Heat transfer correlations of perpendicularly impinging jets on a hemispherical-dimpled surface

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\begin{abstract}
Heat transfer results of an inline array of round jets impinging on a staggered array of hemispherical dimples are reported with the consideration of various parametric effects such as Reynolds number (\textit{Re}_j), jet-to-plate spacing (\textit{H}/\textit{D}_j), dimple depth (\textit{d}/\textit{D}_d) and ratio of jet diameter to dimple projected diameter (\textit{D}_j/\textit{D}_d) for both impinging on dimples and impinging on flat portions. The results were normalized against those from a flat plate. The heat transfer was measured by using transient wideband liquid crystal method. Our previous work (Kanokjaruvijit and Martinez-Botas (2005) [1]) on the effect of crossflow scheme suggested that jet impingement coupled with channel-like flow formed by the crossflow helped enhance heat transfer on a dimpled surface; hence three sidewalls were installed to constrain the spent air to leave in one direction. Throughout the study, the pitch of the nozzle holes was kept constant at 4 jet diameters. The Reynolds number (\textit{Re}_j) ranging from 5000 to 11,500, jet-to-plate spacing (\textit{H}/\textit{D}_j) varying from 1 to 12 jet diameters, dimple depths (\textit{d}/\textit{D}_d) of 0.15, 0.25 and 0.29, and dimple curvature (\textit{D}_j/\textit{D}_d) of 0.25, 0.50 and 1.15 were examined. The shallow dimples (\textit{d}/\textit{D}_d = 0.15) improved heat transfer significantly by 70\% at \textit{H}/\textit{D}_j = 2 compared to that of the flat surface, while this value was 30\% for the deep ones (\textit{d}/\textit{D}_d = 0.25). The improvement also occurred to the moderate and high \textit{D}_j/\textit{D}_d. Thereafter, the heat transfer results were correlated in dimensionless form by using logarithmic multiple regression. The correlations were reported with necessary statistics.

\end{abstract}

1. Introduction

An effective cooling scheme in gas turbine design is necessary for high efficiency and performance. There have been a number of attempts to enhance the heat transfer of the critical components located in the hot sections, where temperatures may reach 2000 °C, i.e. in the inner wall of the leading edge region of blades and the outer wall of combustors. Jet impingement is one of the most effective methods of heat transfer augmentation by forming small-scale vortex pairs, which cause local heat transfer enhancement [2]. Once air leaves a nozzle, it entrains and exchanges momentum with the surrounding stagnant air causing the width of the mixing region to expand continuously. Beyond the end of the potential core, the centerline velocity attenuates because the jet exchanges large amount of momentum with the entrained fluid; however, the turbulence intensity from this point on increases significantly. According to Gordon and Akfirat [3], at the distance 1.2\textit{D}_j above the target plate, the free jet behavior still takes place. Upon impinging, mixing-induced turbulence is continuously generated in the wall jet. The level of the turbulence is dependent of whether the oncoming jet is fully developed. Nevertheless, the turbulence induced by mixing affects the local heat transfer in less degree than the arrival jet velocity, which affects the boundary layer thickness at the stagnation region. The effects of Reynolds number and jet-to-plate spacing are investigated by many previous researchers that determine the turbulence intensity of the jets for fully developed turbulent jet, which Huber and Viskanta [4] affirmed that occurred at \textit{Re} \textgreater 2000.

In order to generate more coherent vortices and detach and restart The boundary layer more often, hence higher heat transfer, there have been numerous developments in surface protrusions, such as rib-rougheners or turbulators [5], pin-banks or pin-fins [6]. Nonetheless, these implementations cause excessive pressure loss as well as a different flow field than discretely mounted ribs, resulting in cooling/heating non-uniformity and thermal stresses [7]. In addition, the high cost of maintenance and manufacturing, and large weight penalties, are important concerns for these types of heat transfer enhancers.

An alternative method of vortex generation is the use of concavity, which is defined as an indentation on the surface forming a recess rather than the protrusion. A dimple is one kind of concavity,
and has a drag reduction characteristic in external flow over bodies – such as is demonstrated by golf balls [8]. In addition, dimples require less material, and induce less pressure loss due to more organized flow motion, while they help enhance heat transfer. Kesarev and Kozlov [9] applied a single dimple to parallel flow, and found a 50% heat transfer augmentation compared to the result of a flat plate. Studies of a channel flow past an array of staggered dimples [5–7,10,11] illustrated that heat transfer enhancement approximately doubled and friction increased by only 1.6—2 times compared to that of a flat plate.

On account of the above findings, the combination of jet impingement and dimples were thought to have potential in enhancing heat transfer more than impingement on a conventional flat surface. In this investigation, experimental heat transfer results in terms of overall average Nusselt numbers were correlated to parametric effects such as Reynolds number (Re), jet-to-plate spacing (H/D), dimple depth (d/Dd) and ratio of jet diameter to dimple projected diameter (Dj/Dd) for dimple impingement on dimples and on flat portions adjacent to the dimples. The statistical method of logarithmic multiple regression was used, and the statistics are reported in ANOVA table.

2. Experimental procedure

2.1. Experimental apparatus

Fig. 1 shows the experimental set-up consisting of a 500 × 500 mm acrylic plenum chamber, which was equipped with a honeycomb sandwiched by two mesh screens to straighten the flow, and a 320 mm × 320 mm hole at the bottom of the plenum. An acrylic plate with an array of 4 × 4 nozzle holes of 20 mm diameter (Dj) and pitch 4Dj was then attached to the hole. Additionally, a drawer was introduced in order to divert the flow and allow it to settle before discharging through the working section. A target plate was hung by four studs, which allowed the required jet-to-plate spacing to be adjusted. Similarly, three sidewalls, see Fig. 2, were also installed by using studs to form a crossflow, which left the plate in a channel flow-like manner. The impinging positions were set into two configurations as shown in Fig. 3: jets impinging on dimples (nozzles inline with dimples) and jets impinging on flat portions adjacent to dimples (nozzles inline with flat portions). Results from both cases were compared to those from the flat plate, which was a baseline case. Fig. 4 shows the dimpled plate.

![Fig. 1. Experimental apparatus.](image-url)
Transient heat transfer experiments were carried out by heating the air that was generated by a fan with a 9-kW heater; the air then flowed through the plenum chamber. The flow was diverted by the drawer in order to allow the conditions such as temperature to be set inside the plenum chamber. The thickness of the target plate was designed under the assumption of one-dimensional unsteady heat conduction and from the relation, \( z = 2\sqrt{\frac{t}{\kappa}} \) [12]. A coating of liquid crystals of temperature range 35–45 °C was applied to the top surface of the target plate. A coat of black paint was sprayed on top of the liquid crystals for the purpose of reducing reflection from the target plate surface. A 3 CCD camera was set underneath the plate to view the liquid crystal color transformation, as shown in Fig. 1. Since the hue is insensitive to the local light intensity and the illumination angle [13], after all the set-up was fixed the calibration was then carried out. The liquid crystal calibration of the hue against temperature was carried out with a calculated uncertainty within ±1.5 °C. As per Yuen and Martinez-Botas [14,15], the curve fitting and regression analysis resulted in the fifth-order polynomial temperature–hue relationship. Each camera image was transferred to a personal computer via a firewire lead, which transferred data directly from the camera to the computer without the use of a frame grabber. Each experiment was filmed in non-compression mode in order to receive the data as complete as possible.

The transient method was applied throughout the heat transfer experiments. Hence, the well-known correlation of one-dimensional unsteady heat conduction was used \( \theta = 1 - \exp(-\beta t \varepsilon) \), where \( \theta = (T_i - T_f)(T_m - T_i) \) and \( \beta = h\sqrt{\frac{1}{\rho \kappa}} \). The bulk temperature, \( T_m \), was measured inside the plenum chamber. The experimental uncertainty throughout this study was within ±12.2% based on 95% confidence levels according to Moffat [16]. The highest uncertainty that might occur was the sum of uncertainties of: the temperature measured by the liquid crystals, the initial and the bulk flow temperatures measured by thermocouples, the time frame measurement and the thermal product \( \sqrt{\rho \kappa} \) for the acrylic substrate. Thermocouples used in this study had an uncertainty of ±0.5 °C, which made the uncertainties of initial and bulk flow temperatures become ±2.5 and ±1%, respectively. The highest uncertainty of time was ±0.67%. Finally, the uncertainty of the thermal properties of the acrylic substrate was typically ±5% [17].

The heat transfer results are reported in terms of Nusselt numbers, which is defined as \( \text{Nu} = \frac{hD_d}{\kappa} \). Local Nusselt numbers are presented in contour plots. Spanwise and streamwise average Nusselt numbers are assigned to the average values in spanwise and streamwise directions of the crossflow, respectively. The overall average Nusselt number, \( \overline{\text{Nu}} \), is defined as an area average value noting that the dimple area taken into account the calculation in this investigation is treated as a circle.

### 2.3. Heat transfer model

In order to correlate the heat transfer results, statistics was employed as a tool. The average heat transfer result was reported in dimensionless form called average Nusselt number, \( \overline{\text{Nu}} \), and it is called “response variable”. Other variables such as \( \text{Re}_{D_j}, H/D_j, d/D_j \) and \( D_j/D_d \) are called “explanatory variables”. Since there was more than one explanatory variable involved in each case, multiple regression was obtained from ANOVA (analysis of variance) table [18].

### Table 1

<table>
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<td>0.25</td>
</tr>
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<td>0.29</td>
<td></td>
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</table>

**Fig. 2.** Crossflow formed by three sidewalls.

**Fig. 3.** Impinging positions.
2.3.1. Flat plate

In general, the correlations shown in previous studies [4,19,20] always included at least two explanatory variables, which were $Re$ and $H/D_j$. Another explanatory variable that was often used in the correlation was relative nozzle open area, which is defined as $\alpha_j = \frac{x_j}{(D_j)^2}$. However, in this study, the relative nozzle open area was kept constant at 0.049. Furthermore, the Prandtl number had to be taken into account, but since it is constant for air jets, it was combined in the constant of the correlation for convenience. Many researchers have employed the multiplicative power form

$$\text{Nu}_{\text{flat}} = C_{\text{flat}}(Re)^{\alpha_{\text{flat}}} (H/D_j)^{b_{\text{flat}}}$$

The regression coefficients $C_{\text{flat}}, \alpha_{\text{flat}}$, and $b_{\text{flat}}$ were estimated by using $F$-test to determine the confidence of the model.
Transforming Eq. (1) to base-10 logarithmic (log10) form before gives

\[
\log_{10}\left(\frac{\text{Nu}}{\text{Nu}_0}\right) = \log_{10}(C_0) + a_0\log_{10}(\text{Re}) + b_0\log_{10}(\text{H/D}_j) + c_1\log_{10}(\text{d/D}_d) + d_1\log_{10}(\text{D}_j/D_d)
\]

(2)

2.3.2. Dimpled plates

Similar to the case of the flat plate, for both cases of dimple impingement, two more explanatory variables: \(d/D_d\) and \(D_j/D_d\) are included as follows:

\[
\text{Nu}_{d1} = C_1\text{Re}_D^2 (\text{H/D}_j)^{0.5}(\text{d/D}_d)^{0.5}(\text{D}_j/D_d)^{0.5}
\]

(3)

\[
\text{Nu}_{d2} = C_2\text{Re}_D^2 (\text{H/D}_j)^{1.0}(\text{d/D}_d)^{0.5}(\text{D}_j/D_d)^{0.5}
\]

(4)

where subscript 1 represents the case of jets impinging on dimples, and 2 the case of jets impinging on flat portions. In logarithmic form, Eqs. (3) and (4) are written as follows:

\[
\log_{10}\left(\frac{\text{Nu}}{\text{Nu}_0}\right) = \log_{10}(C_0) + a_1\log_{10}(\text{Re}) + b_1\log_{10}(\text{H/D}_j) + c_1\log_{10}(\text{d/D}_d) + d_1\log_{10}(\text{D}_j/D_d)
\]

(5)

\[
\log_{10}\left(\frac{\text{Nu}}{\text{Nu}_0}\right) = \log_{10}(C_0) + a_2\log_{10}(\text{Re}) + b_2\log_{10}(\text{H/D}_j) + c_2\log_{10}(\text{d/D}_d) + d_2\log_{10}(\text{D}_j/D_d)
\]

(6)

Note that Reynolds number with subscript \(D_j\) is to ensure that the Reynolds number is calculated based on a jet diameter.

3. Results

3.1. Heat transfer of flat plate – baseline

Throughout this study, the heat transfer results of the flat plate were used as baseline. Therefore, it was worth validating the experimental results against the published ones in the literature. Fig. 5 represents the plots of overall average Nusselt numbers against jet-to-plate spacings for each Reynolds number value, and shows good agreement.

3.2. Parametric effects on heat transfer

Four parametric effects are in consideration for dimple impingement \([1,18,21,22]\): Reynolds number (\(\text{Re}_{D_j}\)), jet-to-plate spacing (\(\text{H/D}_j\)), dimple depth (\(\text{d/D}_d\)) and ratio of jet diameter to dimple projected diameter (\(\text{D}_j/D_d\)), whereas only two parametric effects of \(\text{Re}_{D_j}\) and \(\text{H/D}_j\) are involved for flat plate impingement. In addition, impinging positions also influences the heat transfer of the dimple impingement. Thus, this effect will be mentioned afterward, and each case of impinging position was investigated and correlated separately.
3.2.1. Effect of Reynolds number

In terms of average Nusselt numbers in Fig. 6, the heat transfer at $Re_D = 11,500$ was 51% higher than that of the flat plate while $Re_D$ of 8000 and 5000 led to 38% and 22% improvement, respectively. After jets impinge on a target plate, they form crossflow, commonly known as spent flow, which deflects subsequent jets from their intended target positions, thus a decline in the overall average heat transfer across the target plate occurs. With the presence of dimples, the dimple edges helped detach the spent air in a higher degree than caused by post-impingement on a flat plate.

![Graph](image)

**Fig. 10.** Streamwise average Nusselt numbers of dimples at $H/D_j = 2$, $Re_D = 11,500$ and $D_j/D_d = 0.50$. 

This helped relieve the heat transfer degradation that took place due to the aforementioned jet deflection. In addition, according to the flow visualization reported by Cornero et al. [21], rolling-up vortices were formed downstream of the issuing turbulent jets. When impacting on the concave plate, the radial vortices were formed and oscillated at the stagnation point. This effect accelerated and broke down the boundary layer removing the symmetry of the jet. When the jet velocity at the jet exits was increased such as at a higher value of \( \text{Re}_D \), the velocity of the spent air also increased, and the vortices were more vigorous, leading to an overall heat transfer augmentation.

### 3.2.2. Effect of jet-to-plate spacing

For the flat plate impingement, the narrow \( H/D_j \) led to higher heat transfer as might be expected. However, the dimple impingement reacted to the effect of \( H/D_j \) differently. Fig. 7 shows that at \( H/D_j = 2 \) for \( d/D_{D_4} = 0.29 \) the presence of dimples did not improve the heat transfer compared to the flat plate while at larger \( H/D_j \) values dimples helped enhance the heat transfer. This was thought to be caused by a strong recirculation that disturbed the oncoming jets for the narrow \( H/D_j \). At larger \( H/D_j \) (i.e. \( H/D_j = 12 \)), the jets lost momentum to the ambient air whilst traveling towards the target plate.

From the flow visualization of an impinging jet on a concave surface done by Cornero et al. [21] along the discharging jet, axial rolling-up vortices were formed, which were found at approximately 4.5\( D_j \), and then became radial vortices entrained away toward the exit of the target plate. This effect was thought to increase the turbulence intensity. This could explain the heat transfer improvement for moderate jet-to-plate spacings such as \( H/D_j \approx 8 \).

### 3.2.3. Effect of dimple depth

The movies of the liquid crystal color transformation showed that different dimple depths caused different flow propagation from the dimples. Notwithstanding this, the trend of heat transfer was similar. The magnitudes of Nusselt numbers of the shallow dimples (\( d/D_{D_4} = 0.15 \)) are higher than those of the deep ones (\( d/D_{D_4} = 0.25 \)). The overall average quantities in Fig. 8 show that both \( d/D_{D_4} \) values led to heat transfer improvement compared to the flat surface. The shallow dimples resulted in a 48% increase while the deep ones showed a 23.4% augmentation. The deep dimples caused a higher degree of recirculation of the radial vortices inside the indentation, thus the heat transmission was poorer than that of the shallow ones, from which the vortices were shed downstream more easily and faster. Additionally, despite the fact that the distance from the nozzle exit to the stagnation point for a shallow dimple was narrower than that for the deep one, this led to slightly higher momentum of the impinging jet.

### 3.2.4. Effect of the ratio of jet diameter to dimple diameter

Fig. 9 shows the streamwise average Nusselt numbers for \( D_j/D_{D_4} \) values of 0.25, 0.50 and 1.15 at \( \text{Re}_{D_4} = 11,500 \) and \( H/D_j = 2 \). The small \( D_j/D_{D_4} \) of 0.25 led to the lowest heat transfer compared to the results of the flat plate while the moderate and high \( D_j/D_{D_4} \) led to heat transfer improvement. The movie of liquid crystal color transformation suggests that the heat transfer augmentation occurs only inside the inline dimples. After impinging the jets lose their momentum to a large extent; thereafter, the crossflow is weak. Low heat transfer occurs everywhere else including the flat portions adjacent to the inline dimples. For moderate and high \( D_j/D_{D_4} \) values, an improvement in heat transfer was found. When the jets traversed towards the dimples, stagnation zones downstream were wider than the dimples themselves. Plus the spent air formed by the upstream jets pushed the oncoming jets to impact further downstream ruining the jet symmetry. Therefore, the high velocity region covered the entire dimples and their vicinity, and this allowed the dimple edges to detach and restart the boundary layer on the wall of the plate better. Additionally, radial vortices formed after impinging were also deflected forward assisting to transmit heat from the flat portion. Notwithstanding this, the high heat transfer regions for \( D_j/D_{D_4} = 1.25 \) were found more than \( D_j/D_{D_4} = 0.50 \).

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![Image](https://via.placeholder.com/150)

**Fig. 11.** Contour plot of Nusselt numbers of dimpled plate of 40 mm imprinted diameter and \( d/D_{D_4} = 0.15 \) at \( \text{Re}_{D_4} = 11,500 \), \( H/D_j = 2 \), impinging on dimples.
3.2.5. Effect of dimple positions

The comparisons of the streamwise average Nusselt numbers of impinging jets on dimples and on flat portions at Re_{Dj} = 11,500 and H/D_j = 2 are shown in Fig. 10. For the deep dimples (d/D_d = 0.25), jets impinging on dimples led to higher heat transfer results than that on flat portions (no improvement). The video images of the thermochromic liquid crystal color changes showed trace of strong crossflow deflecting the oncoming jets to impinge slightly further downstream instead of normally on the dimples located inline to the nozzles, i.e. x/D_d = 1, 2, −1, −2 indicate the inline dimples and x/D_d = 0 represents the center, which is a flat portion. This deflection was thought to function more properly together with turbulence promoters such as dimples. For impinging jets on flat portions, the oncoming jets were shifted to impinge onto the upstream edges of dimples, which were located downstream of the corresponding flat portions. Additionally, cold regions were found downstream of the dimples located inline to the nozzles.

Fig. 12. Contour plot of Nusselt numbers of dimpled plate of 40 mm imprinted diameter at Re_{Dj} = 11,500, H/D_j = 2, D_j/D_d = 0.50 impinging on flat portions. Note that the crossflow is from right to left.
inside the staggered dimples for jets impinging on flat portions. The jets lost momentum at the upstream edges; hence, the heat transfer was reduced.

Nevertheless, for the shallow dimples \( (d/D_a = 0.15) \), the results of jets impinging on flat portions were slightly higher than those of jets impinging on dimples. For the case of impinging on dimples, due to jet deflection by the crossflow, jets impinged on downstream halves of the dimples, thus high heat transfer regions covered dimple edges and their small vicinity of adjacent flat portions. Fig. 11 shows the \( Nu \) contour plot for the case of \( d/D_a = 0.15 \) at \( Re = 11,500, H(D_j) = 2 \), impinging on dimples, and it suggested that the crossflow helped shed vortices from the dimples marked as \( (1) \) and \( (2) \), which functioned as vortex sources, toward the staggered dimple marked as \( (3) \). Nevertheless, the flat portions situated downstream showed poor heat transfer. The case of impinging on flat portions exhibited in Fig. 12 gave high heat transfer regions in the intended flat portions, adjacent downstream dimples (i.e. marked as \( (2) \)). Once the radial vortices produced after impinging on flat portions moved into the dimple, it functioned as a source. Additionally, the contour denoted that vortices shed toward the dimple situated in the spanwise direction of the flat portion inline to the jet. This helped promote heat transfer across the target plate without cold regions.

3.3. Effect of lateral heat transfer relative to dimple

A commercial software package called COMSOL was used to perform finite element method over a segment of the dimpled plate of the size 100 mm wide and 25.4 mm thick in 2D axisymmetric coordinates \( (r, z) \) as shown in Fig. 13. The case of impinging on a dimple was considered. The curve boundary was underneath the oncoming jet; hence, the temperature was set at 50 °C equal to that of the jet. Since the flat portion adjacent to the dimple was in contact with spent air, its temperature was assigned to 45 °C. The part of the substrate inline to the exit was exposed to the ambient air was, thus this boundary was set 20 °C. Then, the computational domain was discretised into uneven triangular grids by the Delaunay algorithm. Due to the fact that the thermal boundary condition in the experiment was transient isothermal, time-dependent solver was used with time of 10 s. The post-processed heat conduction has to be taken into account in the area of steep curvature. For experimental approach, new technique of 2D conduction thermography technique will have to be developed.

4. Heat transfer correlations

The least squares fit regression and ANOVA tables for correlations of jets impinging on a flat plate, impinging on dimples, and impinging on flat portions were performed in Tables 1–3. The standard error \( (SE) \) is an estimated standard deviation \( (SD) \) of sampling distribution of the least square estimate, and this is related to the \( t \)-statistic as

\[
t-statistic = \frac{\hat{\beta}_j - \beta_j}{SE(\hat{\beta}_j)}
\]

where \( \hat{\beta}_j \) represents the estimate of a coefficient, and \( \beta_j \) is the value obtained from the hypothesis \( (\beta_j = 0) \). The \( t \)-statistic then bestows a two-sided \( p \)-value, which gives the reader more information for judging the significance of the hypothesis. The smaller the \( p \)-value, the greater is the evidence that the hypothesis is incorrect, meaning how far the coefficient differs from zero. For all cases, \( p \)-values were small for all coefficients, and that suggested that it was convincing that the coefficients differed from zero.

4.1. Heat transfer correlation for flat plate

Fig. 15 illustrates the comparison of the average Nusselt numbers at various \( Re_0 \) at \( H(D_j) = 4 \) of this study to the literature. For \( Re_0 = 5000 \) and 8000, the data agrees well with the correlations of Huber and Viskanta [4] and Kercher and Tabakoff [23]. However, at \( Re_0 = 11,500 \), the data point is slightly higher than the value of \( Nu_{ave} \) of [24]. This was thought to be due to the fact that the set-ups were different – such as the relative nozzle open area mentioned above.

![Fig. 13. Finite element tested segment.](image)

![Fig. 14. Distribution of heat flux underneath the surface of acrylic substrate.](image)
4.2. Heat transfer correlation for two cases of dimple impingement

The ANOVA table in Table 1 gave the correlation

\[ \log \left( \frac{N_u_{\text{flat}}}{\text{Re}_{\text{dip}}} \right) = -(0.8117 + 0.6891 \log(\text{Re}_{\text{dip}})) - 0.4906 \log(H/D_j), \quad R^2 = 0.9731 \]  

(7)

All two-sided \( p \)-values for all coefficients were small suggesting that it was convincing that the coefficients differed from zero. After anti-logging equation (7), a correlation in power form was obtained as follows:

\[ N_u_{\text{flat}} = 0.1543 \text{Re}_{\text{dip}}^{0.69} (H/D_j)^{-0.49} \]  

(8)

4.2.1. Impingement on dimples

For the nozzles set inline to the dimples, the heat transfer logarithmic correlation obtained from ANOVA table in Table 2 had small two-sided \( p \)-values. The value of \( R^2 \) of the regression was 0.8563, and this inferred that 85.63% of the variation in \( \log\left( \frac{N_u_{\text{dimp}}}{\text{Re}_{\text{dip}}} \right) \) was explained by the multiple regression to \( \log(\text{Re}_{\text{dip}}), \log(H/D_j), \log(d/D_j), \) and \( \log(D_j/D_d) \). The standard deviation for the regression was 0.0066. The small \( p \)-values presented the overwhelming evidence that the coefficient of at least one of the explanatory variables differed from zero. After anti-logging equation (5), the correlation became

\[ N_u_{\text{dimp}} = 0.1770 \text{Re}_{\text{dip}}^{0.61} (H/D_j)^{-0.23} (d/D_d)^{-0.60} (D_j/D_d)^{0.85} \]  

(9)

Table 2

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<th>Coefficient</th>
<th>Standard error</th>
<th>( t )-Statistics</th>
<th>Two-sided ( p )-value</th>
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Table 3

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<tbody>
<tr>
<td>Summary of the least square fit to the regression of ( \log(\frac{N_u}{\text{Re}<em>j}) ) on ( \log(\text{Re}</em>{\text{dip}}) ) and ( \log(H/D_j), \log(d/D_j) \text{and} \log(D_j/D_d) )</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Constant C</td>
<td>-0.45948</td>
<td>0.36675</td>
<td>-1.25283</td>
<td>0.22472</td>
</tr>
<tr>
<td>( \text{Re}_j )</td>
<td>0.49691</td>
<td>0.10431</td>
<td>4.78892</td>
<td>&lt;0.0001</td>
</tr>
<tr>
<td>( H/D_j )</td>
<td>-0.15868</td>
<td>0.03342</td>
<td>-4.65979</td>
<td>&lt;0.0001</td>
</tr>
<tr>
<td>( d/D_j )</td>
<td>0.31103</td>
<td>0.10406</td>
<td>2.98892</td>
<td>0.00725</td>
</tr>
<tr>
<td>( D_j/D_d )</td>
<td>0.85266</td>
<td>0.08969</td>
<td>9.50999</td>
<td>&lt;0.0001</td>
</tr>
<tr>
<td>( R )</td>
<td>0.8563</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>( R )-square</td>
<td>0.78374</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>( R )-square Adjusted</td>
<td>0.74049</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>SD</td>
<td>0.05731</td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

Fig. 15. Comparison of experimental results on logarithmic axes of flat plate with the literature at \( H/D_j = 4 \).

Fig. 16. Plots of average Nusselt numbers against \( H/D_j \) of various \( \text{Re}_{\text{dip}} \) – impinging on dimple.
4.2.2. Impingement on flat portions

Similar to jets impinging on dimples, the proposed correlation for jets impinging on flat portions (Eq. (6)) was developed. The least square fit to the multiple regression of log (Nu_dj) on log(Re0), log(H/Dj), log(d/Dj), and log(Dj/Dj0) gives the empirical correlation after anti-logging as follows:

$$\text{Nu}_{\text{fan flat}} = 0.3472 \text{Re}_{Dj}^{0.50} (H/Dj)^{-0.16} (d/Dj)^{-0.64} (Dj/Dj_0)^{0.31}$$  \hspace{1cm} (10)

In addition, Eq. (10) gave $R^2 = 0.7837$ and the standard deviation of 0.05731. F-statistic from ANOVA table in Table 3 led to the small p-values for most coefficients. However, the F-statistic of the logarithmic constant from Eq. (6) gave a not-so-small two-side p-value of 0.2247. This was thought to be the cause of the data points for $Re_0 = 11,500$ for both values of $H/Dj$. However, the value of $R^2$ was acceptable. These show in the shifted data points from the calculated plot at $Re_0 = 11,500$ in Figs. 17 and 18.

5. Conclusions

The idea of coupling the effects of impingement and channel flow was seen to assist the heat transfer of impinging jets on dimples. The dimples functioned more effectively when the strong crossflow was present. The crossflow aided the dimple edges in detaching the boundary layer. Different dimple depths led to the different results in the study on the effect of impinging positions; for the shallow dimples, impinging on flat portions resulted in higher heat transfer than impinging on dimples, and vice versa for the deep dimples. As cold regions were found inside the staggered deep dimples (i.e. $d/Dj = 0.25$) located upstream of the crossflow, this suggested that the stronger recirculation occurred. The shallow dimples possessed better vortex source-sink features with the aid of crossflow.

Parametric effects were investigated by varying Reynolds number, jet-to-plate spacing, dimple depth and ratio of jet diameter to dimple diameter. As has been well established by previous researchers, increasing $Re_0$ means increasing the jet exit velocity, hence, the momentum of the jets at the stagnation zones were higher, and thus the heat transfer was increased.

Jet-to-plate spacing strongly affected the heat transfer. The narrow $H/Dj$ such as $H/Dj < 2$ caused vigorous recirculation from the dimple, and caused heat transfer degradation compared with the results of the flat plate. However, the wider spacing such as $H/Dj > 8$ led to the jets losing the momentum to the ambient air, and the heat transfer decreased.

The experimental results of heat transfer of dimple impingement in terms of the Nusselt number were correlated to dimensionless parameters involved in the experiment such as Reynolds number ($Re_0$), jet-to-plate spacing ($H/Dj$), dimple depth ($Dj$), and curvature ($Dj/Dj_0$). Correlations for cases of jets impinging on dimples and flat portions adjacent to dimples were reported as well as that for flat plate. Logarithmic multiple regression was performed, and gave the following correlations.

For impinging on dimples,

$$\text{Nu}_{\text{fan dimple}} = 0.1770 \text{Re}_{Dj}^{0.61} (H/Dj)^{-0.23} (d/Dj)^{-0.66} (Dj/Dj_0)^{0.85}$$

$$R^2 = 0.8563$$

And for impinging on flat portions,

$$\text{Nu}_{\text{fan flat}} = 0.3472 \text{Re}_{Dj}^{0.50} (H/Dj)^{-0.16} (d/Dj)^{-0.64} (Dj/Dj_0)^{0.31}$$

$$R^2 = 0.7837$$

References