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Influence of Jet to Plate Spacings and Reynolds Number at Stagnation Point on the Heat Transfer Distribution between a Smooth Plate and Impinging Air Jet

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Abstract—*Experiments are performed to observe the effects of nozzle to plate spacings (z/d_e) on heat transfer distribution due to air jet impingement of on a test rig made of smooth and flat plate. In this study, local wall temperature is measured by using thin metal foil technique. Influence of wall temperature (25–125°C) on local heat transfer and effectiveness is studied for different Reynolds numbers (23000-30000) and nozzle to plate spacing(1–10) for circular jets. Nusselt number measured based on equivalent diameter of circular nozzle. The experiment consists of a reciprocating air compressor for delivering compressed air and a plate of dimension 150mm x 120mm x 0.08mm that is heated uniformly. 300 mm long aluminium tube of 300 is used as a circular nozzle to get a fully developed flow. The cold jets of different Reynolds numbers (23000-30000) is used to impinge on the flat test rig of dimension 150mm x 120mm x 0.08mm. Variation in Nusselt's number is analysed over the test rig in the different z/d_e ratio. The main aim of study is to observe the changes in Nusselt number for various values of z/d_e. Experimental analysis for study of changes in heat transfer by using 10 mm round nozzle at z/d_e = 1, 2, 4, 6, 8 and 10 and for the plate angles zero for different Reynolds numbers. From the experiment the maximum Nusselt number is observed at r/d_e =0, where r/d_e is the radial distance from stagnation point on flat rig. The maximum Nusselt number on the flat rig is observed when the ratio of nozzle to plate spacing (z/d_e) is 6. When the angle of test rig is zero degree then the nature of Nusselt number distribution became symmetric.*

Keywords*: Nusselt number, Nozzle to plate spacings, Radial distance from stagnation point*

Nomenclature

A	exit area of the nozzle, mm²
d_e	equivalent diameter, mm, $d_e = \sqrt{(4A/\pi)}$
h	heat transfer coefficient, W/m² K
I	current, Ampere
k	thermal conductivity of air, W/m K
N_u	Nusselt number, hd_e/k
N_{uo}	Nusselt number at the stagnation point
p	perimeter, mm
Q_1	total heat supplied, W/m²
Q_α	heat carried out by convection through impinging jet, W/m²
Q_ε	heat carried out by radiation from the plate, W/m²
Q_k	heat carries out by radiation from the back side, W/m²
r	radial distance from the stagnation point, m
R_e	Reynolds number,
T_j	jet temperature, ºK
T_{aw}	adiabatic wall temperature, ºK
T_w	wall temperature, ºK
T_∞	ambient temperature, ºK
v	velocity, m/s
V	voltage, V
z	jet to plate distance, mm
μ	viscosity of fluid, Pa s
ρ	density of fluid, kg/m³

1. INTRODUCTION

The world is now progressing towards the advanced and modern techniques of heat transfer in different fields of mechanical, electrical/electronics, aeronautical engineering, etc. Today such advanced technologies require large heat emissions and cooling of such devices without affecting their efficiencies has become a challenge. Numerous researches have been done in this field to improve the cooling rate such of heat generating devices. Cooling of microprocessor chips of all modern electronic devices such as laptops/computers is the best example today. In turbine applications, impinging jet flows may be used to cool several different sections of the engine such as the combustor case, combustor can walls, turbine case/liner, and the critical high temperature turbine blades.

Jet impingement is illustrated in Fig.1.1. Forced Fluid (liquid or gas) from fan or blower or pump comes as a jet through nozzle or pipe or duct and hits the target (surface form which heat is transferred). This is called jet impingement.

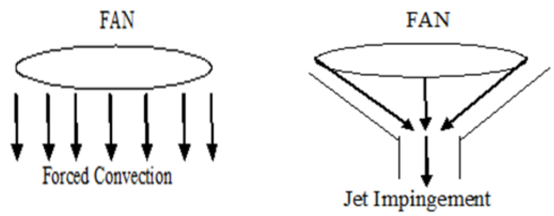


Fig. 1.1: Forced convection to jet impingement

A jet is a stream of fluid that is projected into a surrounding medium, usually from some kind of a nozzle, aperture or orifice by converting part of supplied pressure energy into kinetic energy. Jets can travel longer distance without dissipating. A jet fluid has higher momentum compared to the surrounding fluid medium. In the case where the surrounding medium is assumed to be made up of same fluid jet, and this fluid has a viscosity, then the surrounding fluid near the jet is assumed to be carried along with the jet by a process called Entrainment.

A jet may be classified as free jet or an impinging jet. A free jet is the one in which the decay of velocity takes place in the surroundings of the jet without obstruction in its path.

A Jet which is directed on to a target surface and the decay of velocity takes place partially in the surroundings, partially due to flow over the surface is called an impinging jets. Thus, in a jet flow either fluid-fluid or fluid surface interaction or both occur. During fluid-fluid interactions a shear layer is formed in the interaction region due to difference in velocity gradients in the fluid(s) and during fluid-surface interaction there is formation of boundary layer. This gives rise to greater mass, momentum and heat transfer between the fluid-fluid and the fluid-surface.

2. LITERATURE REVIEW

Jambunathan et al. [1] and Viskanta [2] have done experimental studies on flow visualisation. They found changes in heat transfer for different nozzle shapes and nozzle-to plate spacings (z/d_e) for different Reynolds number in the range of 5000 to 12,400. They used single nozzle at a time in their experiment. Lytle and Webb [3] stated experimental investigation on heat transfer for air jet impingement low jet exit to plate spacing. They calculated heat transfer by using thermal images captured by thermal imaging camera (infrared camera). They also explained flow structure with pressure measurements and laser Doppler velocimetry. Lee and Lee [4], Garimella and Nenaydykh [5], Gulati et al. [6], Gao et al. [7], Lee et al. [8] and Zhao et al. [9] have reported in studies the effects on jet impingement heat transfer using the different nozzle geometry using measurement as well as numerical simulations techniques. Katti and Prabhu [10] have investigated experimentally the effect on the convective heat transfer coefficient for various affecting parameters like Reynolds number, nozzle to plate spacings, turbulent intensities under different practical conditions and their outcome are well received in research. However, in most of the above investigations, the impinging jet is at the same temperature as that of the surroundings. Bouchez and Goldstein [11] measured effectiveness and local heat transfer over the test area by a single circular jet interacting with cross flow for two different nozzle to plate spacings (z/d_e = 6, 12) for a range of mass flow rates (Reynolds number) from 10^3 to 10^6. It is reported that the impingement effectiveness depends on blowing rate and decreases with increase in blowing rate. Striegel and Diller [12, 13] carried out an analytical analysis and conducted experiments. Katti and Prabhu [10] have experimentally studied the effects of entrainment temperature on the local heat transfer using multiple impinging turbulent air jets for the Reynolds number 4000 to162,000. The jet issued in the environment at a temperature which was varied between the initial temperatures of jet to the temperature.

Nakod et al. [14] also observed experimentally the variation on heat transfer distribution and nusselt number for different nozzle-to-plate spacings 0.5, 1, 2, 4, 6 and 7 and they found maximum heat transfer is observed at the stagnation point and heat transfer decreases monotonically in the radial direction for all jet-to-plate-spacings. They also observed that at stagnation point heat transfer is maximum for z/d_e of 0.5, and then it reduces as the z/d_e is increased and again increased in the vicinity of z/d_e of 6 and further reduction in the stagnation point heat transfer is observed at higher z/d_e's.

Vinze et al. [15] are also determined experimentally the variation of heat transfer on different nozzle to plate spacings (i.e. z/d_e = 1, 2, 4, 6, 8 and 10), different jet temperature for different nozzle geometry and they determined the heat transfer increases with increase in Reynolds number. The maximum heat transfer is achieved at approximately 6 $z/d_{e,}$ irrespective of jet temperature. The objective of present investigations is to study the effect of z/d_e (i.e. 1, 2, 4, 6, 8 & 10) on the Nusselt number (i.e. heat transfer distribution) and also effect of Reynolds numbers ranging from 23000 to 30,000 on heat transfer distributions at stagnation points.

3. EXPERIMENTAL SETUP

The layout of the experimental setup is indicated in Fig. 3.1. Air is supplied by an air compressor (capacity 27.58 bar at 50 g/s) through a calibrated venturi meter. Air filter is fitted on upstream line of venturi meter to filter the moisture, dust content of air and pressure regulator are also fitted on upstream line of the venturi meter to maintain the mass flow rate.

Two needle valves are fitted for regulating and controlling the flow rate. One is fitted at upstream side and another at downstream side of the Venturi meter. A constant Reynolds number is determined by varying the mass flow rate with respect to the jet temperature. The Reynolds number can be calculated as

$R_e = 4m/\pi\mu d_e$ (3.1)

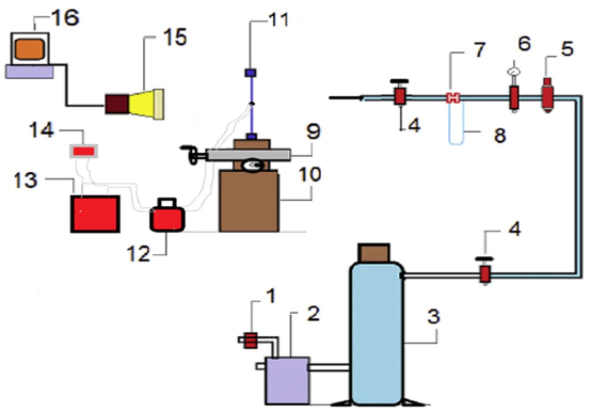


Fig.3.1: Layout of experimental set-up: (1) Air filter, (2) Air compressor, (3) Air receiver, (4) Needle valves, (5) Air filter, (6) Pressure regulator, (7) Orifice, (8) Differential manometer, (9) Traverse system, (10) Table, (11) Impingement assembly, (12) Variac, (13) Transformer, (14) Power Supply, (15) Infrared camera, (16) Computer.

Since the pipe nozzle is circular therefore equivalent diameter will be same as 10 mm. The length to diameter ratio for pipe is used 80 for ensuring fully developed turbulent flow at the nozzle exit for the Reynolds number range for 23000 to 30,000. The target plate (150 mm x 120 mm; 0.08 mm thick stainless steel foil) is clamped rigidly between two copper bus bars. Approximately 5 to 7 mm of the foil on either side is sandwiched in the bus bars to ensure the fine grip. Lateral conduction is negligible due to thinness of foil and surface gives constant heat flux situation as observed by Lytle and Webb [3]. Contour of thermal images are obtained from the thermal imaging camera kept on one side of the target plate opposite to the nozzle-side. A finned heating arrangement is installed using electric energy to heat up the flat test rig to get the required plate temperature. The voltage supplied to test plate is controlled by variac.

The local temperature is measured by thermal imaging camera from uniform heat flux surface painted black using a thin coat of high temperature paint 'Pyromark' (for higher emissivity of order of 0.996) which provides more spatial resolution of temperature than thermocouples. Power is supplied from AC transformer, a voltage stabilizer, and a variac. The voltage and current of the test rig are measured by two digital multimeters using simultaneously. Appropriate voltage taps fitted in each of the bus bars. To obtain different nozzle to plate spacings a traverse system is used. Power losses from the exposed surface of the test plate due to natural convection and radiation is estimated experimentally and is considered in the calculation of Nusselt number. The influence of nozzle to plate spacings on heat transfer distribution is studied by conducting experiments on round nozzle. Due to experimental setup limitations the jet temperature for 30,000 Reynolds number jets is limited to 125°C.

4. EXPERIMENTAL PROCEDURE

The experimental setup have been designed and fabricated to understand the convective heat transfer on flat test rig, subjected an impinging circular jet on a heated test plate.

4.1 Calculation of Reynolds Number

The pressure difference in U-tube manometer attached in venturi meter = H

$(P_1-P_2) = \rho \times g \times H$

Apply Bernoulli's theorem,

$\Delta P = ½ \{ \rho (v_2^2- v_1^2)\}$ (4.1.1)

Form continuity Equation

$a_1v_1 = a_2v_2$

$v_1 = v_2 (a_2/a_1)$

Substitute value of v_1 in equation (2) we get,

$\Delta P = ½ [(\rho v_2^2) . \{1-(a_2/a_1)^2\}]$

Solving the above equation, we get velocity of air of pipe,

$v_2 = \sqrt{(2 \Delta P / \rho)} . 1/\sqrt{\{1-(a_2/a_1)^2\}}$ (4.1.2)

Volumetric flow rate,

$Q = a_2v_2$

$Q = \{\sqrt{(2 \Delta P / \rho)}\} . a_2/\sqrt{\{1-(a_2/a_1)^2\}}$ (4.1.3)

Line area, $a_1= \pi d_1^2/4$

Throat Area, $a_2 = \pi d_2^2/4$

Substituting the value of a_1 and a_2 in equation (4), we get

$Q = (\pi d_2^2/4).\sqrt{(2 \Delta P / \rho)} /\sqrt{\{1-(d_2/d_1)^4\}}$

Since beta ratio, $\beta = (d_2/d_1)$ so

$Q = (\pi d_2^2/4).\{\sqrt{(2 \Delta P / \rho)} /\sqrt{(1- \beta^4)}\}$

The small amounts of energy converted into heat within viscous boundary layers tend to lower the actual velocity of real fluids somewhat. A discharge coefficient C_d is typically introduced to account for the viscosity of fluids

$Q = C_d(\pi d_2^2/4).\{\sqrt{(2 \Delta P / \rho)} /\sqrt{(1- \beta^4)}\}$

The mass flow rate can be found by multiplying Q with the fluid density,

$m = \rho Q$

$m = \rho[C_d(\pi d_2^2/4).\{\sqrt{(2 \Delta P / \rho)} /\sqrt{(1- \beta^4)}\}]$ (4.1.4)

The Reynolds number can be calculated by,

$R_e = 4m/\pi\mu d_e$ (4.1.5)

The above equation is derived for venturi meter carrying a liquid as described in ISO (1991) and in ASME (1971). Since the surface of venturi meter is machined finish, therefore value of discharge coefficient C_d is considered as 0.99 for calculations.

4.2 Calculation of Nusselt's Number

Total heat supplied = (Voltage) x (Current) Watt

Applying the energy balance over the mental sheet at steady state condition, the electric power Q_1 is transferred by three known heat transfer mechanism and can be written in the following form;

$Q_1=Q_\alpha+Q_\varepsilon+Q_k$ (4.2.1)

Where $Q\alpha$ is a heat transfer by convection, Q_ε is the heat transfer by radiation and Q_k is conduction heat transfer through the metal plate thickness, t. Because the heat source the uniformly distributed over the flat metal plate as explained before, therefore electric power can be calculated by,

$Q_1 = I^2 R = I . V$

Radiation and conduction are considered as the heat losses and calculated separate experimentally

$Q_\alpha = Q_1 - (Q_\varepsilon + Q_k)$

The heat transfer by forced convection was caused by the flow impinging against metal surface and was defined by,

$Q_\alpha = h.A\ (T_w - T_{ad})$ (4.2.2)

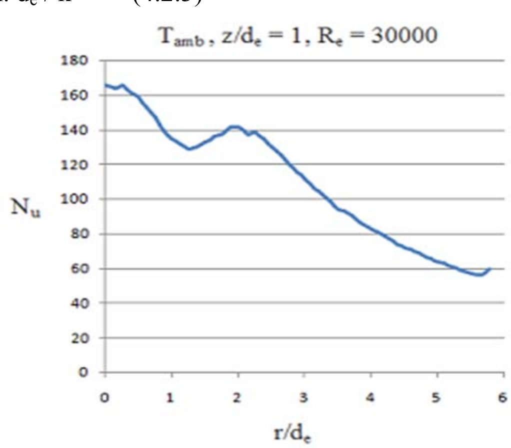
The temperature difference (T_w - T_{ad}) consists of bottom side plate temperature T_w and reference temperature which can be defined as the adiabatic wall temperature T_{ad}.

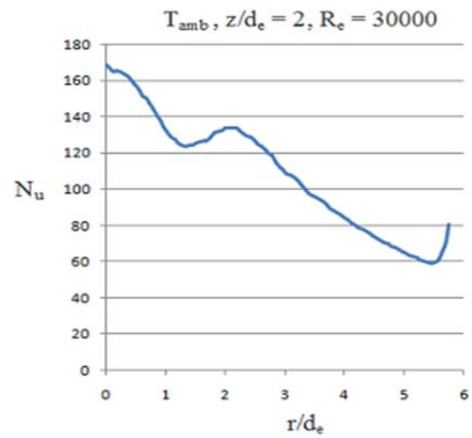
Since adiabatic wall temperature is much closed to the exit air temperature,

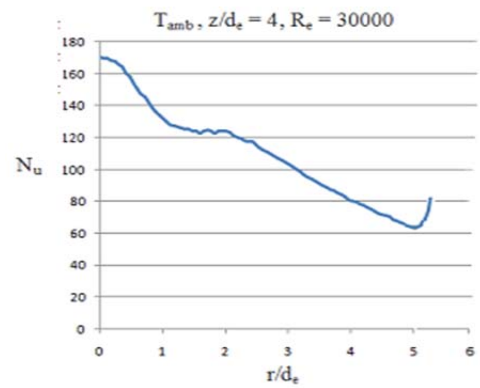
$T_{ad} \approx T_j$

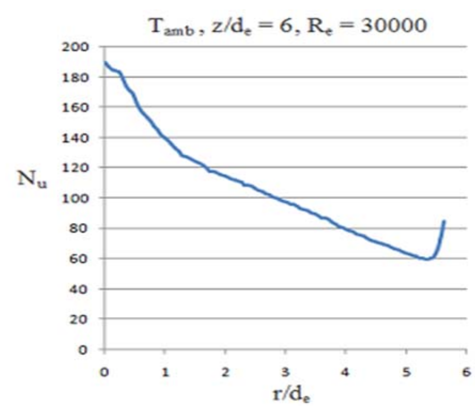
Nusselt Number can be calculated by:

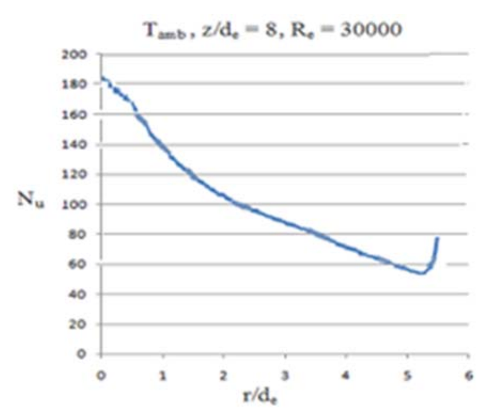
$N_u = h.\ d_e / k$ (4.2.3)











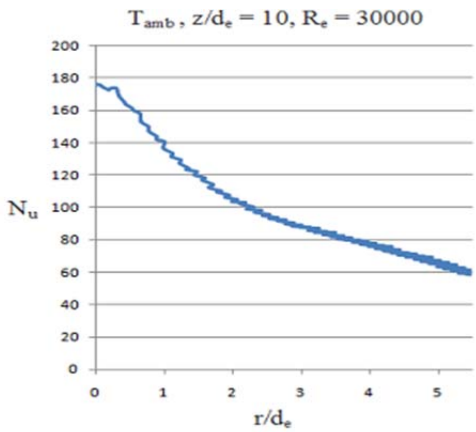


Fig. 4.2: Variation in N_u with r/d_e for different z/d_e

5. RESULTS AND DISCUSSIONS

The present work comprises of experimental investigation of local heat transfer characteristics in an impinging single round jet cooling system on a flat test rig surface, with compressed air flow through jet on test rig. The investigations include different ratio of jet to plate spacings and different Reynolds number ranging from 23000 – 30000 for different radial distances. The observation made and results obtained during the experiments are given below:

5.1. Influence of z/d_e on the Nusselt number distribution

Fig.5.1 shows the Nusselt number variation in radial direction from the stagnation point for different nozzle to plate spacings of 1, 2, 4, 6, 8 and 10. For given nozzle to plate spacing, maximum heat transfer is measured at the stagnation point and heat transfer reduces continuously in the radial direction for all nozzle plate spacings. It is interesting to observe that stagnation point heat transfer, as shown in Fig. 5.1, is maximum at z/d_e of 6 and then it reduces as the z/d_e is increased.

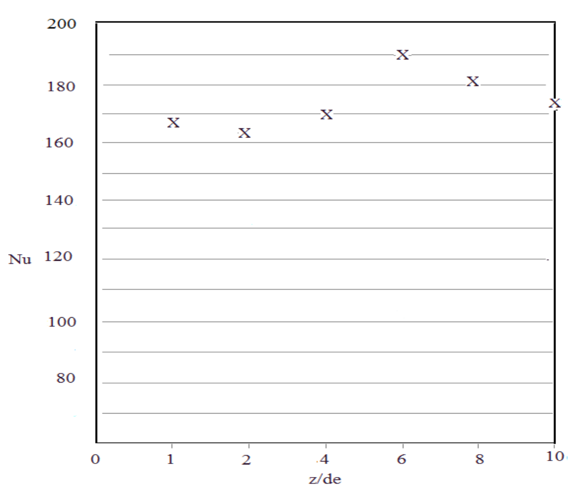


Fig. 5.1: Stagnation point N_u variation with z/d_e

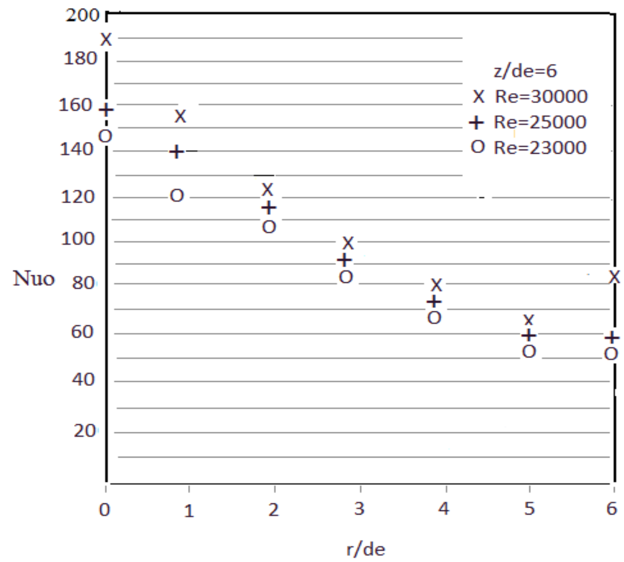


Fig. 5.2: Variation in N_u with r/d_e at a z/d_e=6 for different Reynolds number

5.2. Influence of Reynolds number on the Nusselt number distribution

Fig. 5.2 shows the variation in the Nusselt number in radial direction for different given Reynolds number (23000, 25000 and 30000) at z/d_e = 6. In general, it is seen that heat transfer rate increases with the increase in Reynolds number at all the radial locations. But, increase in heat transfer rate is more at the stagnation point as compared to any other given radial location for a given increase in the Reynolds number. In the plot it is clearly observed that with increase in Reynolds number from 25000 to 30000, the stagnation point Nusselt number increases from 158 to 190 (i.e. 20.25%) whereas at radial location corresponding to r/d_e of 6, the increase in Nusselt number is from 58 to 86 (i.e. 48.26%). This suggests that the stagnation point heat transfer is stronger function of Reynolds number than wall jet heat transfer.

6. CONCLUSION

The influence of nozzle to plate spacings has been observed experimentally for three Reynolds numbers 23000, 25000 and 30000. To investigate influence of nozzle to plate spacings on Nusselt number and also variation in Nusselt number on three Reynolds numbers 23000, 25000 and 30000 for z/d_e=6. The following are the main conclusions that comes from experimental study.

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