Optimization Arrangement of Two Pulsating Impingement Slot Jets for Achieving Heat Transfer Coefficient Uniformity

In this paper, an optimization was performed to achieve uniform distribution of convective heat transfer coefficient over a target plate using two impinging slot (air) jets. The objective function is the root mean square error ($E_{num}$) of the local Nusselt distribution computed by computational fluid dynamic (CFD) simulations from desired Nusselt numbers. This pattern search minimized this objective function. Design variables are nozzle widths, jet-to-jet distance, jet-to-target plate distance, frequency of pulsations (for pulsating jets), and the flow rate. First, an inverse design is performed for two steady jets for simplicity and the obtained errors for three different desired Nusselt numbers, $N_uD_0 = 7, 10,$ and $13$, were $20.73\%$, $20.08\%$, and $22.92\%$, respectively. Uniform distribution of heat transfer coefficient for two steady jets was not achieved. Thus, two pulsating jets are considered. The range of design variables for pulsating state is as same as steady-state and heat transfer rates increased about $400\%$ over steady-state due to the effects of pulsations in inlet velocity. Thus, in the pulsating state, optimization must be performed for the desired Nusselt numbers around four-times $N_uD_0$ in the steady-state, i.e., $N_uD_0 = 28, 40,$ and $52$. The $E_{num}$ reduced less than $0.01\%$ and distribution of heat transfer coefficient for all cases was uniform. An experimental study using an inverse heat conduction method (conjugate gradient method with adjoint equation) has been performed and the experimental results for the case of $N_uD_0 = 52$ are presented. The estimated distribution of Nusselt number on the target plate with the numerical distribution has around $3.2\%$ relative error with optimal configuration. [DOI: 10.1115/1.4033616]

1 Introduction

Impinging jets are widely employed in industrial applications such as gas turbine cooling, rocket launcher cooling, electrical equipment cooling, drying of paper and textiles, and processing of steel and glass, etc. [1]. Heat transfer rate of impingement jets is high. It is possible to get desired heat transfer in jets by altering parameters such as the flow Reynolds number, nozzle to plate distance, and jet-jet spacing.

Some researchers have studied steady jets to achieve uniform heat flux [2–8]. These investigations have used more than two jets. Heat transfer uniformity was investigated by altering effective parameters such as jet–target plate distance, jet–jet spacing, impingement angle, nozzle width (or diameter), and turbulence intensity. It is proposed to use pulsating jets to obtain heat flux uniformity. In homogeneous cooling (or heating), the spatial variation of the temperature is smaller; an important requirement in some applications in which elimination of crack initiation as a result of residual stresses is important.

There are published works on the effect of pulsations on heat transfer of pulsating jets. Zumbrunnen et al. [9] recorded significant enhancement in stagnation point heat transfer for pulse flow for $0.009 < St < 0.042$. Fallen [10] has found no influence of pulsation on heat transfer in laminar flow. Hofmann et al. [16] performed an experimental investigation on flow structure and heat transfer from a single pulsating submerged round jet impinging perpendicularly on a flat plate. They reported heat transfer can be enhanced when $St \geq 0.2$. Persoons et al. [17] have found an enhancement in heat transfer from a pulsating axisymmetric impinging jet due to flow pulsation.

Hewakandamby [18] investigated numerically the effect of pulse velocity on heat transfer for two jets on a flat surface. Based on his results, heat transfer of two oscillating slot jets with phase difference of $\pi/2$ could be improved greatly compared to the use of conventional steady jets. Mohammad Pour et al. [19] examined numerically an optimized arrangement of steady and pulsating impinging multiple submerged slot jets. A composite design consisting of steady as well as intermittent (on/off) or sinusoidal (outlet jet velocity magnitude changes with time sinusoidal) jets in different combinations of four slot jets was simulated. Forouzannemehr et al. [20] achieved to uniform distribution of heat flux over an isothermal heated target surface by using four slot jets. Their results show that heat transfer is not enhanced and convective heat transfer distribution became more uniform in this investigation.

One way to achieve heat flux uniformity is using inverse heat transfer boundary design. In general, an inverse heat transfer problem can be divided into two main groups: estimation of unknown parameter (or function) and inverse design.
The inverse heat transfer problem in first group involves estimation of surface boundary conditions (i.e., the heat flux or temperature [21–23]), thermo physical properties [24,25], an unknown geometry of a section of a body [26], or volumetric heat generation [27] using some temperature measured from locations within or on the surface of the body. Inverse problems are mathematically ill-posed, being highly sensitive to random errors (noise) that inherently exists in measured temperature data. In order to alleviate this problem, regularization techniques are utilized [28]. Inverse design (second group) can be one way for achieving desired distributions of parameters such as constant heat flux [29] or constant temperature [30] on surface or shape design [31].

Few works for achieving uniform convective heat transfer coefficient from an array of pulsating jets have been reported. Electrical equipment was cooled using laminar jets. Turbulent jets were applied in industry.

This work presents an inverse design of two pulsating (air) slot-jets to achieve uniformity of time-average heat transfer coefficient over a heated flat plate in the laminar range. In this case, the inverse boundary design problem is converted to an optimization problem for minimizing heat transfer coefficient error from the desired uniform distribution. The optimization method used is the pattern search method. Both the steady-state and pulsating jets are taken into consideration.

In order to validate the optimization solution, an experiment with the obtained optimal values was performed using the inverse heat conduction in which the distribution of the heat transfer coefficient distribution was estimated using the direct approach. In this method, temperatures are measured only in the target plates which are used as the input data to the inverse algorithm. The methodology of the inverse estimation method is discussed in detail in Sec. 3.

2 Problem Formulation

2.1 Geometry and Design Variables. In order to achieve a uniform heat transfer coefficient distribution, a system consisting of two slot jets is taken into consideration. As shown in Fig. 1(a), two jets having widths of BC = DE = w with an interspacing of CD = s exit the jet plane at a distance of FG = AJ = H from the target surface. The width of the target plate JG = Lplate is equal to 12.5 cm and is constant for all cases. The segment of the plate for which heat flux uniformity is desired is IH = w_t = 7 cm. These lengths are set equal to those in the experimental setup in order to validate the numerical results obtained from the optimization procedure.

Thus, there are five geometrical parameters in this problem, three of which (w, H, and s) are set to design variables. The purpose of this paper is to determine the design variables that lead to uniform convective heat transfer coefficient over the target surface. Two of them (Lplate and w_t) are fixed in all cases.
Among the flow parameters, \( v_{\text{inlet}} \) (inlet velocity) is considered to be the same for the two jets (due to symmetry). First, this problem is studied in steady-state and if this condition does not lead to an acceptable result, transient states will be considered. In this condition, the jet velocity is assumed to oscillate as a sinusoidal to a maximum of \( v_{\text{inlet}} \) with frequency \( f \). It is obvious that steady-state is preferable because of simplicity and low cost if the uniform heat transfer coefficient is achieved.

![Diagram and figure captions](http://heattransfer.asmedigitalcollection.asme.org/ on 04/03/2018 Terms of Use: http://www.asme.org/about-asme/terms-of-use)
Dimensionless geometrical variables ($w/w_t$, $H/w_t$, and $s/w_t$) are used as design variables so that the obtained results are independent of the geometry and can be solved with any desired length of the target surface.

The frequency is expressed by the Strouhal number ($St = f w_t / U_{max}$) that in this definition, the nozzle dimension was replaced with plate dimension, the magnitude of $St$ multiplied. Moreover, due to the limitation of frequency generated by the oscillating valve used in experiments (Fig. 2(a)), frequency must be considered between 20 Hz and 500 Hz. The velocity is nondimensionalized using the Reynolds number, defined as ($Re = p w_{inlet} (4w)/l$). The Nusselt number is defined as ($Nu = h(4w)/k$).

2.2 The Numerical Solution. As mentioned previously, in order to generate uniform convective heat transfer coefficient along a target plate, two impinging slot-jets were used. The changes of temperature are small and it can be assumed that the physical properties are constant over the temperature range. The continuity, momentum, and energy equations for laminar flow in the Cartesian system are given, respectively, as

$$\frac{\partial u_i}{\partial x_i} = 0 \quad (1)$$
\[ \rho \frac{\partial u_i}{\partial t} + \rho \frac{\partial (u_i u_j)}{\partial x_j} = -\frac{\partial P}{\partial x_i} + \rho \frac{\partial}{\partial x_j} \left[ \mu \left( \frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i} \right) \right] + \rho g_i \]  

(2)

and

\[ \rho c_p \left( \frac{\partial T}{\partial t} + \frac{\partial (u_i T)}{\partial x_i} \right) = \frac{\partial}{\partial x_i} \left( k \frac{\partial T}{\partial x_i} \right) \]  

(3)

The computational domain is symmetric with respect to the y-axis, and boundary conditions are described completely in Fig. 1(b). The segment E–F is the confined wall (with no slip and isolation boundary condition) and the segment F–G is the exit plane (with pressure outlet boundary condition). The flow is submerged. The inlet velocity for the pulsating state is given by

\[ u = 0 \]
\[ v = v_{\text{mean}}(1 + A \sin(2\pi f t)) \]  

(4)

where \( A \) is the amplitude of pulsations and \( f \) is the frequency of pulsations. The finite volume method was used for solving the flow and thermal fields. A second upwind discretization scheme was used for advective terms and the SIMPLE algorithm was employed for pressure–velocity coupling. The \( y \)-direction grid was chosen to be nonuniform being refined near the target surface.

In order to guarantee the numerical accuracy and also reduce the computational cost, the grid independence is studied by employing different fine and coarse meshes and the optimum grid size is determined. The grid refinement is continued until halving the grid size resulted in less than 1% change in the computed local \( \text{Nu}_{\text{ave}} \). A grid density of \( 150 \times 50 \) (nonuniform grid) provides satisfactory solution. In the optimization process, the geometry is changed and the mesh is adaptive to these changes. The grid refinements did not result a substantial improvement in the optimal configuration [20]. As this problem is to be solved in an optimization process, number of meshes is changed according to the variation in the input geometric parameters (\( H, w, \) and \( s \)), that is, the mesh is adaptive to these changes.

The time-step also is varied in accordance with the frequency of pulsating jets. Convergence criteria were set for all the residuals being less than 0.0001. For unsteady-state, the time-step independence was checked. Small time-steps were not required in the implicit procedure. Convergence could be guaranteed when time-step size is set at 1/pulsation frequency/10.

By solving conservation equations, velocity, pressure, and temperature fields are obtained. The objective of the numerical calculations is to compute the Nusselt number distribution over the target surface using the temperature field, for input design variables (\( w/w_t, H/w_t, s/w_t, \) Re, and St). The Nusselt number is extracted from the computed temperature field using

\[ \text{Nu}(x, t) = \left( -\frac{\partial T}{\partial y} \right)_{y=0} \times \frac{4w}{(T_s(x, t) - T_{\text{jet}})} \]  

Fig. 4  (a) Convergence history of \( E_{\text{rms}} \) and (b) local Nusselt number at optimal state for \( \text{Nu} = 7, 10, \) and 13 in steady-state
2.3 Inverse Design Formulation. The inverse boundary design heat transfer problems to achieve heat flux uniformity for both steady and pulsating jets are converted to an optimization procedure.

For the steady jet, the objective function to be minimized is defined as the relative standard deviation of the Nusselt number as

\[
 f_{\text{steady}} = \left( \frac{1}{N} \sum_{i=1}^{N} \left( \frac{Nu_D - Nu_i}{Nu_D} \times 100 \right) \right)^2
\]  

(5)

In this case, design variables are \(w/w_t, H/w_t, s/w_t\), and \(Re\).

In the transient pulsating jets case, in which the jet velocity is assumed to be given by Eq. (4), a time-average Nusselt number for each point on the target surface given as

\[
 Nu_{x, \text{average}} = \frac{1}{T} \int_0^T Nu(x, t) dt
\]  

(6)

is utilized. In Eq. (6), \(T\) is the total time of cooling. In every solution of the direct problem, a time-average Nusselt number is obtained for each point. The objective function for the pulsating jet is defined similar to the steady jet as

\[
 f_{\text{transient}} = \left( \frac{1}{N} \sum_{i=1}^{N} \left( \frac{Nu_D - Nu_{i, \text{time average}}}{Nu_D} \times 100 \right) \right)^2
\]  

(7)

As it is difficult to find the global minimum (the best uniformity of heat transfer coefficient) due to the existence of several local minimum values, a pattern search optimization algorithm [32] is used. This is an efficient method for handling nonuniform problems. In order to obtain uniform Nusselt distribution over the target surface for steady-state condition, the following steps have been performed:

1. The initial state (or base) point (design variables) is guessed (that is \(Re, H/w_t, w/w_t, \) and \(s/w_t\)).

2. The CFD solver is called upon in order to compute the Nusselt number distribution over the target surface.

3. The objective function is calculated using Eq. (5).

4. The next base point is determined by the means of a pattern search algorithm (details of which can be found in Ref. [32].)

5. If the convergence criterion is satisfied, the iterations are terminated; otherwise, the procedure is continued by returning to step 2.

The same procedure has been performed in the transient state with the design variables being \(f, Re, H/w_t, w/w_t, \) and \(s/w_t\).

2.4 Limits of Design Variables. It is much easier to set limits for dimensionless design variables. Due to construction limitations, width of the two jets are considered from 2 mm to 7 mm, and also \(H\) varies from 14 mm to 70 mm. Previous studies [1] show that the interaction between the jets is effective when the...
distance between them is somewhere between 2 and 8 times the jet hydraulic diameter. Therefore, \( s \) is varied from 2 mm to 8 mm.

The range of Reynolds numbers is set to be between 100 and 1000 so that the flow remains within the laminar regime. Also, frequency is varied between 20 Hz and 500 Hz. The ranges of these five dimensionless design variables are listed in Table 1. In addition, with reference to Fig. 1, it is obvious that BE must be less than \( w_t \). Thus, the following inequality

\[
2w + s < w_t
\]

must be satisfied. It can be expressed as a function of dimensionless design variables as

\[
2 \frac{w}{w_t} + \frac{s}{w_t} < 1
\]

3 Experimental Setup and Procedure

Figure 2(a) shows the experimental setup. Compressed air (1) flows through an air filter (2), an air regulator/oil filter (3), a needle valve (4), and then a flow meter (Rota meter) (5) to a plenum chamber (7). A pressure gauge (6) is used to correct the flow rate. A baffle plate (8) causes the flow to decelerate and expand in the plenum chamber. One row of stainless steel mesh screen (9) and a honeycomb (10) are placed after the baffle plate in the bottom half of the plenum chamber. Thus, velocity profiles at the nozzle outlets will be uniform. The plenum chamber’s contact zones are sealed with rubber gaskets and silicon glue to hold against any air leakage. A hot-wire probe can be positioned at the right measuring place by using a programmable 3D stage (15) controller.

The jet nozzles are placed at the bottom of the plenum chamber. The nozzle’s width and jet to jet distance are those determined by the results of the optimization process for every case study. The entrance of the 3D vertical structures of slot corner into the heated zone disregarded due to the length of 2D slot jets is considered longer than the target plate (11). The pulsation in air flow is produced using a pulsating valve (20). This valve is driven by a three-phase electric motor. Figure 2(a) shows the schematic of pulsating valve. The target plate is a stainless steel (AISI-304) plate with dimension 250 \( \times \) 70 \( \times \) 5 mm. The target plate surface is polished by grinding process. By using an infrared thermometer gun, the measured plate emissivity is determined to be 0.75 \( \pm \) 0.01. A silicon heater (12) with less than 3 mm thickness placed at the bottom of the plate provides a heat flux of 2000 W/m\(^2\). The power supply (16) was used to turn on the heater. The surface temperatures in the target plate were measured over a distance of seven times the slot width from the stagnation point. Eleven type-K(TP-01) thermocouples are used, one at the stagnation point, nine on the left-hand side, and one on the right side of

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Fig. 6 (a) Convergence history of local Nusselt number and (b) local Nusselt number distribution at optimal state for different iterations at NuD = 40

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the symmetry line to check the symmetry of the heat transfer distribution (Fig. 2(a)). Distance between two adjacent thermocouples is 3.5 mm. Thermocouples were inserted through holes of 4.5 mm depth machined through the thickness of the plate.

The backside of the heater was insulated with wood, fiber glass, and elastomer layer, respectively. The thicknesses of wood, fiber glass, and elastomer insulation were 16 mm, 12 mm, and 6 mm, respectively. The wood–heater surface temperatures were recorded using 11 K-type (TP-01) thermocouples (with accuracy 0.1 °C), with the same arrangement on the target plate. Using the thermocouples in wood and inverse method, distribution of local heat flux’s heater was estimated exactly.

Fig. 7  (a) Convergence history of local Nusselt number and (b) local Nusselt number distribution at optimal state for different iterations at Nu₀ = 52

Fig. 8 Convergence history of E rms for Nu₀ = 28, 40, and 52
In our experimental study, the inverse method used was the conjugate gradients method along with the adjoint equation [33]. This is due to the fact that estimation of the convective heat transfer coefficient is a nonlinear function estimation problem. Using this method, convective heat transfer is estimated directly, as opposed to indirect estimation in which heat flux is estimated using inverse heat conduction problem (IHCP) and heat transfer is derived from using Newton’s law of cooling. The governing heat equation model for target plate is given as (see Fig. 2(b))

\[
\frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial y^2} = \frac{1}{\alpha} \frac{\partial T}{\partial t}
\]

\[
\frac{\partial T}{\partial x} \bigg|_{x=0} = \frac{\partial T}{\partial x} \bigg|_{x=L} = 0
\]

\[
-k \frac{\partial T}{\partial y} \bigg|_{y=0} = q
\]

\[
-k \frac{\partial T}{\partial y} \bigg|_{y=E} = h(T - T_{jet}) + \sigma \left( T^4 - T_{jet}^4 \right) \text{ and } T \bigg|_{t=0} = T_0
\]

where \( q \) is the heat flux of the heater and \( h \) is the unknown. Heat transfer in the target plate was considered two-dimensional. In order to estimate the heat transfer coefficient by using the conjugate gradients method, the error function \( S \) defined as

\[
S(h) = \sum_{i=1}^{N_i} \sum_{m=1}^{N_m} (T_i(t_m) - Y_i(t_m))^2
\]

is minimized. In Eq. (11), \( Y \) is measured temperatures at sensor locations and \( T \) is the corresponding calculated values using the heat equation. The directional derivative of \( S \) for nonlinear problems, that is,

\[
\nabla S = \frac{\partial S}{\partial h} dh
\]

is obtained from the adjoint equation. In Eq. (12), all the required parameters are evaluated at the sensor locations. Using the above equation, the conjugate direction \( d^k \) can be calculated as

\[
d^k = \nabla S(h^k) + \gamma^k d^{k-1}
\]

The conjugate coefficient \( \gamma \) is calculated using

\[
\gamma^k = \frac{\int_{t=0}^{t'} (\nabla S(h^k))^2 dt}{\left( \int_{t=0}^{t'} (\nabla S(h^{k-1}))^2 dt \right)}
\]

where \( \gamma_0 = 0 \). If \( h' = h^k + d^k \) is substituted in heat Eq. (1), then values of \( \Delta T \) will be calculated at the sensor location as

\[
\Delta T = T(h) - T(h^k)
\]
Therefore, the search step size ($\beta$) can be obtained as

$$\beta^k = \int_0^t \left( T(x, y, t; h^k) - Y(t) \right) \Delta T(x, y, t; h^k) dt / \int_0^t \Delta T(x, y, t; h^k)^2 dt$$  \hspace{1cm} (16)

In this method, an iterative procedure is used to estimate the imposed heat transfer coefficient. This iterative method can be summarized by

$$h^{k+1} = h^k - \beta^k \delta^k$$  \hspace{1cm} (17)

where “$\delta$” is a conjugate direction and $h$ is the search step size. Suppose $h$ is available at iteration $n$, then the computational procedure for the solution of this inverse problem may be summarized as following:

**Step 1.** Solve the direct problem for $T$.

**Step 2.** Examine the stopping criterion considering $S(h) < \tilde{\alpha}$ where $\tilde{\alpha}$ is a small-specified number. Continue if it is not satisfied.

In this case, the iterations were stopped when the residuals between measured and estimated temperatures are of the same order of magnitude of the measurement errors. That is, $|Y(t) - T(X, t)| < (i.e.,$ standard deviation) $\Delta T$.

**Step 3.** Compute the gradient of the functional $\nabla S$ from Eq. (12).

**Step 4.** Compute the conjugate gradient $\delta^k$ and direction of descent $\delta^k$ from Eqs. (13) and (14), respectively.

**Step 5.** Set $\Delta h = \delta^k$ in the heat equation of the problem, and solve for calculating $\Delta$. The second integral term on the right-hand side of this equation is simplified by integration by parts and by utilizing the boundary and initial conditions of Eq. (10). After some manipulation, the adjoint differential equations are obtained as

$$\lambda_{xi} + \lambda_{yi} + 2 \sum_{i=1}^{N_s} \frac{(T - Y) \delta(x - x_i) \delta(y - y_i)}{\lambda_i} = -\lambda_i / \alpha$$

$$\lambda_{xi=0} = \lambda_{xi=1} = 0$$

$$\lambda_{yi=0} = 0$$

$$ds = dL = \frac{\partial L}{\partial h} dh = \nabla s = -\lambda(x, E, t) T(x, E, t - T_m)/k$$  \hspace{1cm} (22)

where $\delta s$ is the derivation of the objective function, $x_i$ and $y_i$ are the position of thermocouples in the target plate. Note that in the adjoint problem, the condition $\lambda(t_f) = 0$ is the value of the function $\lambda(x, y, t)$ at the final time $t = t_f$. Thus, this equation must be
solved backward. The gradient of the objective function is obtained from the adjoint equations.

The uncertainties are related to measuring devices (Table 2) such as thermocouples, Rota meter, Machining process, laser cutting machine, etc. Using the method proposed in Ref. [34], the maximum range of relative overall uncertainty in estimated Nusselt number in this experiment is $4.7 \pm 2.2\%$.

4 Results and Discussion

The numerical results of this model for an unsteady semiconfined air slot jet were compared with the original study of Ref. [35] at $Re = 500$ and $H/W = 5$. Figure 3(a) shows the computational domain with boundary conditions for a semiconfined air slot jets (the mesh is the same described in Sec. 2.2). This comparison is shown in Fig. 3(b). The results show good agreement.

Also, the mentioned experimental method and computational model were performed in a steady and pulsating semiconfined air slot jet at $Re = 900$ and $H/D_h = 4$. The obtained results are shown in Figs. 3(c) and 3(d). The numerical results are in good agreement with experimental results and the maximum deviation with experimental results in the pulsating state for $f = 80$ Hz (Fig. 3(d)) occurs in stagnation region. It can be due to the laminar flow assumption in the CFD simulation.

According to the literature [20], four jets is the minimum number of slot jets that are used to obtain uniform heat flux on the iso-thermal plate. Achievement to this goal with increasing number of jets is simpler than two jets. In this paper, uniform heat transfer coefficient by two jets is investigated. The objective function is highly nonlinear and the same initial guesses are used for all desired Nusselt numbers.

4.1 Steady-State. Using the method discussed in Secs. 2.2 and 2.3, the optimal state leads to 20.73%, 20.08%, and 22.92% error for $Nu_D = 7, 10, and 13$, respectively. Figures 4(a) and 4(b) show convergence history of error and Nusselt distribution in the optimal state, respectively. Therefore, heat transfer coefficient uniformity cannot be achieved satisfactorily using two jets in the steady-state condition.

4.2 Transient State. In this case, jet velocity is assumed as Eq. (4). Amplitude of pulsations is half of the inlet velocity. Time-average Nusselt number for 100 points with equal distances on the target surface from its center of symmetry to point H is considered. In every solution of direct problem, time-average Nusselt number is obtained for all points. Sinusoidal flow pulsations can be induced nonsinusoidal responses because of nonlinearities in momentum and energy transfer within boundary layer. The nonlinear dynamic effects reflect that disturbances associated with flow pulsations do not allow momentum and energy transfer to equilibrate within the boundary layer. Thus, both thermal and velocity boundary layers are not fully formed. The thermal boundary layer is disrupted periodically. A stagnation point similar to that of steady jets is formed where the wall jets interact. The stagnation point is not fixed and vibrates around a fixed position due
Table 3 Initial and optimal nondimensional design variables for two jets for steady and transient state

<table>
<thead>
<tr>
<th></th>
<th>NuD = 7</th>
<th>NuD = 10</th>
<th>NuD = 13</th>
</tr>
</thead>
<tbody>
<tr>
<td>w/</td>
<td>Initial</td>
<td>Optimal</td>
<td>Initial</td>
</tr>
<tr>
<td>w/</td>
<td>0.0429</td>
<td>0.0429</td>
<td>0.0429</td>
</tr>
<tr>
<td>s/</td>
<td>0.2857</td>
<td>0.2857</td>
<td>0.2857</td>
</tr>
<tr>
<td>Re</td>
<td>500</td>
<td>215.8269</td>
<td>500</td>
</tr>
<tr>
<td>E_{rms}(%)</td>
<td>33.1564</td>
<td>20.7335</td>
<td>29.8640</td>
</tr>
</tbody>
</table>

Pulsating jets

<table>
<thead>
<tr>
<th></th>
<th>NuD = 28</th>
<th>NuD = 40</th>
<th>NuD = 52</th>
</tr>
</thead>
<tbody>
<tr>
<td>w/</td>
<td>Initial</td>
<td>Optimal</td>
<td>Initial</td>
</tr>
<tr>
<td>w/</td>
<td>0.0429</td>
<td>0.0429</td>
<td>0.0429</td>
</tr>
<tr>
<td>s/</td>
<td>0.2857</td>
<td>0.2857</td>
<td>0.2857</td>
</tr>
<tr>
<td>Re</td>
<td>500</td>
<td>116</td>
<td>500</td>
</tr>
<tr>
<td>E_{rms}(%)</td>
<td>3.5278</td>
<td>11.4299</td>
<td>3.5278</td>
</tr>
</tbody>
</table>

To achieve uniform convective heat transfer coefficient and the flow pattern and temperature contour are completely studied in previous papers such as Refs. [10–18].

In this case, heat transfer to target surface is increased dramatically by choosing the same parameters as the steady-state. This augmentation is due to inlet jet fluctuations. It is observed that Nusselt number in pulsating state is four times the Nusselt number in steady-state with regard to the mentioned range for design variables (Table 1). This matter must be considered and optimization must be performed for the desired Nusselt numbers, 28, 40, and 52 in pulsating state.

In the optimization process, Nusselt numbers of all points on the target surface approach the target Nusselt number as much as possible. Convergence histories of Nusselt distribution for all three target Nusselt numbers in specific iterations and Nusselt distribution in the optimal state for target Nusselt number equal to 28, 40, and 52 are illustrated in Figs. 5–7, respectively. Also, convergence history of error ($E_{rms}$) and design variables ($w/w_i$, $H/w_i$, $s/w_i$, Re, and St) for three Nusselt number are demonstrated in Figs. 8–13, respectively. The initial and optimum values of design variables and errors are listed in Table 3. The objective function is highly nonlinear. Number of iterations for achievement of the desired Nusselt distribution are different for each desired Nu. Design variables in optimization process change to get to desired Nusselt number with minimum $E_{rms}$. The error is reduced to less than 0.01% and decreased dramatically compared to the steady-state. Table 3 shows the changes of St are inversely proportional to Re due to the existent pulsations in jet inlet velocity. The pulsations cause to increase mean velocity in the laminar jet flow field. Thus, a parameter must be considered that shows the effects of the pulsations and jet velocity. This parameter is Re × St. Its values for NuD = 28, 40, and 52 are 1325.868, 1878.249, and 2086.510, respectively. It is observed that the Re × St increased with rising the desired Nusselt number. The nozzle width ($w/w_i$) increases with rising the desired Nusselt number.

In order to verify the uniformity of local Nusselt number, the optimal result is tested experimentally and the experimental results for NuD = 52 are presented here. The obtained optimal values are rounded according to Table 4 for preparing the experimental setup.

The result of experiment base of Sec. 3 is obtained and the Nusselt distribution over the target surface is illustrated in Fig. 14. In this figure, it is evident that numerical and experimental data are close to each other and experimental data have a relative error of 3.2% from the uniform target Nusselt number of 52. This error is due to experimental conditions and experimental limitations in creating an accurate quantity of optimized design variables and rounded optimal values.

5 Conclusion

In this research, an optimization was performed to achieve uniform distribution of convective heat transfer coefficient over a target plate using two impinging slot (air) jets. The optimization algorithm was the pattern search method. The objective function is the root mean square deviation of the local Nusselt distribution calculated from CFD simulations from desired Nusselt number. This optimization minimized this objective function. First, the steady-state for NuD = 7, 10, and 13 is considered because of simplicity and low cost. The minimum variance in the Nusselt number distribution over the target surface was no less than 20%, which was not satisfactory. Thus, the pulsating jets were considered. The heat transfer rate increased by about 400% as compared to the steady jet for the range of design variables. Thus, the design values of the Nusselt number were set at 28, 40, and 52. In this case, uniform heat transfer coefficient was achieved with a variance of less than 0.01%. An experimental validation study was performed for the NuD = 52 case. The inverse method for the
estimation local Nusselt number was used. The experimental results were in good agreement with the numerical results.

Nomenclature

\[ E = \text{plate thickness (m)} \]
\[ E_{\text{rms}} = \text{root-mean square error} \]
\[ F = \text{frequency (Hz)} \]
\[ h = \text{heat transfer coefficient (W/m}^2\text{K)} \]
\[ H = \text{jet-to-surface spacing to hydraulic diameter ratio} \]
\[ k = \text{thermal conductivity (W/m K)} \]
\[ K = \text{turbulence kinetic energy} \]
\[ L = \text{plate length (m)} \]
\[ M = \text{time index} \]
\[ N = \text{number of discrete measurements} \]
\[ N_s = \text{number of sensors} \]
\[ N_u = \text{Nusselt number (h(4\nu)/k)} \]
\[ P = \text{pressure (Pa)} \]
\[ Re = \text{Reynolds number (\rho u D/\mu)} \]
\[ S = \text{sum of squares (K^2)} \]
\[ St = \text{Strouhal number (f H_u/1_\text{max})} \]
\[ T = \text{vector of calculated temperatures (K)} \]
\[ u = \text{x-velocity (m/s)} \]
\[ v = \text{y-velocity (m/s)} \]
\[ W = \text{slot width (m)} \]
\[ x, y = \text{space coordinates} \]
\[ Y = \text{measured temperatures (K)} \]

Greek Symbols

\[ \alpha = \text{thermal diffusivity (m}^2\text{/s)} \]
\[ \sigma = \text{standard deviation} \]
\[ \varepsilon = \text{surface emissivity coefficient} \]

Subscripts

\[ D = \text{desired} \]
\[ D_{\text{desired}} = \text{(Nusselt Number)} \]
\[ \text{Ins} = \text{insulation} \]
\[ J = \text{position index} \]
\[ \text{Surf} = \text{surface} \]
\[ 0 = \text{initial state} \]

References