EXPERIMENTAL STUDY ON HEAT TRANSFER AND FLUID-FLOW ENHANCEMENT OF A SPHERICAL SHAPE OBSTACLE SOLAR AIR PASSAGE

by

Ashok Kumar BHARDWAJ\textsuperscript{a}, Anil KUMAR\textsuperscript{a}\textsuperscript{*}, Rajesh MAITHANI\textsuperscript{b}, Raj KUMAR\textsuperscript{a}, Sunil KUMAR\textsuperscript{a}, and Ranchan CHAUHAN\textsuperscript{a}

\textsuperscript{a}School of Mechanical and Civil Engineering, Shoolini University, Solan, India
\textsuperscript{b}Department of Mechanical Engineering, DIT University, Deharadun, India

Original scientific paper
https://doi.org/10.2298/TSCI170623220S

This paper presents the outcome of experimental examined of Nusselt number and friction factors in a spherical obstacles solar air passage. Investigation has been performed to examine the thermal and hydraulic data from a solar air passage with spherical obstacles on the heated wall. The Reynolds number base on the hydraulic diameter of the solar air passage varied from 45.00 to 16.500, relative sphere diameter varied from 0.130 to 0.217, stream wise spacing of 4.04 and span wise spacing of 4.04. Experimental results pertinent to heat transfer and pressure drop was determined for various sets of roughness and flow parameters. The experimental results show that the heat transfer is increased around 4.7 times than plane surface solar air passage. The thermal and hydrodynamic performance parameter based on equal pumping power was found to be highest for spherical shape dia of 0.195. The superior value of overall thermal performance parameter is 2.83 corresponding to spherical shape dia of 0.195.

Key words: thermal behaviour, flow passage, stream wise spacing, sphere diameter

Introduction

Renewable energy can minimize our dependency on fossil fuels, thereby, renewable energy is getting importance in the recent years because energy can renew and will never run out. Renewable energy is eco-friendly and results in little to no effect to the environment [1]. Out of many renewable energies, solar energy is considerable to be clean source of energy and available on every part of the world [2]. Solar energy is exploited in many application, included heating purposes and generation of electricity [3]. Solar air passage (SAP) is one of the most economical and elementary device which employ to supply the heated air to drying the crops, industrial purposes, heating the building and space [4]. The techniques of local heat transfer improvement attract the interests of researchers [5]. Obstacles are often used to improve local heat transfer among the wall and fluid because they cause stream separation and reattachment, consequently resulting in destroying the laminar viscous layer [6-12]. Bhushan and Singh [8] experimentally investigated the performance of a staggered dimple type roughness SAP. The outcome indicates that the greatest improvement of and factor was 3.12 and 4.16 times, respectively, in comparison to smooth passage. Chang \textit{et al.} [9] examine the comparative full-field distribution on two opposing improved passage walls, including and the thermal performance factor of the two radially rotating obstacles passage with and without dimpled obstacles.

\textsuperscript{*}Corresponding author, e-mail: anilkumar88242@gmail.com
Shen et al. [10] examine the effect of rotation on fluid stream and heat transfer performance of turbine blade with U-shaped passage with the combined structure of obstacles, dimples or protrusions. The outcome shows that rib-protrusion structure found to be the most efficient structure while rib-dimple structure has only minor advantage than ribbed passage. Kumar and Kim [11] investigated the thermal hydraulic performance of a 3-D obstacles-roughened SAP having $W_p/H_p$ of 12.0. They found that thermal hydraulic performance for V-pattern shaped obstacles combined with dimpled obstacles is superior as compared with dimpled obstacles shape and V-pattern obstacles shape SAP. Lian et al. [12] investigated $N_u$ and $f_r$ behaviours of air stream through a passage with hemispherical protrusion/dimple on the heated plate. The outcome shows that the hemispherical dimple roughened air passage is the better choice as compared with smooth passage.

Negi and Pattamatta [13] deal with shape determination of dimples on the target plane in multi-jet impingement, $N_u$. They revealed that the standard deviation in $N_u$, was considerably higher than the reference spherical dimpled profile and the optimized dimple profile shows highest local $N_u$ values calculated to the reference semi-circular dimpled plate optimized form which can be used to get better local temperature hot spots on target surface. Jin et al. [14] presented a numerical study of $N_u$, and $f_r$, characteristics in a SAP channel having multi V-shaped ribs on the absorber plate. It was found that for the range of investigated factors the highest value of the $\eta_p$ parameter was achieved to be 1.93. Ekadewi et al. [15] numerically investigated the influence of delta-shaped obstacles spacing on $N_u$, and pressure drop in V-corrugated canal of SAP. Authors obtained that $N_u$, was improved by 3.46 times and $f_r$, was increased up to 19.9 times.

The literature survey shows that obstacles of dimple, hemispherical, V-type, protrusion and multiple type shapes have been investigated previously which is common shape. In order to increase heat transfer further spherical obstacles have been investigated in this work with aim of different sphere diameter would contribute the turbulence in the flow resulting high heat transfer rate from heated plate. In this regards, experimental study has been conducted to investigate the effect of relative sphere diameter of obstacles attached to heated plate on thermal behaviour.

**Table 1. Ranges of spherical obstacles parameters**

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Range</th>
</tr>
</thead>
<tbody>
<tr>
<td>$D_s/D_H$</td>
<td>0.130-0.217</td>
</tr>
<tr>
<td>$H_s/D_s$</td>
<td>4.04</td>
</tr>
<tr>
<td>$Y_s/D_s$</td>
<td>4.04</td>
</tr>
<tr>
<td>Reynolds number</td>
<td>4.500-16.500</td>
</tr>
</tbody>
</table>

The geometrical parameters for the SAP with spherical obstacles are hydraulic diameter of passage, $D_H$, of 46.15, height of passage, $H_R$, of 25 mm, width of passage, $W_p$, of 300 mm, and length of passage, $L_R$, of 2395 mm. The dimensionsless parameters are relative sphere diameter, $D_s/D_H$, stream wise spacing, $X_s/D_s$, and span wise spacing, $Y_s/D_s$. In this experimental investigation the SAP has $L_R = 1100$ mm, $H_R$ is adjusted on 50 mm, and $W_p$ is set at 300 mm and the $D_H = 46.15$ mm, and different diameter of sphere, $D_s$. The ranges of different parameters are depicted in tab. 1. The schematic and photographic view of spherical shape obstacles are presented in figs. 1 and 2.
Experimental program and procedure

Experimental approach has been adopted to produce the data in form of $N_{uss}$ and $f_s$ for air passage with spherical shape obstacle roughness to search the effect of $D_s/D_H$ and Reynolds number on $N_{uss}$, and $f_s$. The experimental study encompasses the fabrication and installation of indoor test facility. The experimental set-up has been validated by comparing experimental data collected on without spherical obstacle wall with the available standard data. After validation of experimental set-up, extensive experimentations have been conducted on spherical shape obstacle to produce raw data on heated wall temperatures, air stream rates, and entrance and exit temperature of air and pressure drop across the passage under stable conditions. To examine the influence of spherical shape obstacle turbulent promoter on $N_{uss}$ and $f_s$ of air stream, an experimental set-up was designed and made-up according to ASHRAE standard [16]. A schematic diagram of an experimental set-up is shown in fig. 3. The experimental set-up comprised a rectangular wooden channel coupled to a centrifugal blower through a circular galvanized iron (GI) pipe. The rectangular channel had $W_P$ of 300 mm, $H_P$ of 30 mm, and $W_P/H_P$ of 10. The examination was carried out to achieve the experimental values for $N_{uss}$ and $f_s$ in an air stream passage provided with spherical shape obstacle to enhance $N_{uss}$ and $f_s$ descriptions with respect to individual obstacle deviations.

The manufacturing and the appropriate setting of the experimental test set-up were performed, which were validated with existing criterion data on the air passage with smooth surface passage. The optimal validation was achieved to perform advanced examinations with spherical shape obstacle. The testing for air temperature at the entry of the passage and the exit, pressure drop of air from corner to corner of the passage, and the heated wall temperature have been approximated, which is necessary to meet the aims of the investigation. In order to examine the influence of spherical shape obstacle, flat solar air rectangular passage functioning under same

![Figure 2. Photographic view of spherical roughened shape absorber plate](image)

![Figure 3. Schematic of experimental set-up](image)
flow situations was also investigated. Before performing each investigational run, large care was taken to ensure appropriate functioning of all apparatus and that there was no seepage at the joints in the testing set-up. The equivalent instruments provided the data beneath stable condition which was supposed to have been attained when there was no appreciable difference in passage air and collector plate temperature was noticed over a time period of larger than 12 minutes. The following amount of data was reported for each:

- pressure head variation across the orifice plate in order to determine the air-flow rate,
- heated plate temperatures at variant plate positions,
- temperatures of inlet air,
- temperatures of passage air, and
- pressure head drop across the test segment.

**Data reduction**

The data composed have been used to determine Nusselt number and pressure drop. Relevant expressions for the computation of the previous parameters and some intermediate parameters have been given.

The mean temperature of the plate is the average of all temperatures of the heated plate:

\[
T_p = \frac{\sum T_{pi}}{N}
\]  

(1)

The mean air temperature is a simple arithmetic mean of the inlet and outlet temperature of air flowing through the test section:

\[
T_f = \frac{T_i + T_o}{2}
\]  

(2)

where \( T_o = (T_{o1} + T_{o2} + T_{o3})/3 \), \( T_i = T_{A1} \).

Mass-flow rate of air, \( \dot{m}_a \), has been calculated from the pressure drop measurement across the calibrated orifice meter by using the following formula:

\[
\dot{m}_a = C_d A_d \left[ \frac{2 \rho_a (\Delta p)_h}{1 - \beta_R^4} \right]^{0.5}
\]  

(3)

where \( (\Delta p)_h = 9.81(\Delta p)_h \rho_a \dot{m}_a \sin \theta \).

The velocity of air, \( V \), is calculated from the mass-flow rate and given by:

\[
V = \frac{\dot{m}_a}{\rho_a W_p H_p}
\]  

(4)

The hydraulic diameter, \( D_H \), is given by:

\[
D_H = \frac{4(W_p H_p)}{2(W_p + H_p)}
\]  

(5)

The Reynolds number of the air-flow in the rectangular channel is determined:

\[
\text{Re} = \frac{V D_H}{\nu_a}
\]  

(6)

The friction factor, \( f \), is calculated from the measured value of \( (\Delta p)_d \) across the test section length using the Darcy equation:

\[
f = \frac{2 (\Delta p)_d D_H}{4 \rho_a L V^2}
\]  

(7)
where \( (\Delta p)_m = 9.81 (\Delta h) \rho \beta \rho_{m_a} \).

The useful heat gained by air is calculated:

\[
Q_a = m_a c_p (T_0 - T_f)
\]

The heat transfer coefficient for the heated test section has been calculated from:

\[
h_i = \frac{Q_a}{A_p (T_p - T_f)}
\]

The \( h_i \) can be used to determine the Nusselt number, which is given by:

\[
Nu = \frac{h_i D_H}{K_a}
\]

### Uncertainties analysis

Uncertainty is the possible numerical value of the error encountered during experiment. To evaluate uncertainty involve in this experiment method suggested by Kline and McClintock [17] is used. If the data of any parameter is calculated using certain measured quantities then error in measurement of \( y \) (parameter) is given:

\[
\delta y_y = \left[ \left( \frac{\delta y}{\delta x_1} \right)^2 + \left( \frac{\delta y}{\delta x_2} \right)^2 + \left( \frac{\delta y}{\delta x_3} \right)^2 + \ldots + \left( \frac{\delta y}{\delta x_n} \right)^2 \right]^{0.5}
\]

where \( \delta x_1, \delta x_2, \delta x_3, \ldots, \delta x_n \) are the possible error in measurement of \( x_1, x_2, x_3, \ldots, x_n \), \( \delta y \) is the absolute uncertainty and \( \delta y/y \) is known as relative uncertainty.

The important parameters considered for the calculation of uncertainty are: Reynolds number, heat transfer coefficient, Nusselt number, friction factor, etc.

- uncertainty in area of absorber plate:
  \[
  A_p = W_p L_i
  \]
  \[
  \frac{\delta A_p}{A_p} = \left[ \left( \frac{\delta L_i}{L_i} \right)^2 + \left( \frac{\delta W_p}{W_p} \right)^2 \right]^{0.5}
  \]

- uncertainty in area of flow:
  \[
  A_p = W_p H_p
  \]
  \[
  \frac{\delta A_p}{A_p} = \left[ \left( \frac{\delta H_p}{H_p} \right)^2 + \left( \frac{\delta W_p}{W_p} \right)^2 \right]^{0.5}
  \]

- uncertainty in mass-flow rate measurement:
  \[
  \frac{\delta m_a}{m_a} = \left[ \left( \frac{\delta C_{do}}{C_{do}} \right)^2 + \left( \frac{\delta A_s}{A_s} \right)^2 + \left( \frac{\delta \rho}{\rho} \right)^2 + \left( \frac{\delta (\Delta h)_{m_a}}{(\Delta h)_{m_a}} \right)^2 \right]^{0.5}
  \]

- uncertainty in measurement of air velocity in channel:
  \[
  \frac{\delta \dot{m}_a}{\dot{m}_a} = \frac{\delta \rho \dot{m}_a}{\dot{m}_a}
  \]

- uncertainty in measurement of air velocity in channel:
  \[
  V = \frac{\dot{m}_a}{\rho_a W_p H_p}
  \]
\[
\frac{\delta V}{V} = \left[ \frac{\left( \frac{\delta \dot{m}_a}{m_a} \right)^2 + \left( \frac{\delta \rho_a}{\rho_a} \right)^2 + \left( \frac{\delta W_p}{W_p} \right)^2 + \left( \frac{\delta H_p}{H_p} \right)^2}{V} \right]^{0.5}
\]

- uncertainty in useful heat gain:
\[
Q_a = m_a c_p (T_o - T_i) = m_a c_p \Delta T
\]
\[
\frac{\delta Q_a}{Q_a} = \left[ \left( \frac{\delta \dot{m}_a}{m_a} \right)^2 + \left( \frac{\delta c_p}{c_p} \right)^2 + \left( \frac{\delta \Delta T}{\Delta T} \right)^2 \right]^{0.5}
\]

- uncertainty in heat transfer coefficient:
\[
h_i = \frac{Q_a}{A_p \left( T_p - T_f \right)} = \frac{Q_a}{A_p \Delta T_f}
\]
\[
\frac{\delta h_i}{h_i} = \left[ \left( \frac{\delta Q_a}{Q_a} \right)^2 + \left( \frac{\delta A_p}{A_p} \right)^2 + \left( \frac{\delta \Delta T_f}{\Delta T_f} \right)^2 \right]^{0.5}
\]

- uncertainty in Nusselt number:
\[
\text{Nu}_{rs} = \frac{h_i D_H}{K_a}
\]
\[
\frac{\delta \text{Nu}_{rs}}{\text{Nu}_{rs}} = \left[ \left( \frac{\delta h_i}{h_i} \right)^2 + \left( \frac{\delta D_H}{D_H} \right)^2 + \left( \frac{\delta K_a}{K_a} \right)^2 \right]^{0.5}
\]

- uncertainty in Reynolds number:
\[
\text{Re} = \frac{V D_H}{\nu_a} = \frac{\rho_a V D_H}{\mu}
\]
\[
\frac{\delta \text{Re}}{\text{Re}} = \left[ \left( \frac{\delta V}{V} \right)^2 + \left( \frac{\delta D_H}{D_H} \right)^2 + \left( \frac{\delta \nu_a}{\nu_a} \right)^2 + \left( \frac{\delta \mu}{\mu} \right)^2 \right]^{0.5}
\]

- uncertainty in friction factor:
\[
f_{rs} = \frac{2(\Lambda_p)_{ls} D_H}{4 \rho_a L \nu^2}
\]
\[
\frac{\delta f_{rs}}{f_{rs}} = \left( \frac{\delta D_H}{D_H} \right)^2 + \left( \frac{\delta \nu_a}{\nu_a} \right)^2 + \left( \frac{\delta L}{L} \right)^2 + \left( \frac{\delta \rho_a}{\rho_a} \right)^2 + \left( \frac{\delta (\Lambda_p)_{ls}}{(\Lambda_p)_{ls}} \right)^2 \right]^{0.5}
\]

- uncertainty in thermohydraulic performance parameter:
\[
\eta_p = \left( \frac{\text{Nu}_{rs}}{\text{Nu}_{sa}} \right) / \left( f_{rs} / f_{rs} \right)^{0.33}
\]
\[
\frac{\delta \eta_p}{\eta_p} = \left[ \left( \frac{\delta \text{Nu}_{rs}}{\text{Nu}_{rs}} \right)^2 + \left( \frac{\delta f_{rs}}{f_{rs}} \right)^{0.5} \right]
\]
As the uncertainty calculation was done on a single test run (constant Reynolds number), the uncertainty analysis for complete test run for single geometry (complete set of Reynolds number) was carried out and results are presented in tab. 2 for the experimental data.

Results and discussion

The Nu, and f, descriptions of an impingement jet SAP roughened with multiple arcs protrusion ribs, calculated on the sources of investigational data collected for different stream and roughness factors, are discussed.

Validation of experimental set-up

The value of Nu, and f, calculated through experimental outcomes for a smooth channel have been compared with the outcomes obtained from the Dittus-Boelter equation, eq. (22), for the Nu, and modified Blasius equation, eq. (23), for the Kumar and Kim [1].

The Nu, for a smooth passage is given by the Dittus-Boelter equation:

$$\text{Nu}_{ss} = 0.023 \text{Re}^{0.8} \text{Pr}^{0.4}$$

(22)

The f, for a smooth passage is given by the modified Blasius equation:

$$\text{f}_{ss} = 0.085 \text{Re}^{-0.25}$$

(23)

The comparison of the experimental and estimated outcomes of Nu, and f, as a function of the Reynolds number is shown in figs. 4(a) and 4(b), respectively. The average absolute percentage deviation of the experimental Nu, is 5.78% from the value predicted by eq. (22), and the average absolute percentage deviation of the present experimental f, is 4.98% from the value predicted by eq. (23). Thus there is a good agreement between the predicted values and the experimental values of the Nu, and f,. This ensures the accuracy of the experimental data obtained from the present set-up within reasonable limits.

Heat and fluid-flow

The experimental analysis has been perforated for a blockage SAP with spherical obstacles on an absorber plate, and the results are discussed in this section. The results of D3/DH

Table 2. Range of uncertainty in the measurement of essential parameters

<table>
<thead>
<tr>
<th>Parameters</th>
<th>Error range [%]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Heat transfer coefficient</td>
<td>3.98-6.12</td>
</tr>
<tr>
<td>Nusselt number</td>
<td>3.89-6.55</td>
</tr>
<tr>
<td>Friction factor</td>
<td>2.24-4.15</td>
</tr>
<tr>
<td>Thermohydraulic performance parameter</td>
<td>3.45-6.89</td>
</tr>
</tbody>
</table>

Figure 4. Comparison of experimental results with correlation results (a) Nu, (b) f,
on $\text{Nu}_r$, $\text{Nu}_s/\text{Nu}_r$, $f_r$, and $f_s/f_r$ for air-flow are represented in a SAP. The outcomes have been compared with those obtained in case of without obstacles surface working under similar experimental conditions.

The outcomes of $\text{Nu}_r$ have been represents as a function of Reynolds number for the various values of $D_S/D_H$ in fig. 5(a), and for constant values of the other blockage parameters such as $X_S/D_S = 4.04$ and $Y_S/D_S = 4.04$. It has been seen that the $\text{Nu}_r$ increase with increase in the $D_S/D_H$ and attains a highest value matching to a $D_S/D_H$ value of 0.195 in the range of the parameters studied. In all cases, the presence of a surface with spherical blockage produces maximum $\text{Nu}_r$ compares to the without sphrical blockage passage. The spherical blockage can lead to superior $\text{Nu}_r$ performance because of the secondary flow vortices induced by the upper part of spherical blockage. These secondary flow vortices have the form of more than one counter rotating vortices, which carry cold air from the middle core region towards the spherical blockage surfaces. These secondary flow vortices interact with the primary stream, thus affecting the flow reattachment and recirculation between spherical blockage and interrupt the boundary-layer enlargement down ward of the re-attachment regions.

Figure 5(b) presents the values of $\text{Nu}_r$ as function of $D_S/D_H$ for the selected Reynolds number values where a superior in the values corresponding to a $D_S/D_H = 0.195$ for all Reynolds number. The outcomes of $\text{Nu}_r/\text{Nu}_w$ have been represents as a function of Reynolds number for the various values of $D_S/D_H$ in fig. 6(a), and for constant values of the other blockage parameters such as $X_S/D_S = 4.04$ and $Y_S/D_S = 4.04$. It has been seen that the $\text{Nu}_r/\text{Nu}_w$ increase with increase in the $D_S/D_H$ and attains a highest value matching to a $D_S/D_H$ value of 0.195 in the range of the parameters studied. Figure 6(b) presents the values of $\text{Nu}_r/\text{Nu}_w$ as function of $D_S/D_H$ for the selected Reynolds number values where a superior in the values corresponding to a $D_S/D_H = 0.195$ for all Reynolds number.

![Figure 5. (a) Variation of $\text{Nu}_r$ with Reynolds number at different $D_S/D_H$ (b) variation of $\text{Nu}_r$ with $D_S/D_H$ at selected Reynolds number](image)

The outcomes of $f_r$ have been represents as a function of Reynolds number for the various values of $D_S/D_H$ in fig. 7(a), and for constant values of the other blockage parameters such as $X_S/D_S = 4.04$ and $Y_S/D_S = 4.04$. It has been seen that the $f_r$ increase with increase in the $D_S/D_H$ and attains a highest value matching to a $D_S/D_H$ value of 0.217 in the range of the parameters studied. In all cases, the presence of a surface with sphercial blockage produces maximum $f_r$ compares to the without sphercial blockage passage. Figure 7(b) presents the values of $f_r$ as function of $D_S/D_H$ for the selected Reynolds number values where a superior in the values corresponding to a $D_S/D_H = 0.217$ for all Reynolds number. The outcomes of of $f_s/f_r$ have been represents as a
function of Reynolds number for the various values of \( D_s/D_H \) of in fig. 8(a), and for constant values of the other blockage parameters such as \( X_s/D_s = 4.04 \) and \( Y_s/D_s = 4.04 \). It has been seen that the \( f_{rs}/f_{ss} \) increase \( D_s/D_H \) with increase in the \( D_s/D_H \) and attains a highest value matching to a \( D_s/D_H \) value of 0.217 in the range of the parameters studied. Figure 8(b) presents the values of \( f_{rs}/f_{ss} \) as function of \( D_s/D_H \) for the selected Reynolds number values where a superior in the values corresponding to a \( D_s/D_H = 0.217 \) for all Reynolds number.
Thermal hydraulic performance

The SAP with spherical blockage results in highest $\text{Nu}_{rs}/\text{Nu}_{ss}$ as well as $f_{rs}/f_{ss}$ compared to without spherical blockage SAP. So an overall thermal performance needs to be calculated that takes into account both $\text{Nu}_{rs}/\text{Nu}_{ss}$ as well as $f_{rs}/f_{ss}$ to evaluate usefulness. A overall thermal performance parameter based on equal pumping power explained by Webb and Eckert [18] considered both the $\text{Nu}_{rs}/\text{Nu}_{ss}$ and $f_{rs}/f_{ss}$ enhancement. The outcomes of $\eta_\rho$ have been represented as a function of Reynolds number for the various values of $D_s/D_H$ in fig. 9, and for constant values of the other blockage parameters such as $X_s/D_s = 4.04$ and $Y_s/D_s = 4.04$. It has been seen that the $\eta_\rho$ increase with increase in the $D_s/D_H$ and attains a highest value matching to a $D_s/D_H$ value of 0.195 in the range of the parameters studied.

Conclusions

A SAP roughened with sphere blockage was experimentally analysis for variation in sphere of diameter. The following conclusions are drawn.

- Attached a sphere type blockage in the inner side of heated plate results in considerable enhancement in heat transfer of fluid-flow SAP, the enhancement is a strong function of diameter of spherical blockage.
- An increase in heat transfer while decreases in pressure drop with increase in Reynolds number values is observed.
- The highest values of $\text{Nu}_{rs}$ and $\text{Nu}_{rs}/\text{Nu}_{ss}$ are observed for a spherical diameter blockage SAP with a $D_s/D_H = 0.195$.
- The maximum values of $f_{rs}$ and $f_{rs}/f_{ss}$ are observed for a spherical diameter blockage SAP with a $D_s/D_H = 0.217$.
- The superior value of overall thermal performance parameter is 2.83 corresponding to $D_s/D_H$ of 0.195.

Nomenclature

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>$A_p$</td>
<td>heated plate surface area, [m$^2$]</td>
</tr>
<tr>
<td>$A_o$</td>
<td>orifice area, [m$^2$]</td>
</tr>
<tr>
<td>$C_{do}$</td>
<td>coefficient of discharge</td>
</tr>
<tr>
<td>$c_p$</td>
<td>specific heat of air, [Jkg$^{-1}$K$^{-1}$]</td>
</tr>
<tr>
<td>$D_H$</td>
<td>hydraulic diameter of channel, [m]</td>
</tr>
<tr>
<td>$D_s$</td>
<td>diameter of sphere, [m]</td>
</tr>
<tr>
<td>$D_s/D_H$</td>
<td>relative sphere diameter</td>
</tr>
<tr>
<td>$f_{rs}$</td>
<td>friction factor of roughened obstacle</td>
</tr>
<tr>
<td>$f_{ss}$</td>
<td>friction factor without obstacle</td>
</tr>
<tr>
<td>$h_t$</td>
<td>convective heat transfer coefficient, [Wm$^{-2}$K$^{-1}$]</td>
</tr>
<tr>
<td>$H_p$</td>
<td>height of passage, [m]</td>
</tr>
<tr>
<td>$K_a$</td>
<td>thermal conductivity of air, [Wm$^{-1}$K$^{-1}$]</td>
</tr>
<tr>
<td>$L_t$</td>
<td>length of test section, [m]</td>
</tr>
<tr>
<td>$L_p$</td>
<td>length of passage, [m]</td>
</tr>
<tr>
<td>$m_a$</td>
<td>mass-flow rate of air, [kgs$^{-1}$]</td>
</tr>
<tr>
<td>$\text{Nu}_{rs}$</td>
<td>Nusselt number of obstacle surface</td>
</tr>
<tr>
<td>$\text{Nu}_{ss}$</td>
<td>Nusselt number of surface without obstacle</td>
</tr>
<tr>
<td>$\Delta p_a$</td>
<td>pressure drop across test section, [Pa]</td>
</tr>
<tr>
<td>$\Delta p_o$</td>
<td>pressure drop across orifice plate, [Pa]</td>
</tr>
<tr>
<td>$Q_u$</td>
<td>useful energy gain, [W]</td>
</tr>
<tr>
<td>$Re$</td>
<td>Reynolds number of fluid</td>
</tr>
<tr>
<td>$T_a$</td>
<td>average temperature of air, [K]</td>
</tr>
<tr>
<td>$T_i$</td>
<td>inlet temperature of air, [K]</td>
</tr>
<tr>
<td>$T_o$</td>
<td>outlet temperature of air, [K]</td>
</tr>
<tr>
<td>$T_p$</td>
<td>plate temperature of air, [K]</td>
</tr>
<tr>
<td>$V'$</td>
<td>velocity of air, [ms$^{-1}$]</td>
</tr>
<tr>
<td>$W_P/H_P$</td>
<td>passage aspect ratio</td>
</tr>
<tr>
<td>$W_P$</td>
<td>width of passage, [m]</td>
</tr>
<tr>
<td>$X_s$</td>
<td>stream wise spacing, [m]</td>
</tr>
</tbody>
</table>
\[ X_{s}/D_{h} \] – relative stream wise spacing
\[ Y_{s} \] – span wise spacing, [m]
\[ Y_{s}/D_{h} \] – relative span wise spacing

\[ \beta_{p} \] – ratio of orifice meter to pipe diameter, dimensionless
\[ \rho_{a} \] – air density, [kgm\(^{-3}\)]
\[ \nu_{a} \] – kinematic viscosity of air, [m\(^{2}\)s\(^{-1}\)]
\[ \eta_{n} \] – thermal hydraulic performance

\[ \beta_{o} \] – open area ratio, [%]

References