Experimental Investigation of Jet Impingement Cooling On Ribbed Target Surface

Dr. Assim. H. Yousif
Mechanical Engineering Department, University of Technology/ Baghdad
Email: assem_yousif20000@yahoo.com
Ali K. Bayer
Mechanical Engineering Department, University of Technology/ Baghdad
Email:eng.alikareem1990@yahoo.com

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ABSTRACT

Experimental investigation of jets impacting at right angle on a flat surface is introduced in the present study. Round jet holes of inline arrays arrangements with diameter (5mm) and jet to jet spacing of 4 jet hole diameter are considered. Jet Reynolds numbers of 4000 to 16000 and jet height to diameter ratio (space between the jet plate and target plate) of 2.0 is maintained. The heat transfer model consists of a multiple jet holes, ribbed target plate having back side resistive film. Three target models have been considered; model (1) is a clean surface target acting as the baseline case, the model (2) the target surface consists of lateral rib rows and model (3) the target surface consists grids rib arrangement. The ribs height for models (2 and 3) is 0.86 of jet diameter of the triangle shape. The average wall cooling effectiveness and heat transfer coefficients and consequently the Nusselt numbers for each model are estimated. Numerical computations of jets flow field are made. The results show that model (3) enhanced both heat transfer coefficients in the target plat inner surface and wall cooling effectiveness in the target plat outer surface greatly, the maximum increments in heat transfer coefficient and wall cooling effectiveness are 14.5% and 4.5% respectively.

Keywords: jet impingement cooling; ribbed target; grid ribs; lateral ribs.

اختبار تجريبي لتبريد النفاثات الصدمية على سطح هدف مضلع

الخلاصة

تم اجراء تجارب عملية للتبريد الصدمي النفثي بزاوية قائمة على سطح صفيحة الهدف المستوية . وقد كانت النفاثات عبارة عن ثقوب دائرية مرتبة على شكل صفوف تقع على خط واحد ذات قطر(٥ملم) وكان البعد بين ثقب واخر هو ٤ مرات قطر الثقب. وتراوحت قيم عدد رينولدز للثقوب من ٤٠٠٠ الى ١٦٠٠٠ والمسافة بين صفيحة الهدف والثقوب هي مرتين من قطر الثقب. وكان النموذج العملي لعملية انتقال الحرارة هو نتيجة اصطدام عدد من النفاثات على سطح صفيحة مضلعة سطحها الخارجي ذو غشاء مقاوم. وتم اعتماد ثلاثة نماذج لاجراء التجارب: النموذج الاول هو الحالة الاساسية وهي صفيحة الهدف الخالية من الاضلاع والنموذج الثاني هو لصفيحة الهدف ذات الاضلاع العرضية والنموذج الثالث هو لصفيحة الهدف ذات الاضلاع الشبكية. كانت الاضلاع المستخدمة ذات شكل مثلث متساوي الاضلاع وكانت نسبة ارتفاعه الى قطر الثقب هي ٠.٨٦. تم حساب فعالية التبريد ومعامل انتقال الحرارة (عدد نسلت). واظهرت

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النتائج افضلية للنموذج الثالث في تحسين معامل انتقال الحرارة وكذلك فعالية التبريد وكانت اعلى نسبة للزيادة هي 14.5 % و٤.٥ % على التوالي.

Nomenclature
A = Target plate surface area, m^2
A = Hole cross-sectional area, m^2
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
= Mass flow rate, kg/s
$\overline{Nu}$ = Area-averaged Nusselt number
Q = Heat flow rate, Watt
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
¯ = Average wall cooling effectiveness
= 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

Jet impingement cooling is an enhanced heat transfer method capable of cooling a combustor liner without injecting cool air directly into the combustion chamber. Cooling the liner from the backside enables engineers to dissipate the heat load and maintain more uniform temperatures in the combustion region needed for efficient combustion [1]. An impingement array is comprised of a jet plate typically having holes which produces the impinging jets.

The jets strike the surface to be cooled, the surface referred to as the inside surface of the target plate. Traditionally, according to [1], the structure of an impinging jet is broken down into three parts, the potential core, shear layer, and the wall jet. Figure (1) shows the impinging jet structure and the associated region.

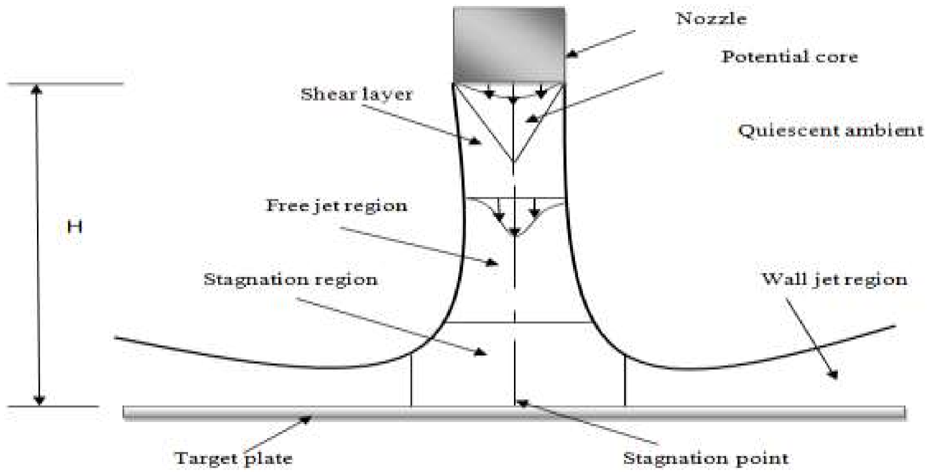


Figure (1) Impinging Jet Structure [1]

At the discharge of the jet plate, the velocity profile of the jet is relatively uniform. The potential core of the jet is defined as the region where the viscous forces have little or no effect on the velocity profile. Once the jet strikes the target plate, the wall jet is formed as the fluid travels along the wall. Due to the viscous forces the peak velocity and causing the wall jet to thicken as it moves away from the stagnation point.

The presence of a cross flow tends to disturb the impinging jet pattern, thicken wall boundary layers, and degrade transfer rates. The turbine blade cooling channel is included the modeling cross flow due to spent flow passing thorough a confined channel.

The heat transfer coefficient is highly dependent upon jet Reynolds number, jet spacing to diameter ratio, jet plate to target plate distance to diameter ratio and jet inclination [1].

Impinging jets are used with other target modifications such as ribbed walls. The inclusion of ribs on the target surface disturbs the wall jet and increases turbulence. The ribs may also function as fins to increase the effective surface area for transfer of energy. While the ribs increase the transfer rates outside the stagnation region, the height end drag of the ribs causes the wall jet to decelerate and disperse more rapidly [2].

Existing Studies

Colucci and Viskanta [3], Van Treuren et al. [4], Son et al. [5], Gao [6] and Donovan [7], concluded that the local heat transfer coefficients for confined jets are more sensitive to Reynolds number and nozzle-to-plate spacing. Dano et al. [8] studied the effect of nozzle geometry on the heat transfer performances of a semiconfined impinging jets array. The results showed that the averaged Nusselt numbers decrease with increasing jet-to-plate spacing when$1<H/d<4$. Uysal et al. [9] made experiment on an in-line array varying the jet hole-size in a systematic manner. They show the influence of the flow rate varied by the jet hole-size on the cross flow. Park et al. [10] investigated the influence of Mach number and Reynolds number on the impingement heat transfer for $H/d=3$ with Reynolds number ranging from 15,000 to 60,000. Azad et al [11], Used line jets array (four row with each row having twelve jet holes) impinging orthogonally on pinned target surface and observed the heat transfer is increased greatly. Nakod et al. [12], Studied the effect of circular air jet impinged finned (rough) flat surface on the local heat transfer coefficient for different Reynolds number. The local heat transfer coefficient for finned surface increased to up to 77%. Maurrer [13] investigated the effect of fixing two successive ribs on smooth channel wall impinged by jets. The heat transfer levels for ribbed channel were of around three times higher than the heat transfer of a smooth channel wall for wide range of Reynolds number. Kartik et al. [14] investigated the results of experimental investigations of the flow and heat transfer from a triangular ribbed heated plate with holes, impinged by a round jet in a confined case built around the plate. The parameters varied were heat flux. These values were further processed to calculate the Reynolds number, heat transfer coefficients and consequently the Nusselt numbers for each combination. Lei Tan [15] experimentally investigated the convective heat transfer on the rib-roughened wall impinged by a row of air jets inside semi-confined channel. Four rows of transverse ribs were arranged in the wall-jet zone downstream from the impinging jet stagnation to enhance heat transfer rate. The ribs on the impinging target do provide stronger convective heat transfer in the wall-jet region. Katii and Prabhu [16] used axisymmetric detached rib-roughness to study the heat transfer performance of flat surface impinged by normal air jet of circular holes and found increase in Reynolds number increases the heat transfer at all the radial locations for a given space.

The aim of present work

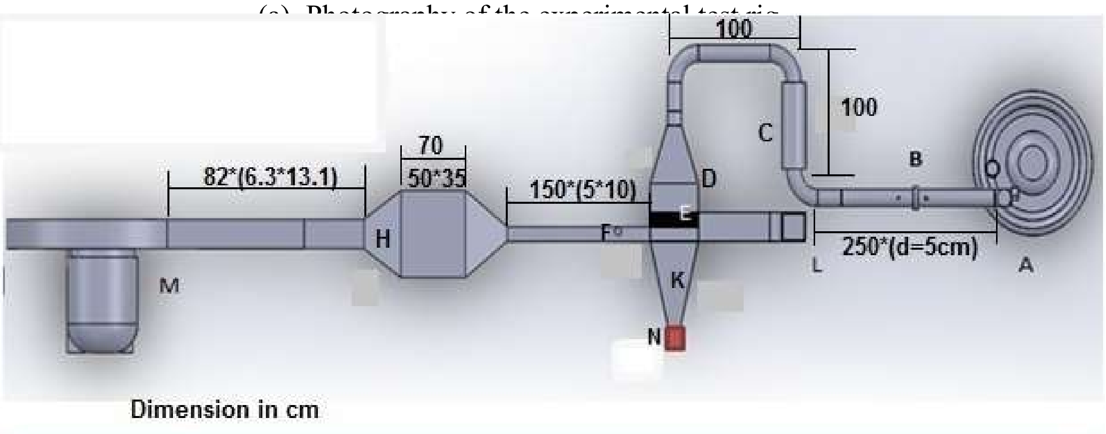
The purpose of present study is to estimate experimentally the heat transfer characteristics of multiple jets system impinged a ribbed flat target. A triangle ribs is used to represent the ribbed target surface. Two types of ribbed surface lateral and grid are examined experimentally.

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 (2) 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.

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 (100C) 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.



(c) schematic of the experimental test rig

Figure (2) 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 cross flow 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 (3) shows the test section 3-D schematic diagram of test rig assembly. Figure (4) shows the schematic diagram of the jet plat geometry and dimensions. Figure (5) shows the ribbed target plats for three models (1, 2 and 3).

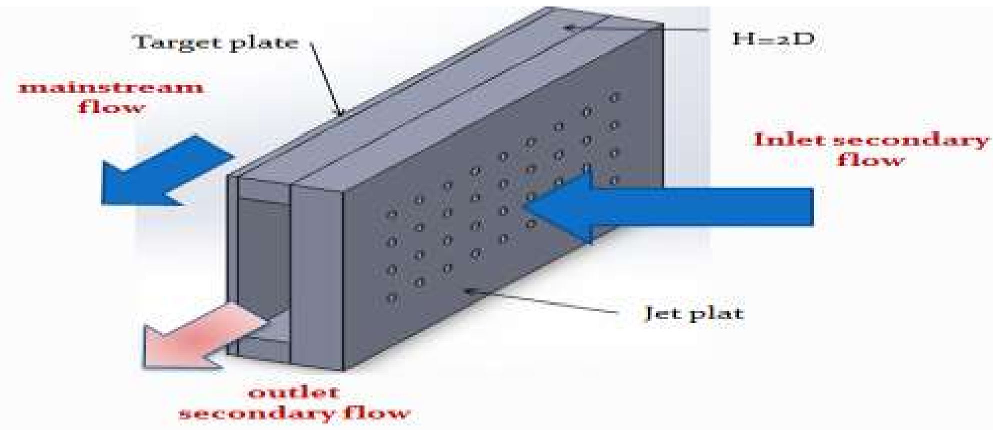
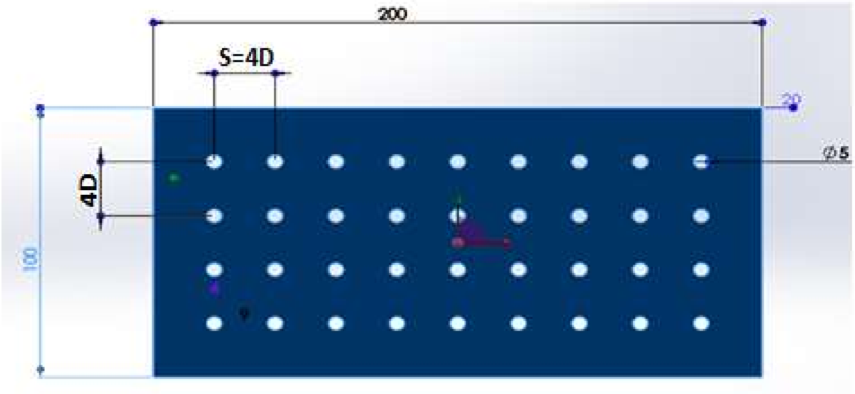
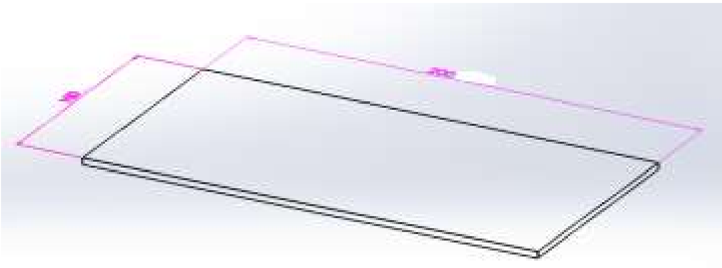


Figure (3) Test section

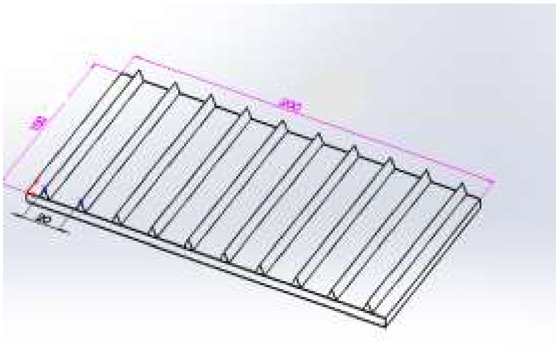


Jet no. 1 2 3 4 5 6 7 8 9

Figure (4) Jet plate dimension



(a) Target plate model 1



(b)Target plate model 2 (c) Target plate model 3

Figure (5) Target plates model

The calculation

Experimetal procedure calculation to estimate the average heat transfer coefficeint can be done as follows, the total heat lost from the impinging jets flow by mainstream flow can be calculated as follows:

Where,

Then

Where,

The non-dimensional wall cooling effectiveness is defined as:

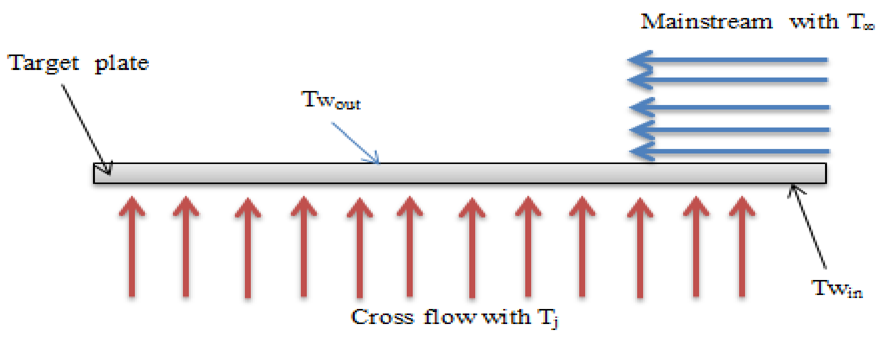


Figure (6) Target plate with cross flow and mainstream

Prediction of jet structure

Computational Fluid dynamics (CFD) output data are representing the flow field structure of the impinging jet. To demonstrate the effect of the turbulence model that involves the solution *(κ – ε)* model, present two equations solved in terms of two turbulence parameters by ANSYS Fluent version (14.5). Three modules are used to solve the flow equations, the preprocessor module is a program that creates the geometry (Modeling of geometry) in three dimensional to represent jets structure by using solidworks2013, Mesh generation and can generate meshing by workbench in the ANSYS Fluent package (14.5) and Boundary condition, and both the mainstream and cross flow are assumed to be incompressible ideal gases.

Experimental Method Verification

To verify the present experimental method and technique of impingement system, the multiple-impinging jets system is indicated that the average Nusselt number at the target plate is in good agreement with experiments results of [14]. Figure (7) shows the results of average Nusselt number ($\overline{\mathrm{Nu}}$) variation with (H/D) for both present and that of [14] results. Measuring data for both results are conducted for the same hole geometry, inline arrangements, number of holes, and jet Reynolds number. Same trend of $(\overline{\mathrm{Nu}})$ variation with (H/D) and approximately close values of $(\overline{\mathrm{Nu}})$ are observed, as seen in Figure (7). It is fair to say that the present experimental method and technique both are found to be a reliable.

Result and Discussion

The flow characteristic of impinging jets for inline arrangements at $(\mathrm{H/D} = 2)$. The result shows the velocity vector within the cooling passage colored by temperature distribution.
In the present study, the parametric variation of the heat transfer coefficient shows an increasing trend with the increase in the Reynolds number for base case. Similarly the Nusselt number can generally be seen to be increasing with the corresponding increase in the Reynolds number; Figure (8) shows the average heat transfer coefficient varaition in spanwise direction model (1), baseline case. The average heat transfer coefficient is increased with (X/D) up to (X/D= 35) for all (Res), further more (h_{av}) tend to decrease beyond (X/D= 35). The region laying between (X/D=0) to (X/D=35) shows best average heat transfer coefficient, 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 (8). 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 (9 and 10) present the variation of average heat transfer coffiecient and average wall cooling effectiveness with Reynolds number respectively, while Figure (11) is represented the temperature distribution of target plate backside for three models (1, 2 and 3). Lateral and grid ribs show relatively higher h_{av} and η_{av} than that of clean target surface (baseline model). Figure (12) represents the jet velocity vectors in jet spacing (between jet and target plates) and show clearly that the jet (No.1) (No. mean sequence of jets for same row of inline array) is stroke the target plate at right angle for both clean and ribbed surfaces. For there more downstream across the jet spacing up to the last jet (No. 9) the jets are deflected toward the exit gradually starting from jet (No.3) up to jet (No.9). The photograph in Figure (13) shows the experimental jet impact tapping on the clean target, it is clearly that this photograph verify the predicted velocity vectors presented in Figure (12), especially the jet deflection behavior.
Figure (15) shows the average heat transfer coefficient variation in spanwise direction for models (1, 2 and 3) for lateral ribs is greater than that of clean plate and for grid ribs is greater than that of lateral ribs. The variation of the increments of the avarage heat transfer coffecient (Δh_{av}) for models (2 and 3) over that of model (1) with R_{ej} are presented in Figure (16). The maximum $(\Delta h_{av})_{max}$ for model (2) is of about $(\Delta h_{av})_{model\ (2)} = 26$ at (R_{ej}=14000) and for model (3) is of about $(\Delta h_{av})_{model\ (3)} = 37.9$ at (R_{ej}=10000), the maximum level of enhancement of model (3) over that of model (1) are (14.5%) in average heat transfer coefficient and (4.5%) in average effectiveness cooling. Figure (14) shows clearly that the predicted side view of the velocity

vectors for grid ribs plat is dominated by high level of turbulence (pair of vortex) that occupied the grid ribs area and the jets spacing.The reasons for (h_{av}) increment are due to the generation of highly turbulent wake and vortices as a results of presented ribbed surfaces. These turbulent wake and vortices are prevented any boundary layer formation near the target plate, shearing of the cross flow over the surfaces of the ribs and then creates turbulence and that can be shown clearly in Figure (12), in which the predicted velocity vectors for three models at the difference jets location show clearly that turbulent and the vorticity.

CONCLUSIONS

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. The ribs shape, geometry and arrangement are examined.

1- For clean target plat the heat transfer coefficient is increased in the downstream direction (X/D) in the region laying between (X/D=0) to (X/D=35) for all (R_{ej}).

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

3- For lateral and grid ribbed target the heat transfer coefficient is increased gradully with (X/D) and the maximum value of (h_{av}) is occured at (X/D=35).

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

5- The maximum $(\Delta h_{av})_{max}$ for model (2) is of about $(\Delta h_{av})_{model\ (2)}$ = 26 at (R_{ej}=14000) and for model (3) is of about $(\Delta h_{av})_{model\ (3)}$ = 37.9 at (R_{ej}=10000), therefore the maximum level of enhancement for model (3) over that of model (1) are (14.5%) in average heat transfer coefficient and (4.5%) in average effectiveness cooling.

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 grid ribs and jet spacing.

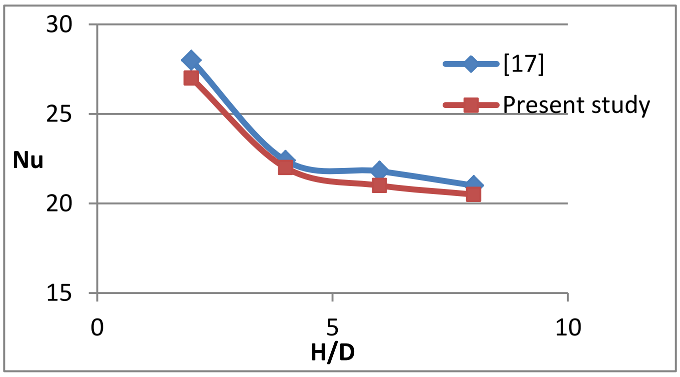


Figure (7) Experimental verification of (Nu) verses (H/D) with that of [17]

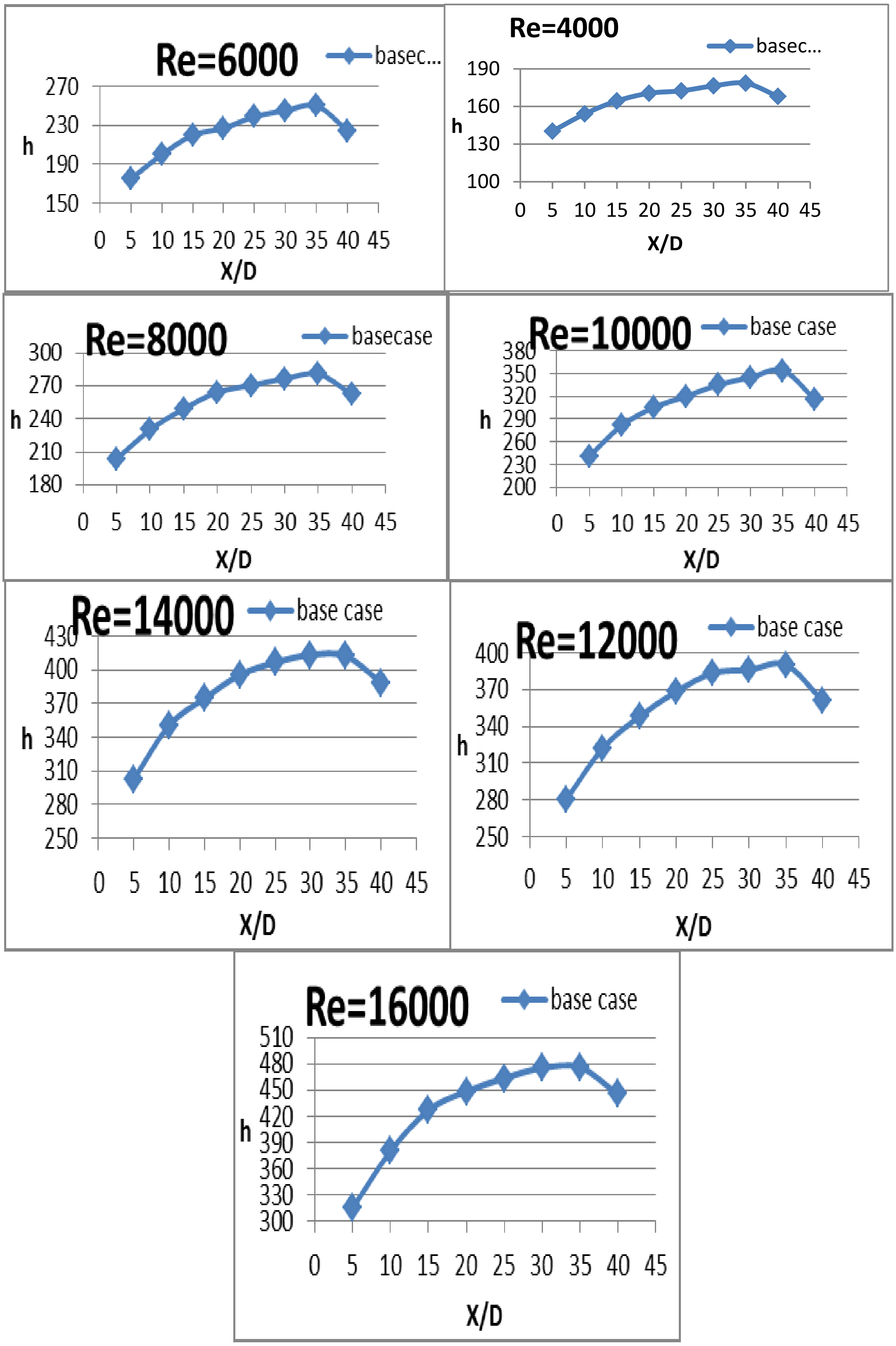

Re=6000
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Figure (8) Heat transfer coefficient variation in the downstream direction, model (1)

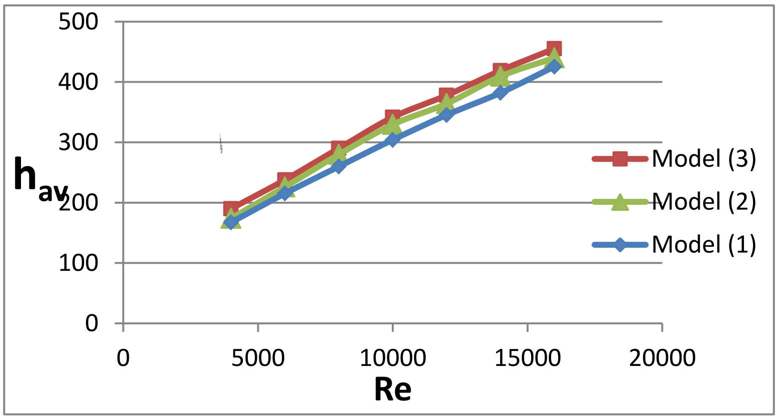
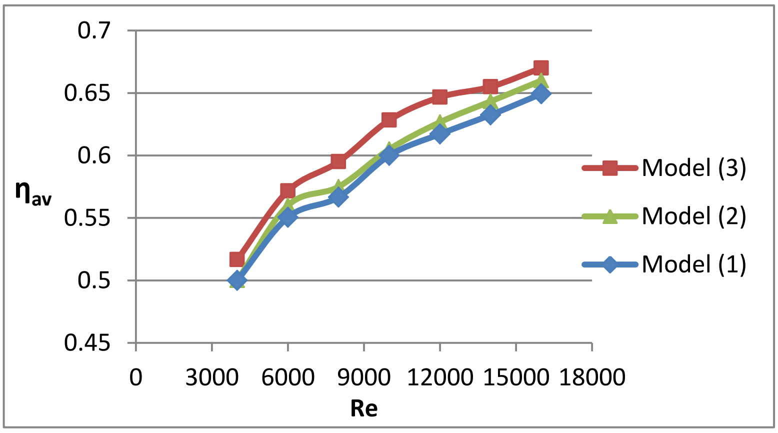
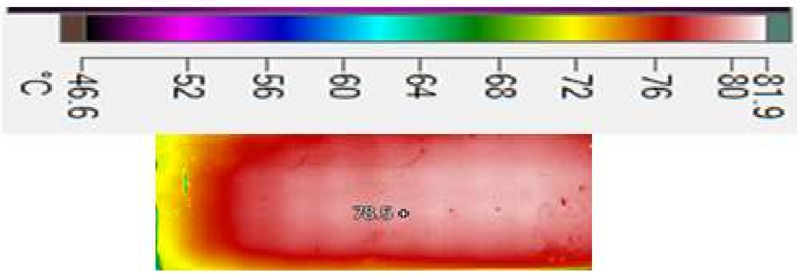


Figure (9) The (h_{av}) variation with Re for models (1, 2 and 3)



Figure(10) (η_{av}) variation with (Re)



Model 1

Model 2

Model 3

Figure(11) Temperature distribution of target plate backside

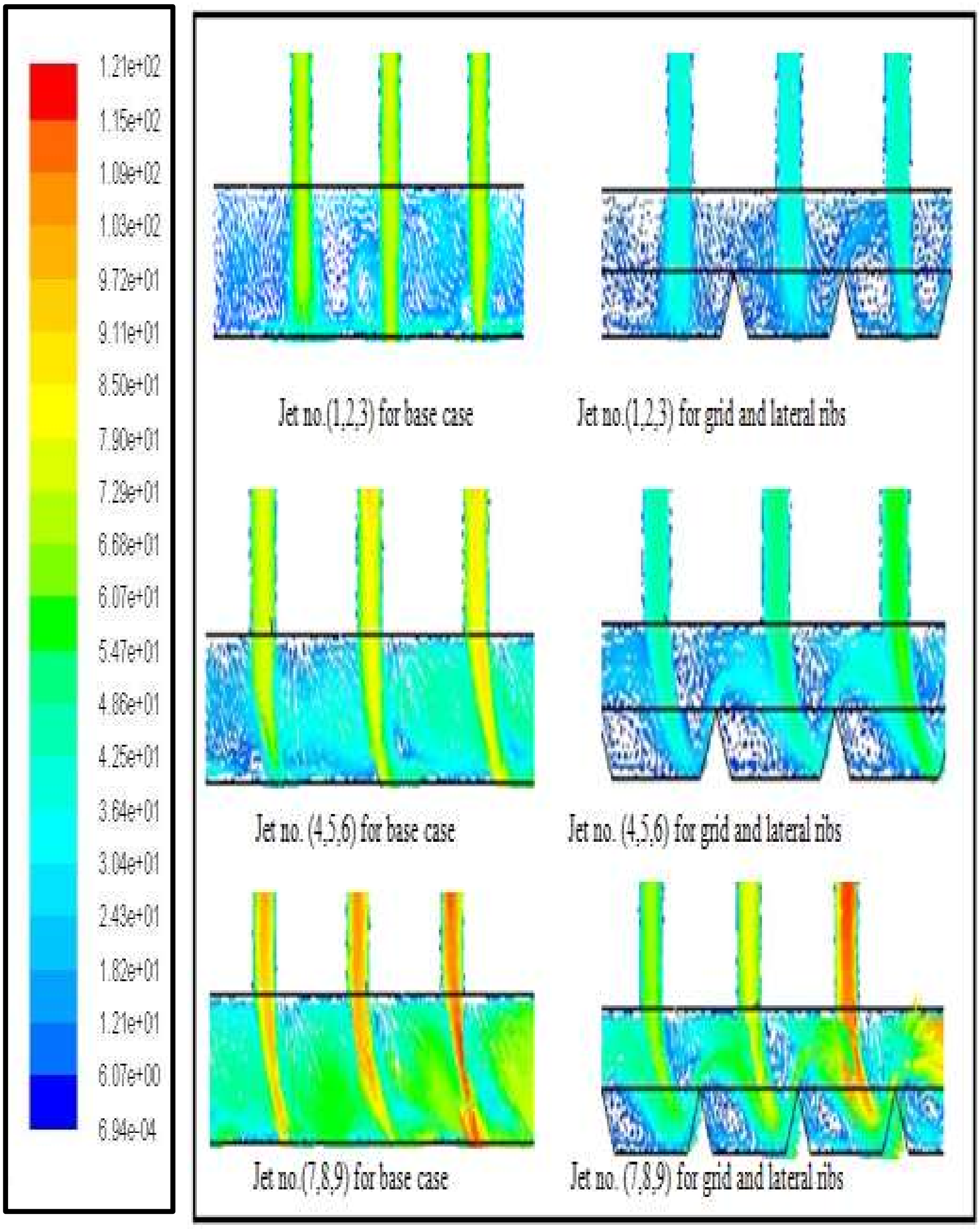


Figure (12) Velocity vector by velocity magnitude (m/s) for three models

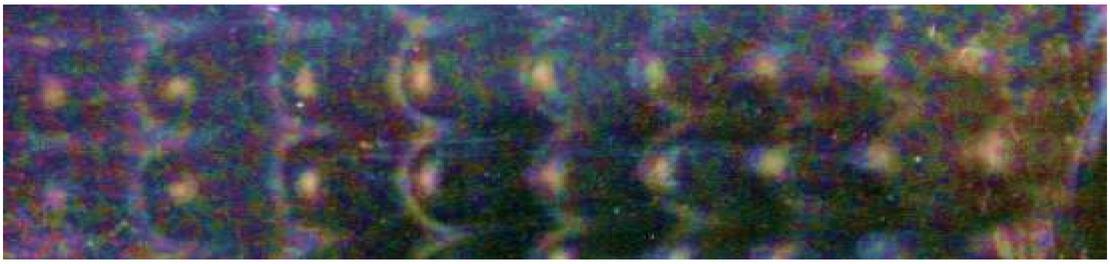
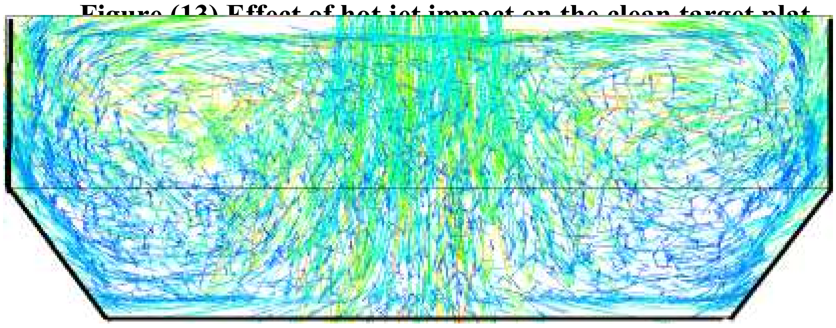
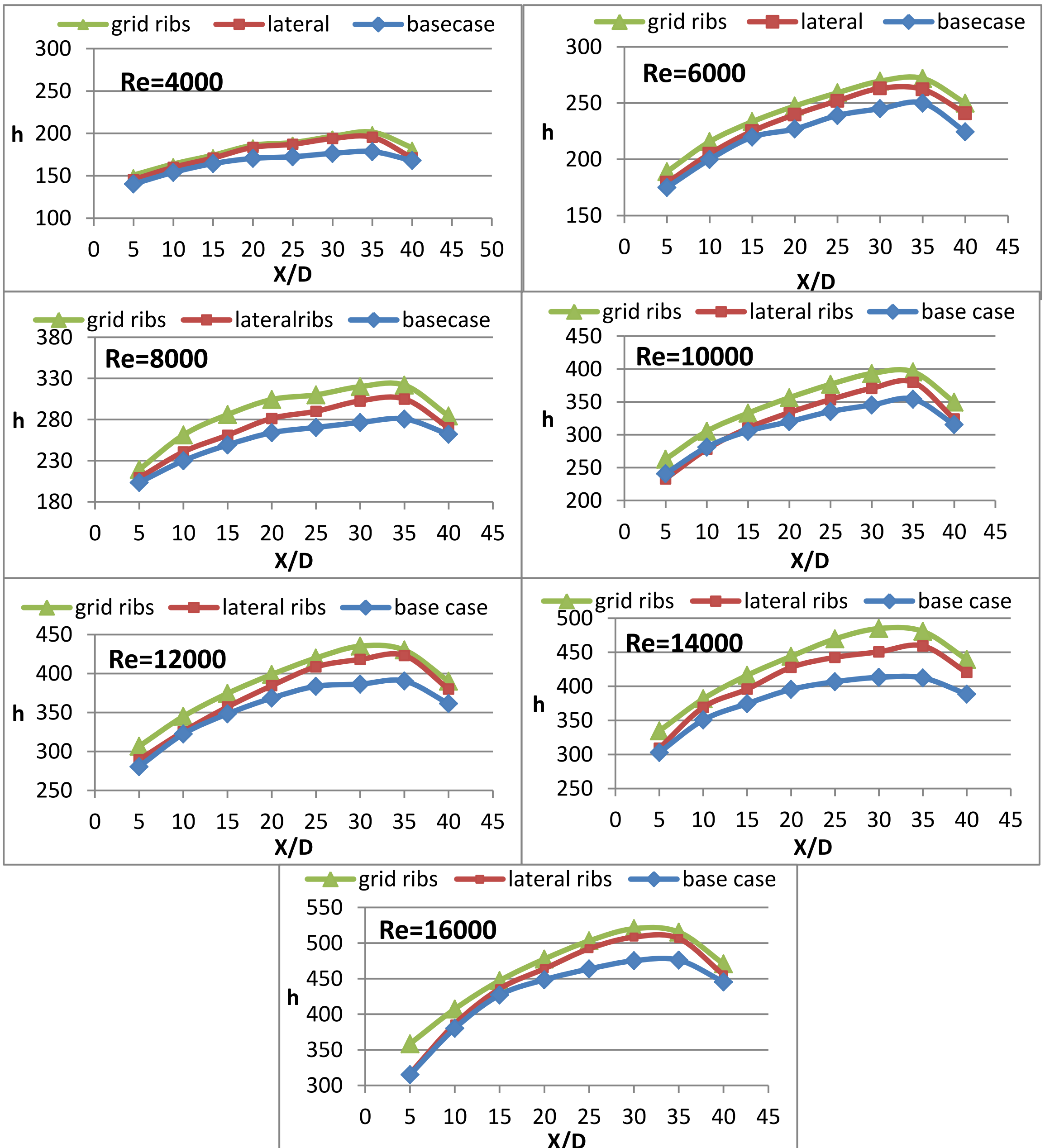


Figure (13) Effect of hot jet impact on the clean target plat



Figure(14) Velocity vector side view, grid ribs



Figure(15) The (h) variation along the downstream direction with different (Re) : models (1, 2 and 3)

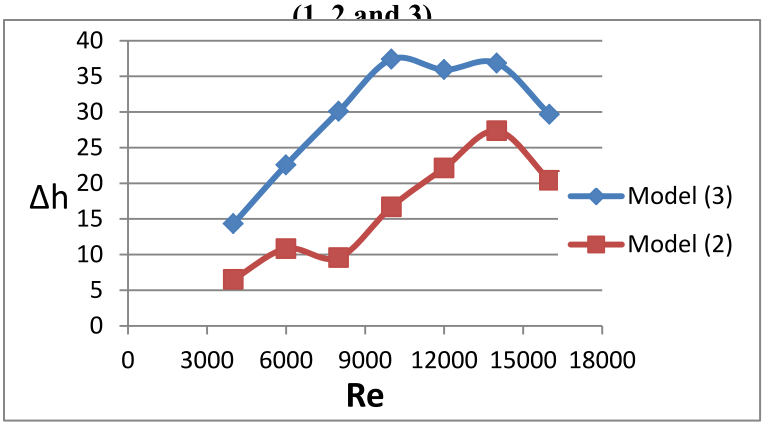


Figure (16) Increment between models (1 with 2 and 3) of (h_{av}) variation with Reynold number.

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