Australian Journal of Basic and Applied Sciences, 10(14) September 2016, Pages: 172-184

AUSTRALIAN JOURNAL OF BASIC AND APPLIED SCIENCES

ISSN:1991-8178 EISSN: 2309-8414
Journal home page: www.ajbasweb.com

Enhancement of Heat Transfer in a Gas Turbine CombustorUsing Jet Impingement on a Flat Plate with Triangular Ribs

[1]Hussain S. Abdand [2]Arkan AlTaie

University of Technology, Baghdad-Iraq

Address For Correspondence:
Hussain S. Abd, University of Technology, Baghdad-Iraq
E-mail: hussainsaad1983@gmail.com

ARTICLE INFO

Article history:
Received 26 July 2016 Accepted 21 September 2016 Published 30 September 2016

Keywords:
jet impingement, gas turbine, triangular ribbed target, laterial ribs.

ABSTRACT

Development of combustor material and / or enhancement of liner cooling are drawing much attention. These will increase power and / or combustor life. The aim of this work is to investigate cooling enhancement by employing triangular ribs shape. In present work, triangular ribbed target surface (laterial arrangement) has been proposed to increase the effectiveness of wall cooling in order to investigate impingement cooling enhancement. To apply this idea, 36 holes with diameter of 5mm for each hole were punched and arranged in inline array with spacing between the jet of 4 times the jet diameter. Reynolds numbers was taken in the range 5000 to 15000 While, ratio of height to diameter of jets that represents the distance between the jet plate and target plate are 2 and 3 times the hole diameter. A multiple jet holes with ribbed target plate having back side resistive film has been employed for the experimental model of the heat transfer process. Two target plate models have been used which are, a target surface without ribs that represent the base line case as a first model while, a laterial target surface with triangular ribs rows arrayed at45^0 as a second model. Then, estimation of the average wall cooling effectiveness and coefficient of the heat transfer (Nusselt numbers) have been conducted for each model. The results revealed that effectiveness of the discharge coefficient with the Reynolds number and the jet spacing superior of the second motel to enhance coefficient of heat transfer and effectiveness of wall cooling. Where, the highest enhancement was20.41%,17.64%at$\frac{H}{D}=2$ and 18.11%,12.95% at$\frac{H}{D}=3$ for coefficient of the heat transfer (Nusselt numbers) and effectiveness of the wall cooling respectively. In addition, the results showed increasing efficiency of the cooling through increasing the coefficient of heat transfer (Nusselt numbers) when using ribs that leads to reduce the combustor liner temperature as well as increasing its life.

Nomenclature:

A = Target plate surface area, m^2
A_h = Hole cross-sectional area, m^2
C_D = Discharge coefficient.
D = Jet hole diameter, m
H = Jet to target spacing, m
h_{av} = Averaged heat transfer coefficient, $W/m^2.k$
k = Thermal conductivity of the target plate, W/m.K
$\dot{m}$ = Mass flow rate, kg/s
Nu= Nusselt number
$\overline{Nu}$ = Area-averaged Nusselt number
Q = Heat flow rate, Watt

ToCite ThisArticle: Hussain S. Abd and Arkan AlTaie.,Enhancement of Heat Transfer in a Gas Turbine Combustor Using Jet Impingement on a Flat Plate with Triangular Ribs. ***Aust. J. Basic & Appl. Sci.,*** *10(14): 172-184, 2016*

Re = Reynolds number
S= Jet to jet spacing, m
t= thickness, m
T= Temperature, degree centigrade
Tw= Wall temperature, degree centigrade
X = Local length of the target plate, m
$\bar{\eta}$ = Average wall cooling effectiveness
ΔP = Change in pressure, N/m^2
ρ = Density of the air, kg/m^3

Subscript:
s = crossflow
av.= average
in = inner target surface
j= jet
out = outer target surface
∞ = mainstream flow

INTRODUCTION

The jet impingement configuration is applied to many processes, especially where there is a need for high transfer of the heat generated such as in applications concerning electronic cooling. Prior studies have focused their attention on a variety of parameters *i.e.* variation in the nozzle geometry, Reynolds number, angle of incidence and the geometry configuration. Narayanan *et. al.* (2004) studied the mechanics of an impinging slot jet flow concluding that the mean and RMS – averaged fluctuating surface pressure, and local heat transfer coefficient peaked at the impinging region and decreased monotonically in the wall bounded flow past impingement. The study also tabulates prior important studies and their variation parameters. Shyywoei*etal* (2009) studied the heat transfer characteristics of impinging a jet onto concave and convex dimpled surfaces with effusion. Among other geometries studied, Yan and Mei (2004) examined the angled rib effects by considering both continuous and broken V-shaped configurations with different exit flow orientations.

Katti and Prabhu (2008) in their pursuit to understand and enhance the heat transfer in the detached rib configuration found, contrary to results of the smooth surface, that there is a continuous increase in the heat transfer coefficient from the stagnation point in the stagnation region. Rallabandi et al. (2010) studied the heat transfer characteristics of both jet impingement and channel flow conditions. The range of the Reynolds number for the flow was 5000 to 40000. The study was also made on the heat transfer characteristics of inline and staggered ribs.

In yet another study, Duda et al. (2008) concluded that when a cylindrical pedestal is placed the flow quickly separates over the pedestal edge leading to three distinct regions of the wall jet. There is a recirculation region at the base of the pedestal, a separation zone where flow detaches from the pedestal surface and does not reattach downstream (for low H/d spacing), and a region of separated flow which reattaches after the pedestal boundary and forms the wall jet. Donovana and Murray (2008) in their work studied the effects of rotational motion provided to the Al-foam which was being studied for heat transfer. In view of these studies, geometry with triangular ribs in parallel orientation was manufactured, with holes drilled between the ribs. The aim was to understand the effect of holes on the air flow patterns and heat transfer characteristics in the enclosure.

The aim of present work:

The purpose of present study is to estimate experimentally the heat transfer characteristics of multiple jets system impinged a ribbed flat target. A triangular rib is used to represent the ribbed target surface. Two types of target plate are examined experimentally.

Experimental Set-up and Procedures:

Experimental test rig and set-up:

All experiments were carried out in a low-speed air flowing system is designed and constructed at the University of Technology-Mechanical Engineering Department.

Figure (1) shows the test rig schematic diagram and dimensions and photography. The air of the mainstream is drawn by a (2.5 kW) electrical blower (M) running with 2800 rpm. Main airspeed in the test section (E) is controlled by manually partially open gate and measured velocity by pitot tube that designed according to British standard (F) to maintain (20 m/s) through the test. The mainstreams temperature is flowing through settling chamber (H). In order to allow the air to reach the desired temperature (40C^o), it is initially routed out away from the test section by using a by-bass gate passage until maintaining the desired temperature.

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The secondary flow (the jets crossflow) is regarded as the hot air of the heat transfer process, while the main stream flow is regarded as a cool air to save energy. The jets crossflow is drawn, by (3.0 kW) air pressure blower (A),to the plenum (D). The crossflow flow rate is measuered by using orifice meter (B) located at the crossflow piping system. The crossflow is heated (100°C) by using an electrical heater (C). Both mainstream air and secondary flow are discharged through sigle exit (L) and their temperatures are measured befor get mixed at the test rig exit.

Fig. 1-a: Photography of the experimental test rig

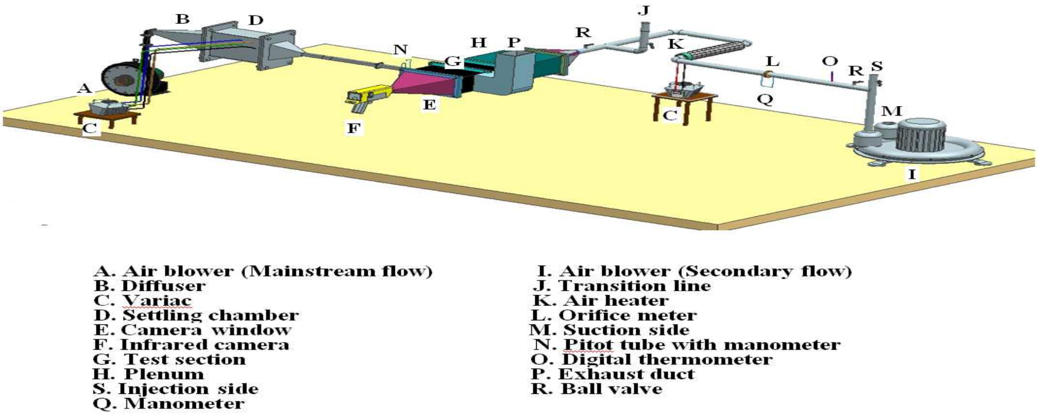

Fig. 1-b: schematic of the experimental test rig

Fig. 1: Experimental test rig

Both air stream temperatures are obtained by digital electronic reader type (TM-903A) with the aid of thermocouples (Type-K)at the test section. The crossflow temperature is taken at one chosen hole, since the pre-testing showed that all jet holes are indicated the same flow rate and temperature conditions. Thermography Infrared camera (Fluke Ti32), (N), is measured the thermal energy emitted from the backside target plate by the resistive air film as a temperature distribution through camera window (K).

Boundary condition:

The mainstream flow physical variables are fixed at (V_∞ = 20 m/s) and (T_∞= 40 ^{0}C), and the secondary flow is at (T_s=100 ^{0}C) with varied mass flow rate according to the required Reynolds number. The flow is assumed to be turbulent.

Test section and test models:

Figure (2) shows the test section 3-D schematic diagram of test rig assembly. Figure (3) shows the schematic diagram of the jet plat geometry and dimensions. Figure (4) shows the ribbed target plats for two models (a and b).

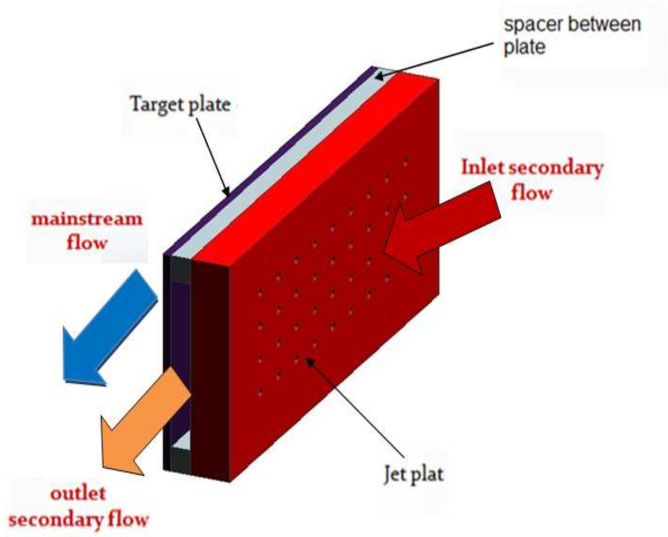

Fig. 2-a: Test section

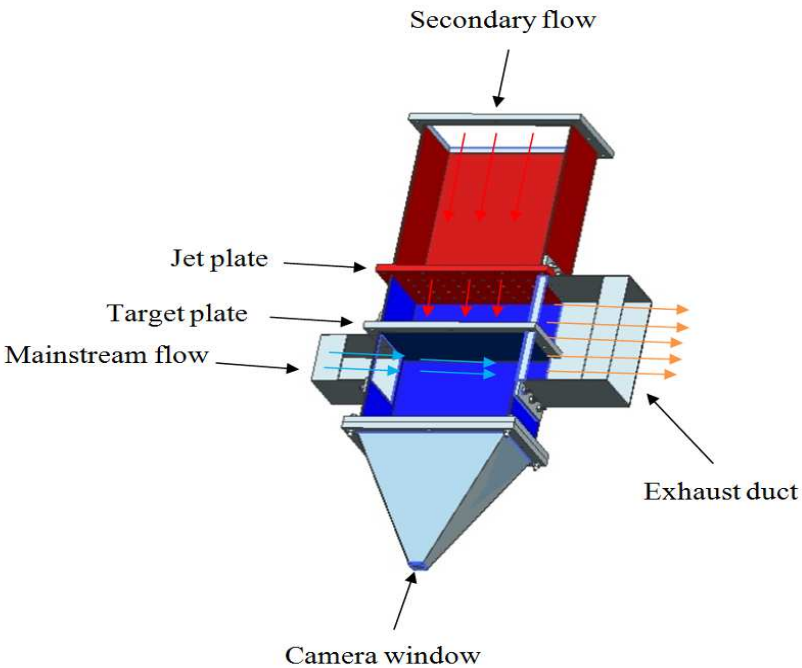

Fig. 2-b: Mechanzism Test section
Fig. 2: Test section

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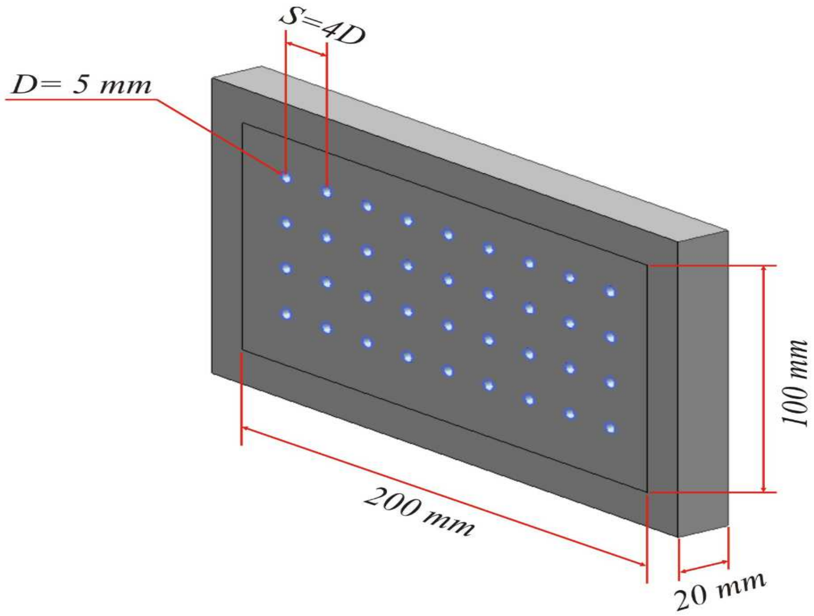

Fig. 3: Jet plate dimension

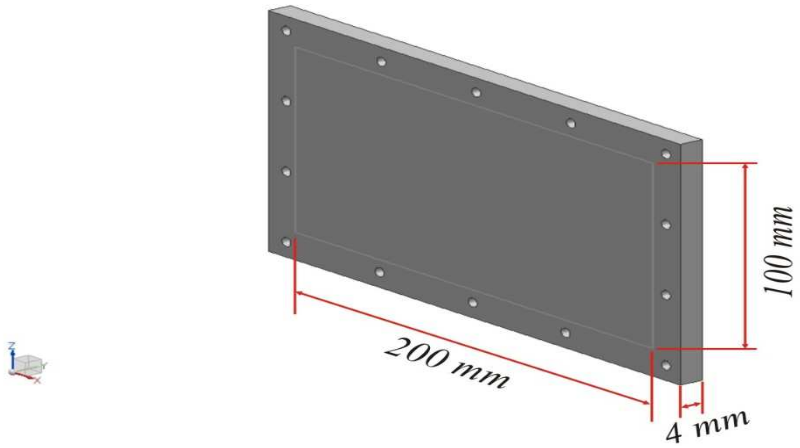

Fig. 4-a: Smooth Target plate

Fig. 4-b:ribbed Target plate(triangular ribbed)
Fig. 4: Target plates models

The calculations:

Experimetal procedure calculation to estimate the averageheat transfer coefficeintcan be done as follows, the total heat lost from the impinging jets flow by mainstream flow can be calculated as follows[9]:

$$Q_L = m^{\cdot} C_p \Delta T \quad = KA \frac{\Delta T_{av.}}{t} = h_{av} A (T_j - (Tw_{in})_{av}) \quad (1)$$

Where,

$$\Delta T_{av} = ((Tw_{in})_{av.} - (Tw_{out})_{av.}) \qquad (2)$$

Then

$$(Tw_{in})_{av.} = \frac{Q_L * t}{KA} + (Tw_{out})_{av.} \qquad (3)$$

where,

$$h_{av} = \frac{Q_L}{A\,(T_j - Tw_{in})} \qquad (4)$$

Therefore, the average Nusselt number ($\overline{Nu}$) obtained is:

$$\overline{Nu} = \frac{\bar{h}D}{K_{air}} \qquad (5)$$

The non-dimensional wall cooling effectiveness is defined as:

$$\bar{\eta} = \frac{T_\infty - Tw_{out}}{T_\infty - T_S} \qquad (6)$$

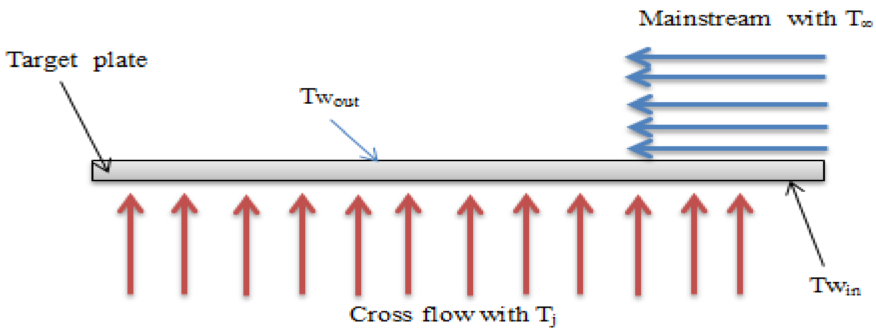

Fig. 5: Target plate with cross flow and mainstream

Discharge Coefficient Evaluation (c_D):

The flow velocity and the geometrical parameters on the discharge coefficient can be calculated according to [10], the pressure losses at the impingement side are defined by the non-dimensional discharge coefficient as:

$$\Delta P = \frac{\dot{m}_h^{\,2}}{2\,A_h^{\,2} CD^2 \rho} \qquad (7)$$

Therefore,

$$C_D = \frac{\dot{m}_h}{A_h\ \sqrt{2\,\rho \Delta P}} \qquad (8)$$

The term (ΔP) represents the difference in the pressure across the impingement wall up to the exhaust which has been measured by a differential manometer.

Experimental Method Verification:

To verify the present experimental steady state test method of impingement case where IR technique was used to measure the wall temperature, the results of average Nusselts number ($\overline{Nu}$) with (Re_j) werecompared with the test methods given by [8], as shown in figure (6).The wall temperatures were measured using IR technique. Both tests were conducted for the same hole's geometry of inline arrangements, number of holes, and ($\frac{H}{D} = 2$and $\frac{H}{D} = 3$).values. Same trend of($\overline{Nu}$) variation with (Re_j) was observed, and approximately both dictated same level of ($\overline{Nu}$) values. It is fair to say that the present experimental method is approved to be a reliable method.

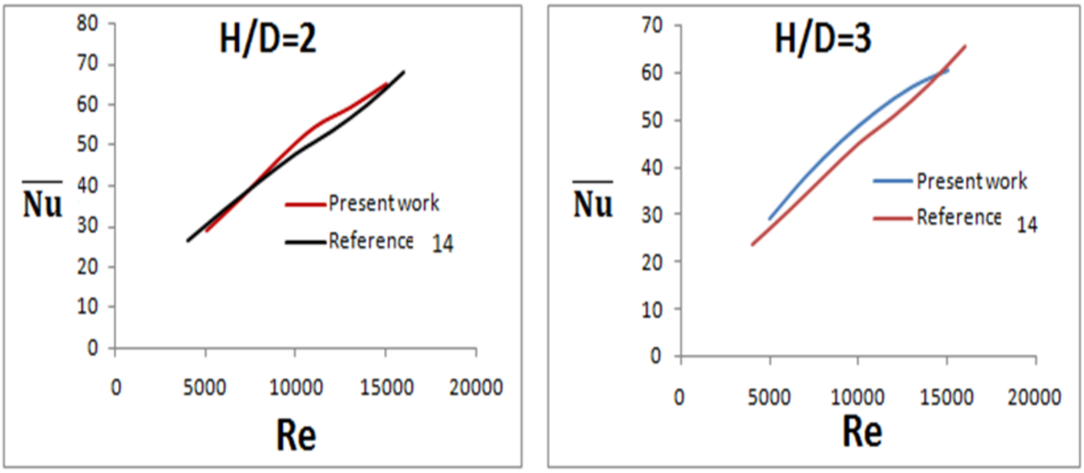

Fig. 6: Experimental verification of ($\overline{Nu}$) verses (H/D) with that of (8)

Experimental work:

RESULTSAND DISCUSSION

The flow characteristic of impinging jets for inline arrangements at($\frac{H}{D}=2$ and $\frac{H}{D}=3$). In the present study, the parametric variation of the heat transfer coefficient (Nusselt number) shows an increasing trend with the increase in the Reynolds number for all case. Similarly the Nusselt number can generally be seen to be increasing with the corresponding increase in the Reynolds number, Figure (7)shows the Nusselt number varaition in spanwise direction model (1) smooth flat plate baseline case at($\frac{H}{D}=2$and$\frac{H}{D}=3$),and model (2) triangular ribs at$\left(\frac{H}{D}=2 \text{ and } \frac{H}{D}=3\right)$.The Nusselt number is increased with (X/D) up to (X/D= 32) for all (Re)at$\left(\frac{H}{D}=2\right)$andup to (X/D= 28) ($\frac{H}{D}=3$)further more Nusselt number tend to decrease beyond(X/D= 28) at($\frac{H}{D}=3$). The region laying between (X/D=0) to (X/D=28) shows best averageNusselt numberat ($\frac{H}{D}=3$), in which the impingement jets are deflected away towards the downstream direction due to the effect of cross flow induced by upstream jets. The deflection becomes significant as the flow progresses downstream of the first raw as seen in Figure (7). The momentum of the impingement jets is reduced due to the interaction with cross flow, and this affects the rate of heat transfer at stagnation region. This cross flow shows a positive enhancement on heat transfer at the downstream region where high momentum and flow velocity are creating due to jet flow accumulation, and this will enhance the wall effectiveness in the downstream direction.

Figures (8) present the variation of average Nusselts number ($\overline{Nu}$),Average wall cooling effectiveness($\overline{\eta}$)and discharge coefficient (CD)with Reynolds number respectively, and shows theNusselts number ($\overline{Nu}$) increase whenReynolds number(Re) increase for all cases.Figure (9) is represented the temperature distribution of target plate backside for twomodels baseline and lateral triangular ribs at $\frac{H}{D}=2$ and at$\frac{H}{D}=3$with Re no.between(5000-13000).

Figure (10) shows the experimental jet impact tapping on the clean target and ribbed target plate.

The maximum increments in coefficient of the heat transfer (Nusselt number) and effectiveness of the wall cooling are 20.41%,17.64% at$\frac{H}{D}=2$ and at$\frac{H}{D}=3$ are 18.11%,12.95%respectively.

In this work the effect of the ribs shape, geometry and arrangement are examined and effect jet Reynolds number(R_{ej}) on the average heat transfer coefficient and average wall cooling effectiveness are evaluated for clean and ribbed target surfaces.

Pressure Losses and Discharge Coefficient:

The pressure loss is expressed in term of discharge coefficient (C_D), since the pressure drop is a combination of flow contraction in the impingement jet plate and the shear force induced due to friction take place within the cooling passage. Figures (8) show the influence of (H/D) on the discharge coefficient (C_D) for inline array. The result shows a significant influence of H/D on the discharge coefficient or pressure drop for inline array. For inline array, (C_D) is increased as H/D increased indicating a lower pressure drop as (H/D) increased for the same Reynolds number or jets velocity. In low jet distance (H/D), the cross flow passage is narrowed, therefore the flow shear effect is increased leading to low (C_D) values.

Heat Transfer Correlations:

The data of cooling performance are presented in terms of Nusselts and Reynolds number. For all (H/D) values, ($\overline{Nu}$) values are increased with increasing of (Re_j)and maximum heat transfer was obtained at (H/D = 2), for both models. The average Nusselts number of the present tests results can be correlated using the conventional non-dimensional approach that considering the above parameter as follows:

$(\overline{\mathbf{Nu}} = \mathbf{CRe_j}^{\mathbf{n}}(\mathbf{H/D})^{\mathbf{m}}\mathbf{pr}^{\mathbf{v}})$

where, C, n, m and v are constants determined by experiments.

The experimental results were gathered, and the least square mathematical technique was implemented to obtain the following correlation for both smooth target plate and triangular ribs target plate for inline array.

Smooth target plate:

$$\overline{Nu} = 0.0833 Re_j^{0.711} \left({H}/{D}\right)^{-0.06875} Pr^{0.33} \ldots \ldots \quad (9)$$

Triangular ribs target plate:

$$\overline{Nu} = 0.073 Re_j^{0.761} \left({H}/{D}\right)^{-0.224} Pr^{0.33} \ldots \quad (10)$$

The maximum deviation between the experimental($\overline{Nu}$) and correlated($\overline{Nu}$) for smooth flat plate andtriangular ribbed target plat is 4.83%,4.61 respectively, from range of (Re_j =5000 to 15000).

Conclusions:

The heat transfer characteristics and pressure loss have been investigated and the following conclusions can be derived from the present work for impingement cooling system.

1- For clean target plat the heat transfer coefficient (Nusselt number) is increased in the downstream direction (X/D) in the region laying between (X/D=0) to (X/D=32) for all (R_{ej}) at $\frac{H}{D} = 2$.

2- the avarage heat transfer coefficient (Nusselt number) and average wall cooling effectiveness are highly depended upon the jet Reynolds number and ribs arrangement.

3- For laterial triangular ribbed target plate the heat transfer coefficient (Nusselt number) is increased graduilly with (X/D) and the maximum value of (Nusselt number) is occured at (X/D=32) at$\frac{H}{D} = 2$.

4- The average heat transfer coefficient (Nusselt number) and average wall cooling effectiveness are the best for laterial ribs case for all jet Reynolds number and (H/D).

5- The maximum increments in coefficient of the heat transfer (Nusselt number) and effectiveness of the wall cooling are 20.41%,17.64%at$\frac{H}{D} = 2$ and 18.11%,12.95% at$\frac{H}{D} = 3$ respectively.

6- From the structure of jet impingement flow field the high level of turbulence is generated with pair of vortex in the space between the triangular ribs and jet spacing.

7- Both jet spacing and Reynolds number have an evident effect on the discharge coefficient. For both cases, low (C_D) values are obtained at jet spacing $\frac{\boldsymbol{H}}{\boldsymbol{D}} = 2$ and high (C_D) at jet spacing$\frac{\boldsymbol{H}}{\boldsymbol{D}} = 3$.

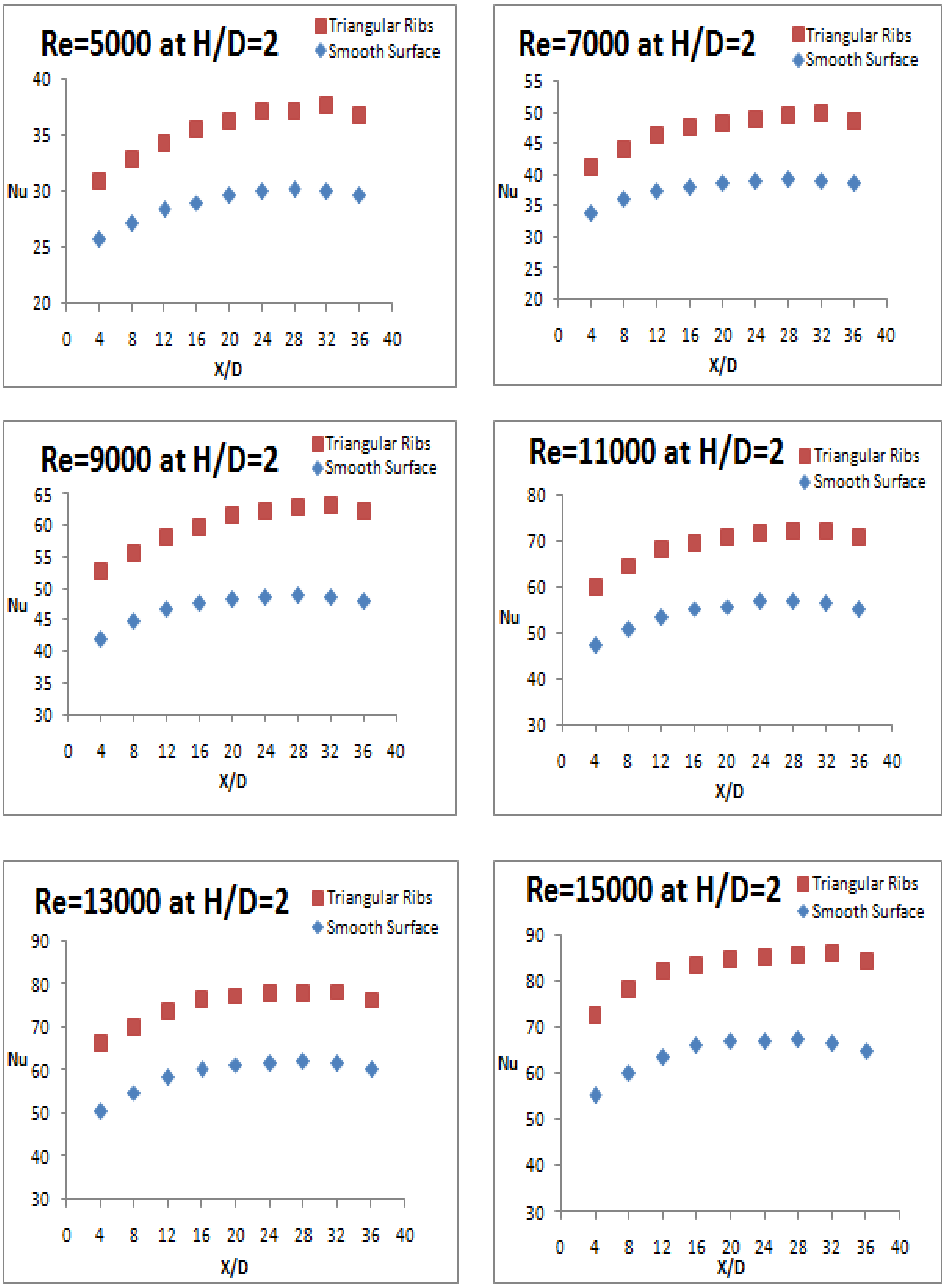

Fig. 7-a: The (Nu) variation along the downstream direction with different (Re) : models (1, 2) at $\frac{H}{D} = 2$

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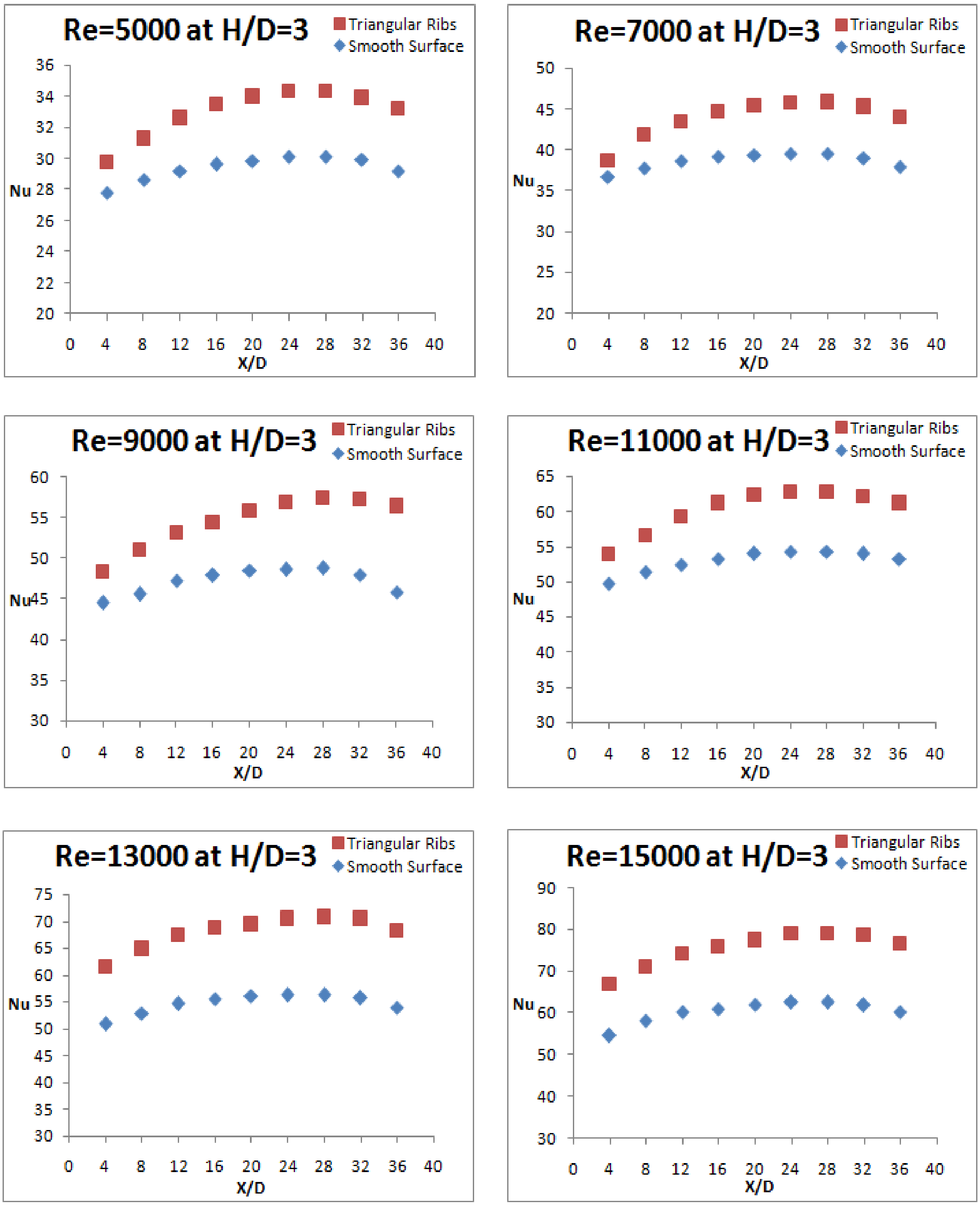

Fig. 7-b: The (Nu) variation along the downstream direction with different (Re) : models (1, 2) at $\frac{H}{D} = 3$

Fig. 7: The (Nu) variation along the downstream direction with different (Re): models (1, 2)

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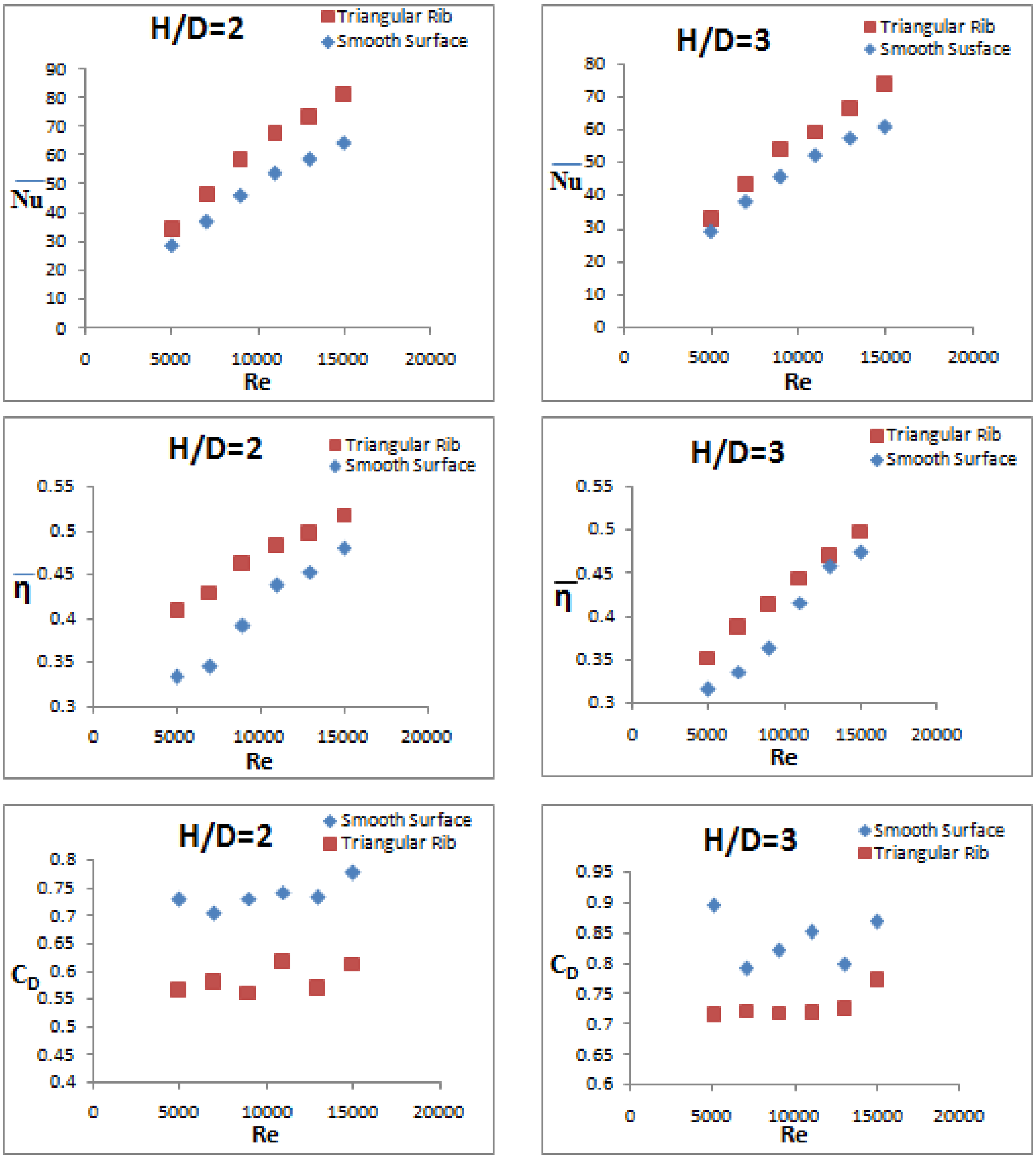

Fig. 8:present the variation of average Nusselts number($\overline{Nu}$) , average wall cooling effectiveness ($\overline{\eta}$) and discharge coefficient (C_D) with Reynolds number(Re)

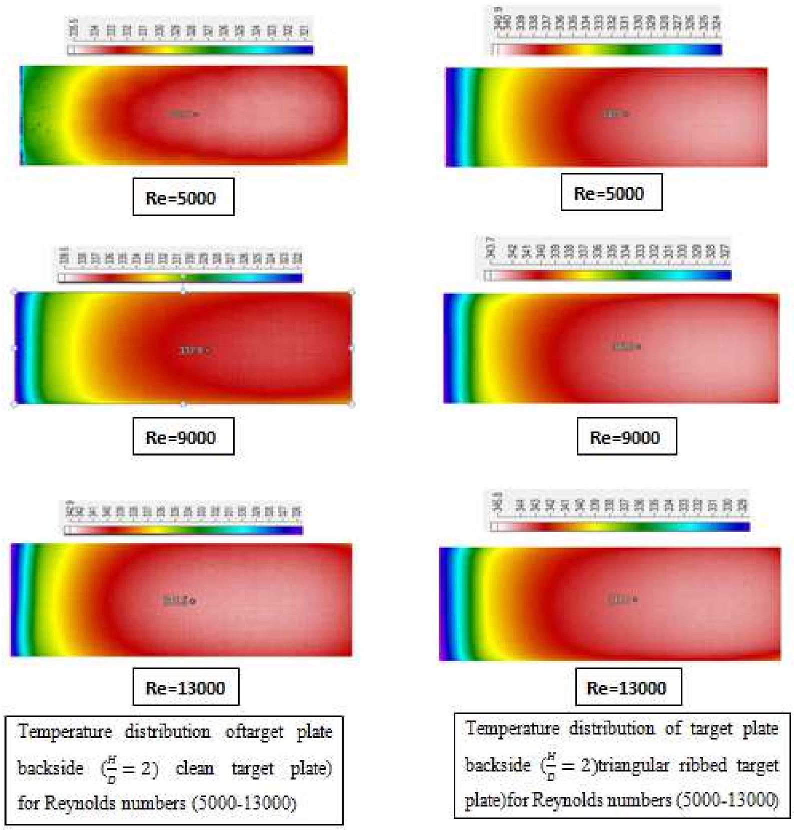

Fig. 9-a: Temperature distribution of target plate backside at ($\frac{H}{D}=2$)

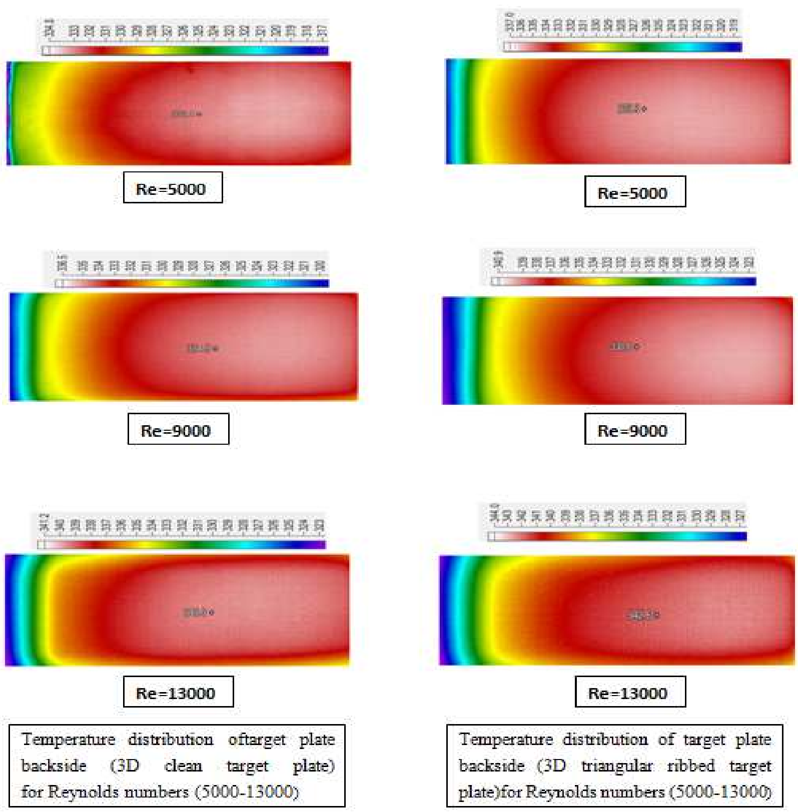

Fig. 9-b: Temperature distribution of target plate backside at ($\frac{H}{D}=3$)

Fig. 9: Temperature distribution of target plate backside

Fig. 10-a: Effect of hot jet impact on the clean target plate

Fig. 10-b: Effect of hot jet impact on the triangular ribbed target plate
Fig. 10: Effect of hot jet impact on two models of target plate

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