angle of impingement plate
q	density
b	velocity gradient (s_1)
l	dynamic viscosity
m	kinematic viscosity
D	change in particular quantity
e	emissivity
Ø	Equivalence ratio
X	oxygen enrichment ratio
b	body or target
e'	condition at outer edge of body
eq	equilibrium
f	frozen condition

1. INTRODUCTION

Flame impingement heating of solids has been used for many years. Some typical applications include melting of scrap metal, shaping glass and heating metal bars. Another application is heating metal billets in a reheat furnace prior to rolling or shaping. In addition, it is also used in metal fabrication and assembly applications including soldering, brazing, cutting and welding. Use of direct flame impingement in industrial furnaces significantly enhances the convective heat transfer rates from combustion products to the load, which ultimately increases productivity, reduces fuel consumption and lowers pollutant emission. Jackson and Kilham [1] studied the impingement of hot gases on a cylindrical surface at right angle. The cylindrical surface was rotating at 40 rpm. This rotation had no effect on the heat transfer because the tangential velocity of the rotating tube never exceeded 0.1% of the free stream gas velocity. Woodruff and Giedt [2] investigated oxy-acetylene laminar flames flowing past a thin molybdenum test plate in a direction parallel to the plate surface. It is seen in the literature that most of the studies are related to flame impingement normal to a flat surface. Other configurations studied are very few. Milson and Chigier [3] studied methane and methane–air flames striking normal to a plate surface. The pressure and axial velocity profiles of the flame jet are very similar to those of the isothermal jet. A number of review articles relating to some specific aspects of flame impingement heat transfer is also available in literature [4-5]. Comprehensive knowledge of flame impingement will be a valuable asset for engineers and technologists working in this field. A number of experimental, empirical, semi-analytical and numerical studies have been conducted. The experimental results of Van der Meer [6] also reveal this similarity but show that the flame jet axial velocity decays slightly faster than in the isothermal jet. The primary difference between the flame jet and the isothermal jet is the presence of the reaction zone in the free jet and possibly in the stagnation and the wall jet regions along with the temperature gradient in the flame jet. This caused the area of the plate around the stagnation point to be cooler than the maximum temperature, which was achieved some distance away. Excellent review papers [7-12] have been published highlighting different issues. The dominant feature of both flames was the presence of a cool central core of un-reacted gas, which impinged on the plate (nozzle-plate distance/burner diameter equals 10–16). Rigby and Webb [8] studied the heat transfer from normal impinging diffusion flame jets. Six heat transfer mechanisms have been identified in previous flame impingement studies. These mechanisms include conduction, convection, thermo chemical heat release (TCHR), radiation, condensation and boiling [13,14]. The different configurations studied by researchers Baukal and Gebhart [15,16] investigated the heat transfer from oxygen enhanced/natural gas flames impinging normal to a plane surface. The aerodynamics of the single flame jet is similar to those of the isothermal jet. Zhang and Bray [17-18] studied different impinging flame shapes for normal flame impingement on a plane surface..For some of these mechanisms, there have been several types. For example, both forced and natural convection have been considered. The relative importance of a mechanism depends on the experimental conditions. For example, when targets have been located inside a furnace, radiation from the hot walls has been very important. However, for targets located in a large room at ambient conditions, radiation from the environment is negligible. In some of the studies, multiple mechanisms have been important. However, the relative contribution of each mechanism has not usually been determined. For example, forced convection has commonly been combined with TCHR. In other studies, some mechanisms have been completely ignored without sufficient justification. Carefully controlled experiments are required to improve the understanding of convective and radiative heat transfer. Dong et al. [19-21] mentioned that complete understanding of the impingement flame jet is not possible yet because of the limited information available from the literature, and also information about the heat transfer characteristics from flame jets is very much less. Dong et al. [19-21] performed an experimental study to determine the heat transfer characteristics of a pre-mixed butane/air round flame jet of low Reynolds number impinging upwards normally on a flat rectangular plate. Here, S and O denote stagnation and geometrical impingement points, respectively. It was also observed that the position of the maximum heat flux point would be shifted away from the geometrical impingement point by reducing the angle of incidence. Su and Liu [21] investigated experimentally the effect of impingement angle. The angle was varied, and different values like 45^0, 75^0 and 90^0 were used. It is mentioned that the optimized impinging angle is important for the arrangement of injectors and combustion efficiency in self ignition propellants liquid rocket engine. It is seen that perpendicular impingement has a higher combustion efficiency, but the impinging flame may induce flashback and damage the combustor injector.

The most recent and comprehensive review on flame impingement heat transfer is provided by Chander and Ray [22]. The heat transfer due to the impingement of a flame on a target solid surface depends upon the flame structure, the temperature field in the near vicinity of the plate, and both convective and radiative properties of the constituent species of the flame. The flame temperature is influenced by the flowfield of impinging jet which comprises the free jet region, stagnation region, and wall jet region [23-24].

2. THEORETICAL ANALYSIS

The figure shows the arrangement for the flame impingement through surface perpendicular to the flame jet axis.

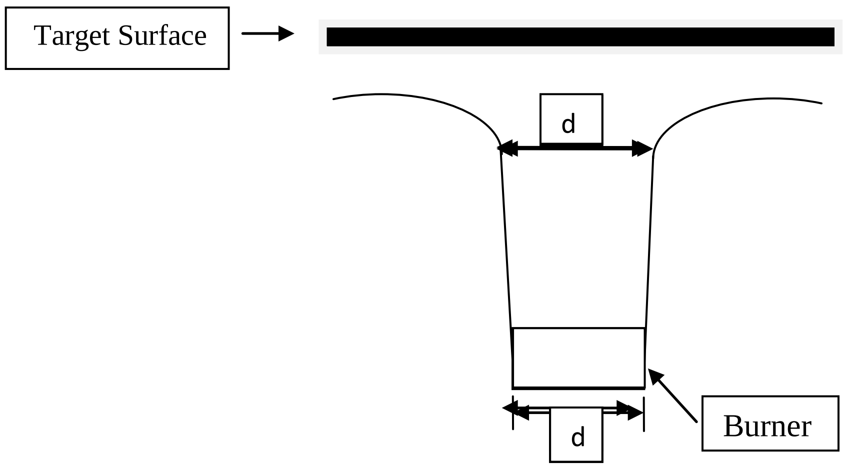

Fig:-1 Flame jet impinging normal to target surface

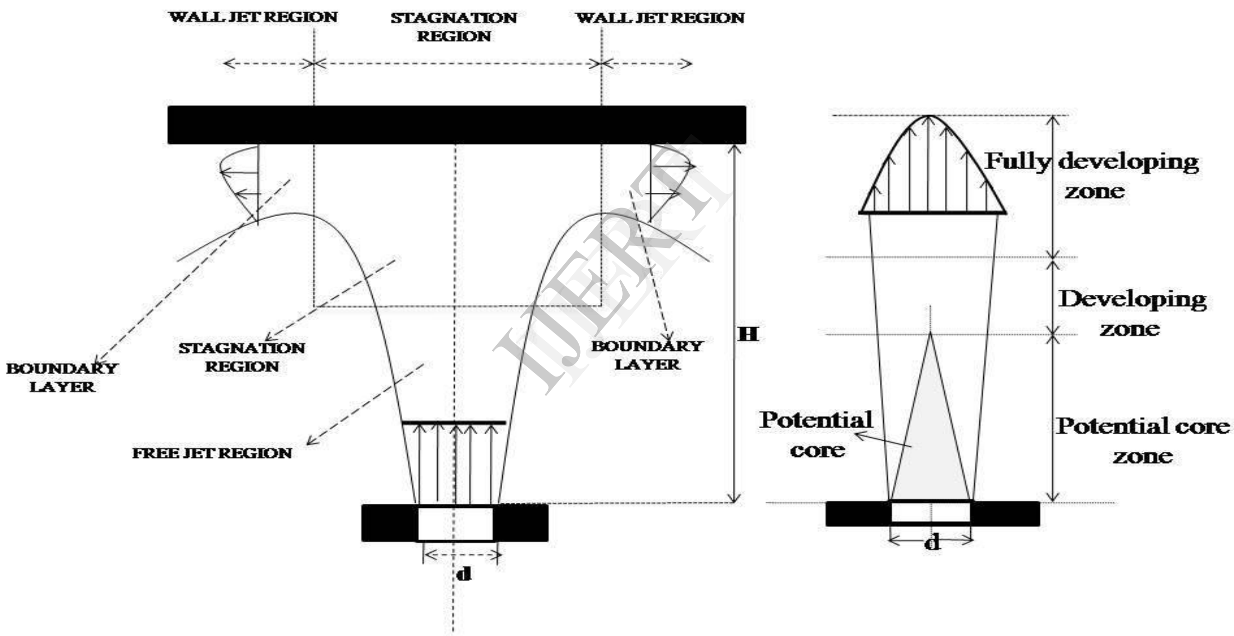

Fig:-2 Function of flame at the time of impingement

Laminar flows. In related studies, Hoogendoorn et al. (1978) and Popiel et al. (1980) determined two empirical correlations, for the heat transfer to the stagnation point. These were based on the axial distance between the target and the burner. The correlation for 2 < L < 5 was:

$$q''_{s,conv} = \frac{k_{ref}}{d_n}\{(.065+0.084\mathrm{L})Re_{n,ref}^{0.5}\,Pr_{ref}^{0.4}\}\frac{h_e^s-h_w^s}{Cp_{ref}} \quad (1)$$

The correlation for L > 12 was

$$q''_{s,conv} = \frac{k_{ref}}{d_n}\{(1.37\text{-}1.8\text{L})10^{-3}\, Re_{n,ref}^{0.75}\; Pr_{ref}^{0.4}\}\frac{h_e^s - h_w^s}{Cp_{ref}} \quad (2)$$

Both correlations closely matched the data. For 5 < L < 12, the data showed a peak for Nu. No correlation was given. A modified form of a semi-analytic solution was also determined. It was a form of the equation recommended by Sibulkin (1952). Sibulkin's equation, along with many variations of it, is discussed by Baukal and Gebhart (1996). The modified correlation was:

$$q''_{s,conv} = \frac{k_{ref}}{d_n}\{2.37(\text{L}+1)^{-0.5}\, Re_{n,ref}^{0.5}\, Pr_{ref}^{0.4}\}\frac{h_e^s - h_w^s}{Cp_{ref}} \quad (3)$$

valid for 8 < L < 20. This underpredicted the data by as much as 60%. This was believed to have been caused by the failure to include the effects of free jet turbulence. Another correlation was recommended, which included those effects:

$$q''_{s,conv} = \frac{h_{ref}}{d_n}\{[\frac{2.37}{(L+1)^{0.5}} + 2.22(\frac{TuRe_{n,ref}^{0.5}}{100})$$

$$-2.76(\frac{TuRe_{ref}^{0.5}}{100})^2]Re_{n,ref}^{0.5}\, Pr_{ref}^{0.4}\}\frac{h_e^s - h_w^s}{Cp_{ref}} \quad (4)$$

valid for L > 8. This compared favorably with the data. For L > 4, the maximum heat flux occurred at the stagnation point. For L = 2 and 3, the maximum flux occurred at about R = 0.5 and 0.2, respectively. Kataoka et al. (1984) studied an air/O_2/CH_4 flame. The heat transfer to the stagnation point was correlated by:

$$q''_{s,conv} = \frac{k_e}{2r_{1/2v}}\{1.44Re_{2r_{1/2v}}^{0.5}\; Pr_e^{0.5}\,(L - X_v)^{0.12}\}(t_e - t_w) \quad (5)$$

rl/2v is the radius from burner axis, near the stagnation point, where the velocity is 1/2 of the velocity along the axis of symmetry. This correlation is valid only for L >X,. with X,, correlated by:

$$X_r = 2.82(\frac{\rho_n}{\rho_x})^{0.29}\; Re_{n,X=0}^{0.07} \quad (6)$$

In a subsequent study, Kataoka (1985) found the maximum heat flux for the same flame occurred at L = X. Turbulent flows. Shorin and Pechurkin (1968) empirically determined

$$q''_{s,conv} = \frac{k_e}{d_n}\{4.04Re_{n,e}^{0.2}Pr_e\}(t_e - t_w) \tag{7}$$

This was valid for L < X. The correlation matched the experimental data within 15%. Another correlation was given for the heat flux for L >X_v.

$$\frac{Nu_{s,e}\ (X_v<L<14)}{Nu_{s,e}\ (L<X_v)} = 0.8e^{-0.36\ \frac{(L-X_v)^2}{L}} \tag{8}$$

The local heat transfer, as a function of R, was correlated by:

$$q''_{r,conv} = \frac{k_e}{Rd_n}\{3.22Re_{n,e}^{0.4}Pr_e e^{-0.36\ \frac{(L-X_v)^2}{L}} - 3.6\frac{R}{L}\}(t_e - t_w) \tag{9}$$

for 0 <*R/L* < 0.9 and X_v <*L* < 14. Vizioz and Lewes (1971) investigated industrial-scale, air and oxygen-enriched air (I~ = 0.30) natural gas flames impinging on cooled flat plate targets, located inside a furnace. Three different types of flames, with various levels of swirl, were tested. The data were correlated by:

$$q''_{r,conv} = (k_e/r)\{Nu_{r,e}\}(t_e - t_w) \tag{10}$$

Empirical correlations are generally given in two forms. The first is the Nusselt number, which is a dimensionless heat transfer coefficient. The second form is directly in terms of heat flux. Baukal and Gebhart [11] published an extensive review paper on empirical flame impingement heat transfer correlations. The correlations developed are functions of target geometry, location on the target where the desired heat flux is calculated, heat transfer mechanism(s) and type of flow (laminar or turbulent). The correlations can be for stagnation point heat flux, local heat flux at some radial location or the total heat flux. Hoogendoorn et al. [28] conducted an experimental study with a rapid heating tunnel burner and mentioned that extremely high convective heat fluxes can be obtained by causing the jet to impinge on the heated flat surface. They developed the following correlations for suitably fitting the experimental data.

$$\mathrm{Nu} = (0.65+0.85\frac{x}{D})Re^{0.5}Pr^{0.4} \quad \text{for } 2\leq \frac{x}{D} \leq 5 \tag{11}$$

The above equation closely represents the experimental results within the heat transfer core region.

$$\mathrm{Nu} = (137 - 1.8\frac{x}{D}) \times 10^{-3} \times Re^{0.5}Pr^{0.4} \quad \text{for} \rightarrow x \geq 12 \tag{12}$$

Here, x is the distance measured from the nozzle, and D is the burner nozzle diameter. An empirical relation for calculation of the heat transfer on a flat surface and also for normalizing the experimental results.

The relation used was:

$$q_0 = 0.9(\frac{U}{r_{0.5,T}})^{0.5} (\rho\mu)^{-0.5} (\overline{\rho\mu})^{-0.5} (Pr_f)^{-0.6} \Delta h_e \tag{13}$$

Here, $r_{0.5T}$ is the flame temperature half radius (the radius at which the temperature increment above ambient is half the corresponding axial temperature increment over ambient) and subscripts ''f'' and ''e'' refer to frozen and equilibrium conditions.

Dong et al. [12] deduced correlations with a multiple regression method for their study related to flame impingement on flat surfaces. For the laminar flame correlation, the maximum Nu is

$$Nu_{max}\, Pr^{-0.4} = 3.988Re^{0.2117}\, \emptyset^{-0.6458}\, (\frac{H}{D})^{0.07217} \tag{14}$$

The above expression is true for $600 \leq Re \leq 1500, 0.7 \leq \emptyset \leq 1.2$ *and* $1 \leq H/d \leq 8$

Average heat flux for the entire surface can be expressed by a single expression like

$$\overline{Nu}Pr^{-0.4} = 0.159Re^{0.4745}\, \emptyset^{-0.382}\, (\frac{H}{D})^{0.2908} \tag{15}$$

$600 \leq Re \leq 1500, 0.7 \leq \emptyset \leq 1.2, 1 \leq H/d \leq 8 \leq$ *and* $r/d \leq 6.5$

Here, the maximum deviation from the experimental result is13.3%

When the Re > 1500, the experimental results greatly exceed the results predicted by the Sibulkin equation because of the occurrence of turbulence, which enhances the air/fuel mixing and combustion. Dong et al. [25] observed that the maximum heat flux point shifted towards the stagnation point as the Re was increased from laminar to turbulent. In the laminar region, the stagnation point heat transfer is given by

$$Nu_s = 0.56Re^{0.492} \quad (16)$$

Laminar flows: You (1985) studied pure fuel jet flames impinging on a plate. The fuel flow from the burner nozzle was laminar. However, the buoyant plume impinging on the plate was turbulent. It was found that the convective heat flux in the stagnation zone was essentially constant:

$$q''_{b,conv} = 31.2\left(\frac{q_f}{l_j^2}\right)Ra_e^{-1/6}Pr_e^{-3.5} \quad (17)$$

for R < 0.16. The flux decreased with R in the wall jet region:

$$q''_{b,conv} = 1.46R^{-1.63}(q_f/l_j^2)Ra_e^{-1/6}Pr_e^{-3/5} \quad (18)$$

for R> 0.16 In both cases, $10^9 < Ra < 10^{14}$ and $Pr \cong 0.7$ The Rayleigh number was defined as

$$Ra = g\beta_e\, q_f l_j^2/\rho_e Cp_e v_e^3 \quad (19)$$

No correlations were given for the measured radiation heat flux. This accounted for up to 26% of the total heat flux. The radiant flux was specifically excluded from the above correlations.

3. Results and Discussion

There are the results shown, which distinguishes the difference between different parameters such as Reynolds number, H/d separation ratio, Equivlence ratio, Avg nusselt number etc. These factors shows the behavior of operating conditions while the jet impingement process is going on.

ISSN: 2278-0181
Vol. 2 Issue 8, August - 2013

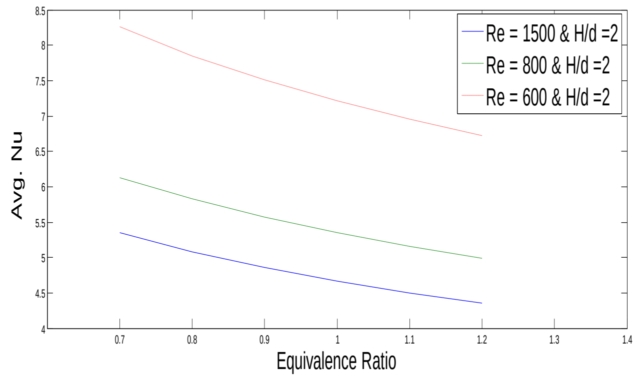

Fig:-3 The variation of Avg.Nusselt number with respect to Equivalence ratio for H/d = 2

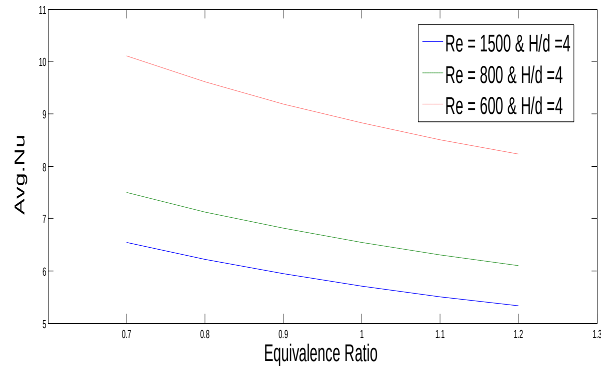

Fig:-4 The variation of Avg.Nusselt number. with respect to Equivalence ratio for H/d = 4

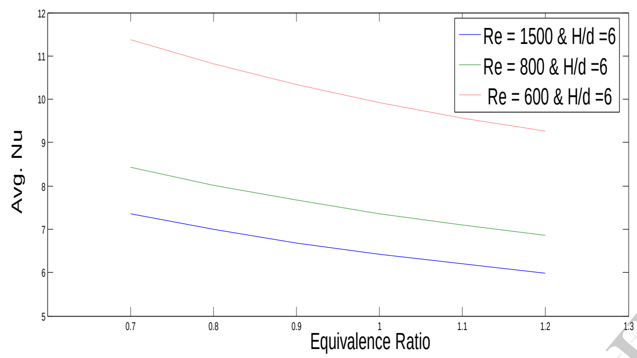

Fig:-5 The variation of Avg.Nusselt number with respect to Equivalence ratio for H/d = 6

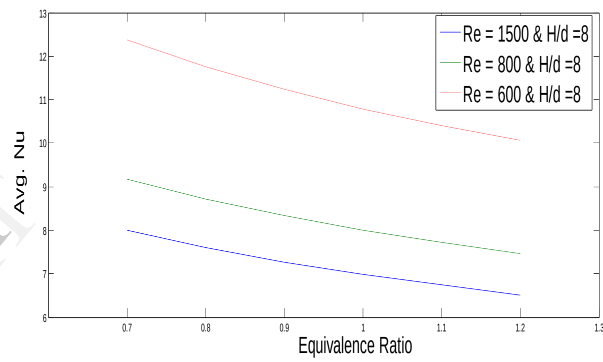

Fig:-6 The variation of Avg.Nusselt number with respect to Equivalence ratio for H/d = 8

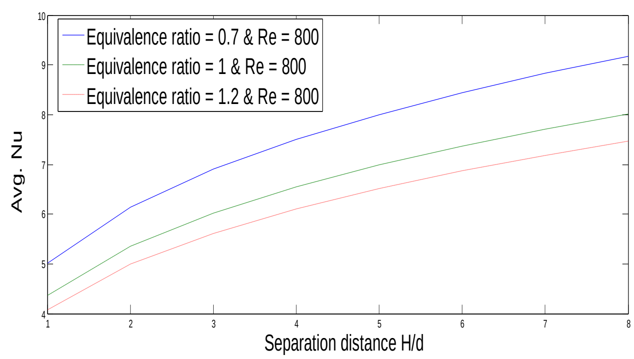

Fig:-7 The variation of Avg.Nusselt number with respect to Separation distance H/d & Equivalence ratio 0.7, 1.0 & 1.2 at constant Reynolds number =800

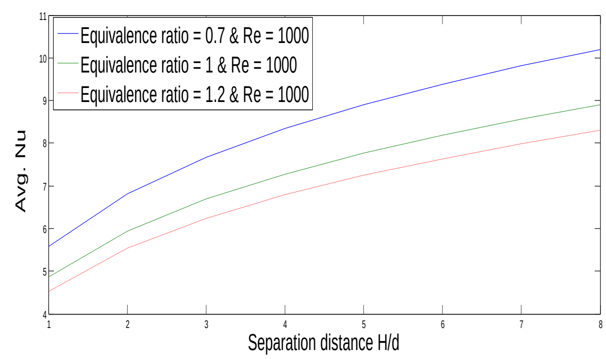

Fig:-8 The variation of Avg.Nusselt number with respect to Separation distance H/d & Equivalence ratio 0.7, 1.0 & 1.2 at constant Reynolds number =1000

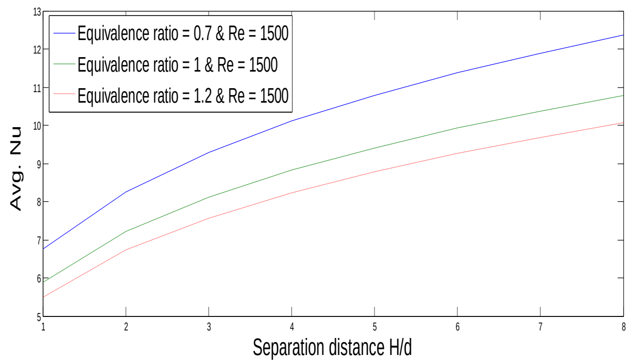

Fig:-9 The variation of Avg.Nusselt number with respect to Separation distance H/d & Equivalence ratio 0.7, 1.0 & 1.2 at constant Reynolds number =1500

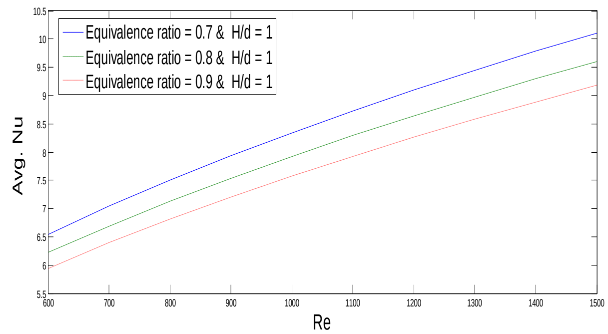

Fig:-10 The variation of Avg.Nusselt number with respect to Reynolds number for H/d = 1 & Equivalence ratio 0.7, 0.8 & 0.9

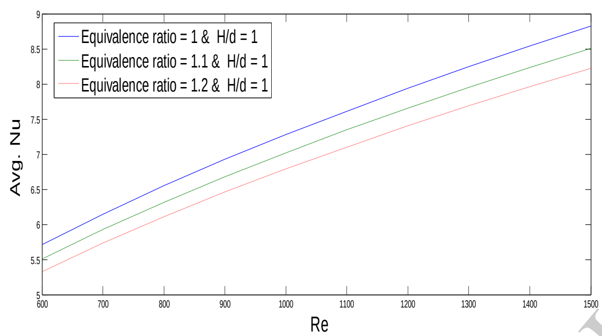

Fig:-11 The variation between Avg. Nusselt number and Reynolds number upto 1500 for H/d=1 & Equivalence ratio 1, 1.1, 1.2

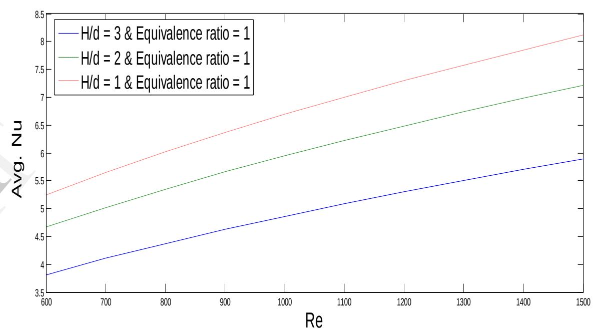

Fig:-12 The graph plotted between Avg. Nusselt number and Reynolds number for different H/d=1, 2, 3 & constant Equivalence ratio =1

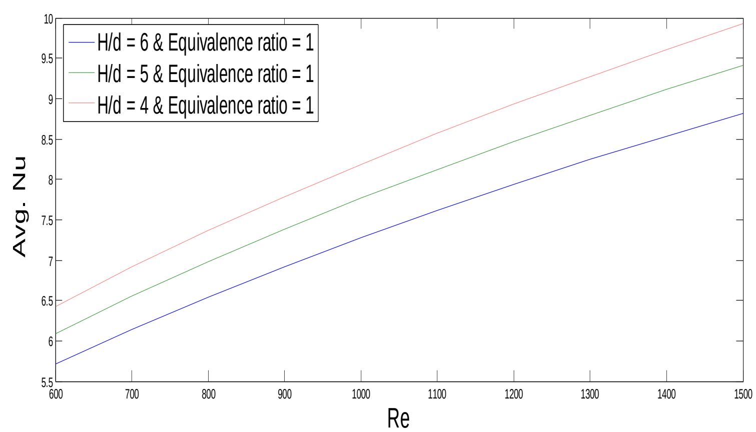

Fig:-13 The graph plotted between Avg. Nusselt number and Reynolds number for different H/d=4, 5, 6 & constant Equivalence ratio =1

The fig. no. 3 represents the variation of Avg. Nusselt Number versus equivalence ratio for reynold number 1500, 800 & 600 with ratio of distance between flame origin to the heated plate to the nozzle diameter as 2. For H/d = 2 the value of Avg. Nusselt Number enhances extremely to around 8.25. Avg. Nusselt Number decreases with increasing equivalence ratio for different reynold number. The value of Avg. Nusselt number increases with high Reynolds number.

The fig. no. 4 represents the variation of Avg. Nusselt Number versus equivalence ratio for reynold number 1500, 800 & 600 with ratio of distance between flame origin to the heated plate to the nozzle diameter as 4. Avg. Nusselt Number decreases with increasing equivalence ratio for different reynold number. The value of Avg. Nusselt number increases with high Reynolds number.

The fig. no. 5 represents the variation of Avg. Nusselt Number versus equivalence ratio for reynold number 1500, 800 & 600 with ratio of distance between flame origin to the heated plate to the nozzle diameter as 6. For H/d = 6 the value of Avg. Nusselt Number enhances extremely to around 11.5. Avg. Nusselt Number decreases with increasing equivalence ratio for different reynold number. The value of Avg. Nusselt Number increases with high Reynolds number.

The fig. no. 6 represents the variation of Avg. Nusselt Number versus equivalence ratio for reynold number 1500, 800 & 600 with ratio of distance between flame origin to the heated plate to the nozzle diameter as 8. For H/d = 8 the value of Avg. Nusselt Number enhances extremely to around 12.5. Avg. Nusselt Number decreases with increasing equivalence ratio for different reynold number. The value of Avg. Nusselt Number increases with high Reynolds number.

The fig. no. 7 represents the variation of Avg. Nusselt Number versus separation distance for Equivalence ratio 0.7,1.0 &1.2 with ratio of distance between flame origin to the heated plate to the nozzle diameter as ranged from 1 to 8. Avg. Nusselt Number increases with increasing separation distance for Reynolds number 800. The value of Avg. Nusselt number increases with low separation distance.

The fig. no. 8 represents the variation of Avg. Nusselt Number versus separation distance for Equivalence ratio 0.7,1 &1.2 with ratio of distance between flame origin to the heated plate to the nozzle diameter as ranged from 1 to 8. Avg. Nusselt Number increases with increasing separation distance for Reynolds number 1000. The value of Avg. Nusselt Number increases with low separation distance.

The fig. no. 9 represents the variation of Avg. Nusselt Number versus separation distance for 1Equivalence ratio 0.7,1 &1.2 with ratio of distance between flame origin to the heated plate to the nozzle diameter as ranged from 1 to 8. Avg. Nusselt Number increases with increasing separation distance for Reynolds number 1500. The value of Avg. Nusselt number increases with low separation distance.

The fig. no. 10 represents the variation of Avg. Nusselt Number versus reynold number for Equivalence ratio 0.7, 0.8 & 0.9 with ratio of distance between flame origin to the heated plate to the nozzle diameter as constant parameter 1. Avg. Nusselt Number increases approximately linearly with increasing Reynolds number from 600 to 1500

The fig. no. 11 represents the variation of Avg. Nusselt Number versus reynold number for Equivalence ratio 1, 1.1 & 1.2 with ratio of distance between flame origin to the heated plate to the nozzle diameter as constant parameter 1. Avg. Nusselt Number increases approximately linearly with increasing reynold number from 600 to 1500. As equivalence ratio increases the value of Avg. Nusselt number decreases.

The fig. no. 12 represents the variation of Avg. Nusselt Number versus Reynolds number for Equivalence ratio 3, 2 & 1 with ratio of distance between flame origin to the heated plate to the nozzle diameter as constant parameter 1. Avg. Nusselt Number increases approximately linearly with increasing Reynolds number from 600 to 1500. As equivalence ratio increases the value of Avg. Nusselt Number decreases.

The fig. no.13 represents the variation of Avg. Nusselt Number versus Reynolds number for Equivalence ratio 6, 5 & 4 with ratio of distance between flame origin to the heated plate to the nozzle diameter as constant parameter 1. Avg. Nusselt Number increases approximately linearly with increasing reynold number from 600 to 1500. As equivalence ratio increases the value of Avg. Nusselt Number decreases.

4. CONCLUSION

Flame impingement heat transfer is a very important area. A lot of work has been done, and much more is required to be done. Flame impingement on a flat surface is the most extensively studied configuration. Local and average heat flux data are available under different types of experimental and operating conditions. The results are very useful, as these help in optimizing the heating process. The effects of Reynolds number (laminar to fully turbulent), equivalence ratios (fuel lean to fuel rich) and separation distances are well understood. The effect of impingement of the cool central core of unburned gases has been studied extensively. The effect of different surface treatments on augmentation in the heat transfer value is quantified. The effect of using oxygen or oxy-enhanced air as an oxidizer is studied, and the results will be very useful for designing rapid heating flame impingement furnaces. Average Nusselt number curve falls down with increasing equivalence ratio for different Reynold number and separation distance. Average Nusselt number gradually inrceases with increasing separation distance for different Reynold number and equivalence ratio. Average Nusselt number increases in a linear manner with increasing Reynold number for different values of separation distance and equivalence ratio. Heat flux rate for all the fuels increases with an increase in velocity gradient. Ethane and Propane shows similar heat flux rate having a marginal difference. Natural Gas heat flux rate dominates Methane, Ethane, Propane, Acetylene and Hydrogen for flame impingement as a fuel. Oxygen also shows a good sign being used as an oxidizer.

5. SCOPE FOR FUTURE WORK

Regarding theoretical analysis of flame impingement heat transfer, an attempt has been going on to give a numerical solution and simulation of this heating process using some software. A number of semi-analytical and empirical correlations are available for different geometries and flow conditions. Further, it has been observed from the discussions on the various investigations that a number of issues are still to be explored. There is a need to generalize the experimental results. Studies that examined the flame impingement phenomenon theoretically are very few in number. In nearly all the previous studies reported, tests have been conducted when the flame is impinging perpendicularly to a flat surface. Other configurations studied are very few. Being of much industrial use, a detailed investigation is required for the local heat flux distribution under different operating conditions when the flame is striking normally or at an angle to cylindrical or spherical surfaces. Stagnation point calculations for heat transfer in these cases are required. Some more work is required in configurations like flames moving parallel to the plane surfaces and flames striking at an angle to plane surfaces. There are many other configurations, e.g., square, rectangular and triangular shaped bodies that are of much interest from an applications viewpoint but have not been studied so far. In most of the previous studies, generally, the tests were conducted with one burner diameter. Other burner diameters could be tested to determine the relevance of normalization. There is a paucity of information on how burner design features influence the heat transfer rates and, hence, the fuel economy. The effects of various burner types on the heat transfer to surfaces with various treatments should be tested. It has been observed that most of the studies employed round burner geometry. Only a few studies are available for slot burners. Other burner geometries have not been used. There are many possible combinations of fuel, oxidizer and equivalence ratio that have not been tested.

REFERENCES

[1] Jackson EG, Kilham JK. Heat transfer from combustion products by forced convection. Ind Eng Chem 1956; 48(11):2077–79.

[2] Woodruff IW, Giedt WH. Heat transfer measurement from partially dissociated gas with high Lewis number. J Heat Transfer 1966; 88:415–20.

[3] Milson A, Chigier NA. Studies of methane air flames impinging on cold plate. Combust Flame 1973;21:295–305

[4] R. Viskanta, M.P. Mengue, Radiation heat transfer in combustion systems, Prog. Energy Combust. Sci. 13 (1987) 97–160.

[5] Viskanta R. Enhancement of heat transfer in industrial combustion systems: problem and future challenges. In: Lloyd JR, Kurosaki K, editors. Proceeding ASME/JSME thermal engineering joint conference, vol. 5. New York: ASME; 1991. p. 161–73

[6] Van der Meer TH. Stagnation point heat transfer from turbulent low Reynolds number jets and flame jets. Exp Thermal Fluid Sci 1991; 4:115–26.

[7] R. Viskanta, Heat transfer to impinging isothermal gas and flame jets, Expt. Thermal Fluid Sci. 6 (2) (1993) 111–134

[8] Rigby JR, Webb BW. An experimental investigation of diffusion flame jet impingement heat transfer. In: Fletcher LS, Aihara T, editors. Proceedings of ASME/JSME thermal engineering joint conference 1995, vol. 3. New York: ASME; 1995. p. 117–26

[9] Baukal CE, Gebhart B. A review of flame impingement heat transfer part1: experimental conditions. Combust Sci Technol 1995; 104:339–57.

[10] Baukal CE, Gebhart B. A review of flame impingement heat transfer part 2: measurements. Combust Sci Technol 1995; 104:359–85

[11] C.E. Baukal, B. Gebhart, A review of empirical flame impingement heat transfer correlations, Int. J. Heat Fluid Flow 17 (1996) 386–396

[12] C.E. Baukal, B. Gebhart, A review of semi analytical solutions for flame impingement heat transfer, Int. J. Heat Mass Transfer 39 (14) (1996) 2989–3002.

[13] Milson A, Chigier NA. Studies of methane air flames impinging on cold plate. Combust Flame 1973;21:295–305

[14] Baukal CE, Farmer LK, Gebhart B, Chan I. Heat transfer mechanism in the flame impingement heating. In: Dolenc DA, editor. Proceedings of 1995, international gas research conference, vol. 2. Rockville, MD: Govt. Inst., Inc.; 1996. p. 2277–87.

[15] Baukal CE, Gebhart B. Surface condition effect on flame impingement heat transfer. Exp Thermal Fluid Sci 1997; 15:323–35.

[16] Baukal CE, Gebhart B. Heat transfer from oxygen-enhanced/natural gas flames impinging normal to plane surface. Exp Thermal Fluid Sci 1998; 16:247–59.

[17] Viskanta R. Convective and radiative flame jet impingement heat transfer. Int J Transport Phenom 1998; 1:1–15

[18] Zhang Y, Bray KNC. Characterization of impinging jet flames. Combust Flame 1999;116:671–4

[19] Dong LL, Cheung CS, Leung CW. Heat transfer characteristics of an impinging butane/air flame jet of low Reynolds number. Exp Heat Transfer 2001; 14:265–82.

[20] Dong LL, Cheung CS, Leung CW. Heat transfer characteristics of premixed butane/air flame jet impinging on an inclined flat surface. Heat Mass Transfer 2002; 39:19–26.

[21] Su A, Liu Y-C. Pulsation of impinging jet diffusion flames on the flat plate. In: Proceedings of PSFVIP-2, May 16– 19, 1999, Honolulu, USA. p. 1–6. S. Chander, A. Ray / Energy Conversion and Management 46 (2005) 2803–2837, 2835

[22] S. Chander, A. Ray, Flame impingement heat transfer: a review, Energy Convers. Manage. 46 (18–19) (2005) 2803–2837.

[23] M.J.Remie ,G. Sarner , M.F.G. Cremers , A. Omarane , K.R.A.M. Schreel, M.Alden, L.P.H. de Goey, Extended heat transfer relation for an impinging laminar flame jet to a flat plate, Int. J. Heat and Mass Transfer 51 (2008) 1854-1865

[24] G.K.Agrawal, Suman Chakraborty, S.K.Som , Heat transfer characteristics of premixed flame impinging upwards to plane surfaces inclined with the flame jet axis.(2010) 1899-1907