The Influence of Confinement on Local Heat Transfer Distribution Due to Impingement of Slot Air Jet on a Flat Surface.

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***Abstract*— An experimental investigation is carried out to study the heat transfer enhancement from a flat surface with axis symmetric detached confinement plates on nozzle due to normal impingement of slot air jet. a single slot where aspect ratio chosen as 25 with Reynolds number as 10000 and varying different Z/D_h's as 0.25-8.0.The local and average Nusslet number on the heated target plate is reported with various L_c/D_h's of confinement in range of 0.0- 27.23.**

***Index Terms*- Confined Slot Air Jet, heat transfer impingement, Nusselt Number ,Stagnation Region, ,.**

I. INTRODUCTION

Impinging jets are widely used in many industrial applications to enhance the coefficient for convective cooling, heating or drying. In addition the applications include tempering of glass plate, drying of textile, annealing of metal sheets, and paper products, and cooling of heated components in gas turbine engines and de-icing of aircraft systems, Computers and electronic instruments. The heat transfer characteristics and fluid flow of an impinging jet are affected from a number of parameters such as, the nozzle-to-plate spacing (Z/D_h), nozzle geometry and Reynolds number. The impinging jets are classified by their boundary as confined flow field, where the radial spread of the jet is bounded by a confinement plate, Confined geometry, has been investigated extensively in literature due to importance of industrial applications. The cooling of the gas turbine blade is performed using cool air often bled from the compressor.

II STRUCTURE OF IMPINGING JET

Unconfined jet impingement system is one in which jet after impingement on a surface is exited without confinement. Fig. 1 shows the flow field of an impinging jet on to an orthogonal plate. The flow structures of impinging axis-symmetric jet can be subdivided into three characteristic regions.

1. The free jet region
2. The stagnation flow region.
3. The wall jet region.

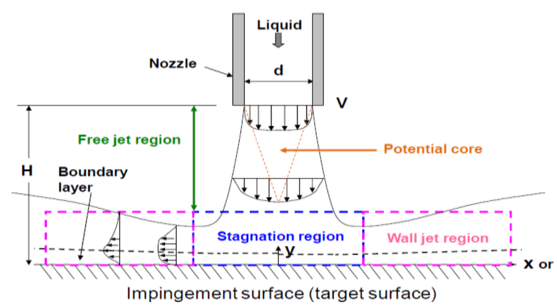

Fig. 1 Flow Regions for an Impinging Jet

The impinging jets travel from the nozzle exit as a free jet to within a distance 1.2 nozzle diameters from the target plate surface. Here, flow starts deceleration and static pressure increases due to conversion of kinetic energy of the jet. At stagnation region the Boundary layer of constant thickness is formed a radius of 1.1 nozzle diameter. In the axial component is decelerated at stagnation flow region the axial velocity and converts to an accelerated tangential one. The exit Reynolds Number conversely corresponding to relative limit layer thickness. Because of the quickened tangential stream changes to a decelerated wall jet stream, trade of energy with stagnant surroundings and wall friction rubbing. It is watched that the vacillations of speed of free jet are distrubuted along the wall jet region. Be that as it may, turbulence is an expansion in the wall jet region area relies on upon turbulence in the jet before impingement. The heat transfer rate in the locale of wall jet region is observed to be higher than stream wise over the plate.

III LITERATURE OUTLINE

Out lining important research trends on experimental investigations has been done to determine the structure of the submerged jet and various regions of jets and also conducted to determine the local distribution of the heat transfer impinging. On different geometry of strong surface recognizing potential gaps in learning are talked about underneath.

Robert Gardon and J.Cahit Akfirat [1965] studied on to determine the effect of heat transfer characteristics on impinging jets turbulence distributions in submerged jets. On heat transfer characteristics is notable that maximum in variation of stagnation point heat transfer co efficient with secondary peaks and nozzle to plate spacing distribution in radial direction of local heat transfer coefficients this phenomena specifically to slot jets.

K. Jambunathan et.al[1992],surveyed on Single Circular jet of Heat Transfer information that turbulent jets with spout way out of Reynolds number 5000-124000 and Nusselt number at radii more noteworthy than of six nozzle diameters from stagnation point. Jets issuing from square edged orifices openings give higher Jets exchange contrasted from elliptical and streams to circular jets and noticed that the surrounding temperature is not equivalent to air temperature at nozzle exit.

V. Narayanan et.al [2004] experimentally studied on submerged slot jets on flat plate. With hydraulic diameters of 0.5 and 3.5 which correspond to potential – core region and transitional region respectively. It is reported that, heat transfer coefficient of impingement region is peaked at that level and in wall bounded region it decreased monotonically.

E.Baydar, Y. Ozmen [2005] conducted experiments on the effect of higher Reynolds number for confined air impinging jets on flat plate and numerical investigated experimental values. They chose Reynolds number of 30000 and 50000 z/d in the range of 0.2-6.0. Deceleration of the jet occurs due to presence of impingement plate. At z/d up to 2 the sub atmospheric region is observed. As the nozzle exit to plate distance increasing the sub atmospheric pressure moves radially outward from stagnation point. Concluded that link occurred between heat transfer co-efficient, turbulence intensity and sub-atmospheric region.

M. Nirmalkumar, Vadiraj Katti and S.V. Prabhu [2011] experimented on the heat transfer behavior of a slot jet in the stagnation region ($0\leq x/b\leq2$), transition region ($2 \leq x/b \leq 5$) and wall jet region ($x/b \geq 5$). For a given z/b, heat transfer Coefficient increases with the increase in the Reynolds number in the stream wise direction. In a slot jet, secondary peak is not evident at lower Reynolds numbers and larger z/b s. and is strongly evident at maximum Reynolds number of 12,000 and for the $z/b\leq1$.

Vadiraj V. Katti, Adimurthy.M & Ashwini M Tamagond [2014] Experimentally Investigated the local distribution of wall static pressure coefficient due to impinging slot air jet on smooth and rough surface. Co-efficient of wall static pressure is seen to be independent of Reynolds number in range of 5000-20000 for a given jet to plate distance in case of a smooth surface. Wall Static pressure Co –efficient decreases with increases in jet to plate distances due to entrainment of surrounding quiescent air and looses effect of impact on the target plate. wall static pressure co-efficient is maximum at the stagnation point for all the configuration studied and decreases along the stream wise direction which may be attributed to increase in the velocity along the plate. It is observed that wall jet region starts around x/Dh of 1.5.

R.K. Brahma[1992] studied on slot jet impinges on a flat surface and predicted fluid flow and heat transfer charectertics at stagnation point by numerically correlated with Stanton number at Stagnation Point by keeping different nozzle to jet placing and Reynolds number. Results were obtained in terms of velocity gradient at stagnation point and compared the results with two dimensional jets on flat surface and velocity profile at nozzle outlet in parabolic shape.

IV EXPERIMENTAL SET UP

Schematic diagram showing various components of the experimental set-up are as shown in Fig. 2. For heat transfer studies measurements. Air is supplied by a blower through a calibrated orifice meter. The flow rate is control by a flow control valve of orifice meter and Venturimeter. To achieve the uniform flow rate at the outlet of the nozzle the airflow is directed to the plenum chamber through a diffuser and two meshes in the plenum. Velocity profile remains uniform because of the diffuser upstream of the plenum. Nozzle is made of an acrylic sheet with 45 mm height and 4 mm wide. The aspect ratio is maintained at nozzle outlet of 25. Temperature at the jet exit is measured using a Thermocouple (Chromel-Alumel - K-type. The output of the thermocouple is measured by milli voltmeter is used measure output of thermocouple. The target plate is kept on 2-D traverse table, so it is easily moved for various jets to plate spacing by operating the traversing system.

Air blower is used in the present work. It is capable of delivering air at volume 3.5 m3/min and weighs 1.7kg. Power rating of the blower is 600 watts. Set up consist of flow devices like orifice meter and Venturimeter, these are a device used to quantify the rate of air through the test rig as appeared in Fig. 2 and an Orifice is a circular plate, it is made up of copper coated M.S plate which has sharp circular edged hole called orifice. Its diameter is 10.3mm. The orifice meter is installed in the flow line of pipe diameter of 26.2mm. Orifice meter is calibrated using water as the fluid for the same Reynolds number as required with air flow. This is valid the length of flow is incompressible. The estimation of coefficient of release is evaluated 0.7. The Venturimeter is presented in the stream line parallel to the orifice meter to get the required flow rate i.e turbulent stream nature. It is comprised of copper covered M.S plate and its throat distance across is 12.6mm. Venturimeter is aligned parallel to the orifice meter for higher Reynolds number to get high flow rate.

The Jet-to-plate distances (Z/D_h) of 0.25, 0.5, 1.0, 2.0, 4.0, 6.0, &8.0 are considered in the present study. The confinement plate made of rectangular acrylic sheet (size ranging from 50×80-200×80 mm) in which a Slot is cut is attached at the nozzle exit. The heat transfer study on the target plate (155mm x 60mm x 0.06mm – stainless steel foil) is tightly clamped and stretched between copper bus bar. Approximately 5mm of the coil on either side is sandwiched in bus bars to ensure firm grip. Target plate assembly shown in fig4.1 because thinness of the foil surface provides constant heat flux .The target plate is heated by supplying power through regulated DC power supply. The “Matt completion Asian” paint which emissivity is (0.92) is painted on the back surface of warmer component.

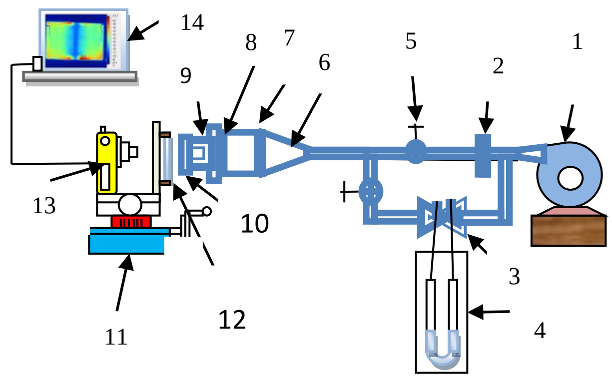

Fig. 2 Experimental set up for Heat transfer distribution on Confined Slot Air Jet.

1] Air blower 2] Orifice meter 3] Venturimeter 4] U-tube manometer 5] Control valves 6] Diffuser 7] Plenum chamber 8] Flow stream 9] Slot Nozzle 10] Confinement plate 11] Traverse table 12] Target plate 13] I R Camera 14] I R Image.

V METHODOLOGY

The stream of air from blower is passes through orifice meter or Venturimeter to diffuser. Air is controlled by a gate valve to set required Reynolds number of 10000. The confinement ratio is varied between 6.8 and 27.23 on slot jet. Confinement plate having a rectangular slot of size

60mm 12mm with thickness of 10mm is kept parallel to target plate. Legitimate arrangement of limited opening confined slot jet to target plate dividing with measuring of orthogonally of Z/D_h = 0.25-8. For clear perspective of picture from delicate I R camera is kept without miss coordinating of dark paint coated target plate. The thermal images captured are analyzed using SMART VIEW soft ware supplied with the device.

VI DATA REDUCTION

For given area of exit and inlet with the variation of limb height in manometer & density of water and air with constant co-efficient of discharge the Flow rate calculated as.

$$Q = C_d \frac{A_i A_O}{\sqrt{A^2{}_i - A^2{}_o}} \sqrt{2g\left[\frac{\rho_w}{\rho_{air}} - 1\right] h_{mano}} \ldots m^3/s$$

For rectangular shape of size of length and breadth the Hydraulic diameter is found to be as

$$D_h = \frac{4\times \text{Area}}{2\times perimeter} = \frac{(2\times a\times b)}{(a+b)} \quad \text{m}$$

For perfect voltage and current with the area and ambient temperature of medium the heat transfer co-efficient is estimated as

$$h = \frac{\text{V}\times\text{I}}{A(T_W - T_\infty)} \qquad \text{W/m}^2\text{K}$$

For given hydraulic diameter, velocity and density of medium with kinematic viscosity the Reynolds number is found to be as

$$\text{Re} = \frac{\rho V_j D_h}{\mu}$$

For given heat transfer coefficient with hydraulic diameter and thermal conductivity of material the Nusselt Number found as

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

VII RESULT'S & DISCUSSION.

Impact of confinement ratio and jet to- target distance on the heat transfer co-efficient distribution is analyzed. Jet widths extended from 0.6 mm to 2 mm, and jet to-target plate dispersing from 0.5 to 10. The plane Reynolds number went from 26.8 to 1000 in the laminar boundary. For the case with limit condition, for example, W = 2 mm, H = 5 mm and Re = 26.8 laminar boundary layer (Z.Q. Lou et al. [2005]). This result gives only 1% better performance for laminar flow keeping this in mind we tried for turbulent flow i.e. Re 10000.This understanding of note to be useful for confined slot air jet for Re 10000.

a] Un -Confined Jet

In this area, the jet to-target dispersing Z/D_h ranges from 0.25 -8.0 for unconfined jet at Re = 10000.Fig. 3 presents distribution of local Nusselt number along the objective target plate with variable Z/D_h. At the stagnation point, the local Nusselt number for Z/D_h = 0.25 is just about 1.5 times of that for Z/D_h = 8. The estimation of Nusselt number basically lies on heat transfer coefficient. Whenever Z/D_h shifts from 8.0 to 0.25 nearby Nusselt number increments significantly in the stagnation point as well as downstream region. Whenever Z/D_h is bigger than 1, the heat transfer coefficient move in the wall jet region is not influenced to a great extent by the lessening of Z. Certainly, a reduction of Z/D_h causes a sharp increase of Nusselt number in the core jet region.

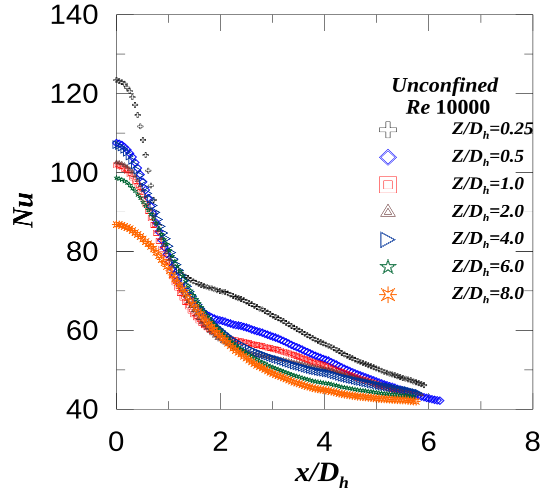

Fig. 3Local heat transfer distribution for Z/D_h =0.25-8.0 at *Re*=10000 and L_c/D_h =0.0

b] Confined Jet

Effect of nozzle to target spacing's: In this area, the jet to-target dispersing Z/D_h ranges from 0.25 -8.0 with L_c/D_h of 6.8 to 27.23 for Re = 10000. Fig. 4 presents distribution of local Nusselt number along the target plate with variation of Z/D_h. The general observation is, at the stagnation point, the local Nusselt number for Z/D_h = 0.25 is just nearly about 1.5 times of that for Z/D_h = 8. The estimation of Nusselt number basically lies on heat transfer coefficient. Whenever Z/D_h shifts from 8.0 to 0.25, the Nusselt number increases significantly at the stagnation point as well as downstream region. Whenever Z/D_h is bigger than 1, the heat transfer coefficient move in the wall jet region is not influenced to a great extent by the lessening of Z. Certainly, a reduction of Z/D_h causes a sharp increase of Nusselt number in the core jet region.

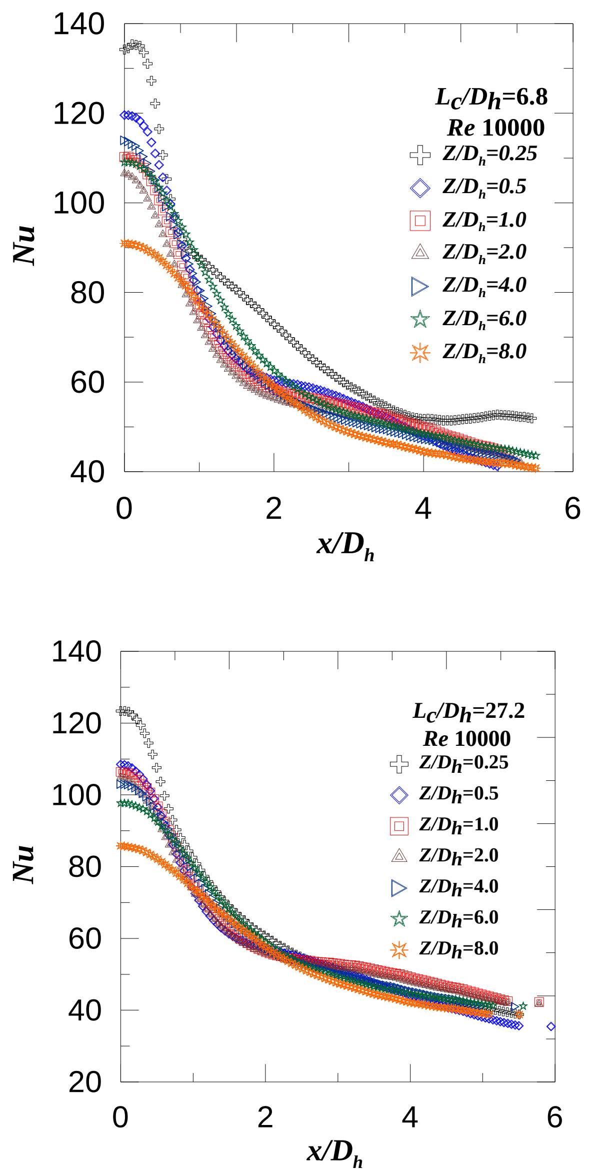

Fig. 4Effect of the Z/D_h ratio on Local heat transfer distribution for Confined Slot Jet size L_c/D_h =6.8 and L_c/D_h =27.23 at Re 10000

Effect of confinement ratios: The confinement ratio (Lc/Dh) used in the current study ranges from 6.8 to 27.23 whereas the Reynolds Number (Re) is kept constant at 10000. It is seen from Fig. 5 that the coefficient of heat transfer for different confinement ratio's for particular Z/D_h =0.25. As x/Dh decreases surface of Nusslet number increase at the stagnation point and higher the x/Dh Nusslet number at wall jet region decreases. As lower the x/Dh sharper core jet region. Here for confinement ratio of L_c/D_h of 6.8 heat transfer rate in stagnation region get maximum hence its preferred heat transfer rate at Z/D_h =0.25 and L_c/D_h =6.8 gives better thermal performance. By comparing the data from the Table I, it can be concluded that for the given Z/D_h =0.25 at Re = 10000, confinement ratio L_c/D_h of 6.8 gives +8.76% better thermal efficiency, L_c/D_h of 17.0 & 20.6 has good preferences but efficiency is less than L_c/D_h of 6.8. L_c/D_h of 13.6 and 27.23 has no influence on Nuo L_c/D_h of 10.2 & 23.8 gives reduction in Nusselt number in comparisons with the unconfined jet and are not preferable.

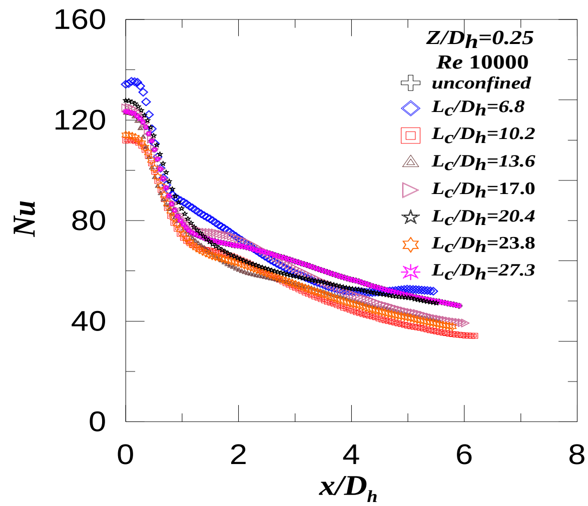

Fig. 5 Effect of the L_c/D_h ratio on Local heat transfer distribution for Confined Slot air Jet for Re 10000 at Z/D_h at 0.25.

Table I Details for influence of confinement.

Unconfined slot air jet Nu_o =123.41 at Z/D_h = 0.25 for Re=10000

L_c/D_h	Nu_o	Percentage of Nuo on Confined Slot Air Jet			
6.8	134.2	+	08.7	%	Better preferences
10.2	111.8	-	09.3	%	Not preferred
13.6	122.7	-	00.5	%	No influence
17.0	124.8	+	01.1	%	Good preferences
20.6	127.0	+	03.6	%	Good preferences
23.8	114.2	-	07.0	%	Not preferred
27.2	123.7	-	00.0	%	No influence

The confinement proposition is ranges from 6.8 to 27.23. Other parameters Re = 10000 were kept constant. It is seen from Fig. 6. that the coefficient of heat transfer for different confinement ratio's for particular Z/D_h =0.5 and as the x/Dh decreases surface at Nusslet number increment at close to the stagnation point and higher the x/Dh Nusslet number at wall jet region decreases. As lower the x/Dh sharper the potential core region. Here for confinement ratio of L_c/D_h of 17.0& 6.8 heat transfer rate in core region get maximum. Hence the preferred heat transfer rate at Z/D_h =0.5 the L_c/D_h =6.8 gives better thermal performance. In wall jet region heat transfer co-efficient at unconfined and L_c/D_h =17.0 at Z/D_h =0.5 gives maximum thermal performance. By comparing the data from the Table II, it can be concluded that for the given Z/D_h =0.5 at Re = 10000, confinement ratio L_c/D_h of 20.6 gives +11.47 % better thermal efficiency, L_c/D_h of 6.8, 13.6, 17.0 & 27.2 has good preferences but efficiency is less than L_c/D_h of 20.6. L_c/D_h of 13.6 and 27.23 has no influence on Nuo. L_c/D_h of 10.2 & 23.8 gives reduction in Nusselt number in comparisons with the unconfined jet and are not preferable.

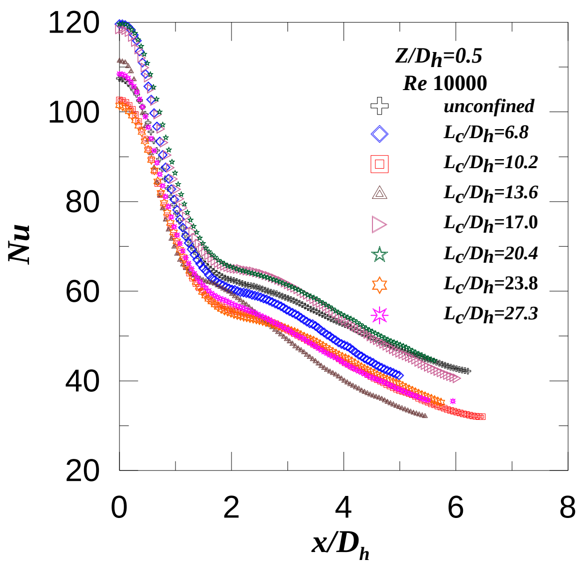

Fig. 6Effect of the L_c/D_h ratio on Local heat transfer distribution for Confined Slot air Jet for Re 10000 at Z/D_h at 0.5.

Table II Details for influence of confinement.

Unconfined slot air jet Nu_o =107.29 at Z/D_h = 0.5 for Re=10000

L_c/D_h	Nu_o	Percentage of Nuo on Confined Slot Air Jet			
6.8	119.5	+	11.43	%	Good preferences
10.2	102.6	-	04.33	%	Not preferred
13.6	111.3	+	03.80	%	Good preferences
17.0	118.4	+	10.36	%	Good preferences
20.6	119.6	+	11.4	%	Better preferences
23.8	101.4	-	05.4	%	Not preferred
27.2	108.4	+	01.1	%	Good preferences

The confinement proposition is ranges from 6.8 to 27.23. Other parameters Re = 10000 were kept constant. It is seen from Fig. 7that the coefficient of heat transfer for different confinement ratio's for particular Z/D_h =1.0 as x/Dh decreases surface of Nusslet number is build in the stagnation region and higher the x/Dh Nusslet number at wall jet region decreases. As lower the x/Dh sharper the potential core region. Here for confinement ratio of L_c/D_h of 17.0heat transfer rate in core region get maximum hence its preferred heat transfer rate at Z/D_h =1.0 and L_c/D_h =17.0 gives better thermal performance. In region of wall jet the heat transfer co-efficient at L_c/D_h =17.0 gives maximum thermal performance. By comparing the data from the Table III ,it can be concluded that for the given Z/D_h =1.0 at Re = 10000,confinement ratio L_c/D_h of 17.0 gives +20.12 % better thermal efficiency, L_c/D_h of 6.8, 13.6, 20.6 & 27.2 has good preferences. .

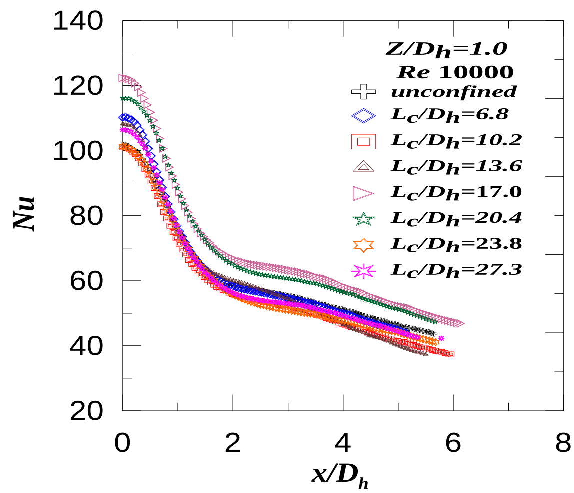

Fig. 7Effect of the L_c/D_h ratio on Local heat transfer distribution for Confined Slot air Jet for Re 10000 at Z/D_h at 1.0.

Table III Details for influence of confinement.

Unconfined slot air jet Nu_o =101.82 at Z/D_h = 1.0 for Re=10000

L_c/D_h	Nu_o	Percentage of Nu_o on Confined Slot Air Jet.			
6.8	110.0	+	08.2	%	Good preferences
10.2	101.0	-	00.7	%	Not preferred
13.6	108.0	+	06.1	%	Good preferences

17.0	122.3	+	20.1	%	Good preferences
20.6	116.0	+	13.9	%	Good preferences
23.8	101.2	-	00.5	%	Not preferred
27.2	106.4	+	04.5	%	Good preferences

By comparing the data from the Table IV ,it can be concluded that for the given Z/D_h =2.0 at Re = 10000,confinement ratio L_c/D_h of 20.6 gives +15.30% better thermal efficiency, L_c/D_h of 6.8, 13.6, 23.8 &27.2 has good preferences. L_c/D_h of 10.2 gives reduction in Nusselt number in comparisons with the unconfined jet and is not preferable.

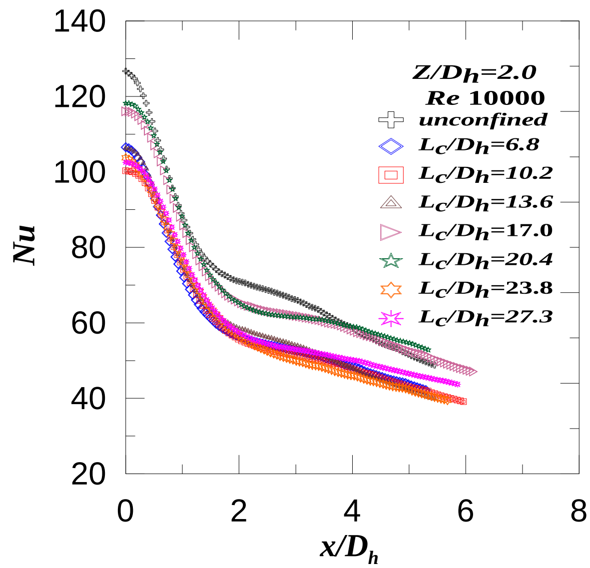

Fig. 8Effect of the L_c/D_h ratio on Local heat transfer distribution for Confined Slot air Jet for Re 10000 at Z/D_h at 2.0.

Table IV Details for influence of confinement.

Unconfined slot air jet Nu_o =102.55 at Z/D_h = 2.0 for Re=10000

L_c/D_h	Nu_o	Percentage of Nu_o on Confined Slot Air Jet.			
6.8	106.5	+	03.90	%	Good preferences
10.2	100.2	-	02.20	%	Not preferred
13.6	106.1	+	03.40	%	Good preferences
17.0	116.0	+	13.10	%	Good preferences
20.6	118.2	+	15.30	%	Good preferences
23.8	103.7	+	01.14	%	Good preferences
27.2	104.9	+	02.33	%	Good preferences

VII. CONCLUSION

Heat transfer co-efficient distribution on a flat surface Impinged by a confined slot air jet are experimentally analyzed for the confinement ratios, Lc/Dh=0.0 to 27.2 and nozzle slot jet to plate spacing (Z/D_h =0.25-8.0) at Re= 10000. Following conclusions elaborated from the present study:

- Heat transfer distribution and Nusselt number at Re 10000 is to be found to be symmetrically for confined and unconfined slot air slot air jet.
- There is a continuous enhancement in Nusselt number at stagnation point for the L_c/D_h =0.0-27.23 is found for unconfined and confined.
- The heat transfer is sharper at core jet region for unconfined and confined.
- The heat transfer is not influence on higher the jet to plate spacing for unconfined and confined.
- At wall jet region heat less compared to core –region for unconfined and confined slot air jet.
- For confinement ratio of L_c/D_h =6.8 gives good preference for all Z/D_h ‘s.

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