Transactions of the TSME (2018) Vol. 6, No. 1, 29-39
Journal of Research and Applications in Mechanical Engineering

ISSN 2229-2152 print
DOI: 10.14456/jrame.2018.4

Research Article

PERFORMANCE IMPROVEMENT OF SOLAR AIR HEATER WITH V-BAFFLES ON ABSORBER PLATE

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ABSTRACT:
The article deals with an experimental study on heat transfer and flow friction characteristics in a solar air heater duct roughened artificially with V-shaped baffles. The absorber plate is mounted with V-baffle vortex generators to improve the performance of the solar thermal system for energy saving. The experiment in the test duct having the aspect ratio (AR) of 10 is conducted for Reynolds number (Re) based on the hydraulic duct diameter ranging from 5300 to 22,600. In the current work, V-baffles are placed on the absorber with three relative baffle heights (R_B=b/H = 0.1, 0.2 and 0.3) and pitches (R_P= P/H = 0.5, 1.0 and 1.5) at a single attack angle (β) of 60°. The experimental results reveal that the use of V-baffle vortex generators yields the considerable increase in heat transfer over the smooth duct around 2.32–4.3 times while the friction loss increases around 4.08–36.9 times. The heat transfer and the friction loss tend to rise for increasing R_B but show the reversing trend for increasing R_P. The maximum thermal performance for the V-baffle vortex generators around 1.57 is seen at R_B = 0.2 and R_P = 1.0.

Keywords: *V-baffle, Solar air heater, Artificial roughness, Heat transfer*

1. INTRODUCTION

In today's world, energy resources have a trend to decline and the demand for energy is increasing every year due to population and economic growth. For decades, fossil fuels (oil, natural gas, coal, etc.) have been used so much and caused more harm to the ecosystem in terms of water and air pollution, global warming and damage to public health. Also, the utilization of fossil fuels having a decreasing trend will be depleted. Thus, renewable energy like solar energy, wind energy, etc., is regarded as clean and alternative energy source to meet the energy demands and a key agent in industrial development. Solar energy considered to be one of clean and sustainable energy sources has been harnessed by humans since ancient times for a range of ever-evolving technologies. Solar air heater as an effective device to harness solar energy has been employed for heating processes such as crops drying, space heating, seasoning of timber, etc. In general, the heat transfer coefficient between the typical smooth absorber plate (see Fig. 1) and air in the heater is somewhat poor, resulting in lower efficiency of the system. The conventional solar air heater can be generally improved by means of various enhancement techniques with several types of artificially roughened surfaces. This is because the artificially roughened surface in the form of repeated ribs/baffles [1-4] placed on the absorber is viewed as an effective technique to augment the heat transfer in a solar air heater duct.

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Several researchers have widely studied the effect of geometrical parameters of ribs/baffles on heat transfer and pressure loss in a rectangular duct with one or two roughened surfaces. Yadav and Bhagoria [5] investigated numerically the heat transfer behaviors for a square sectioned transverse rib mounted on the absorber of a solar air heater and reported that the ribbed absorber plate with relative pitch, P/b = 10.71 and relative height, b/D = 0.042 offers the best thermo-hydraulic performance. Yadav and Bhagoria [6] again conducted the 2-D CFD based analysis of an artificially roughened solar air heater having equilateral triangular sectioned rib roughness on the absorber plate and showed that the maximum Nusselt number around 3.073 times over the smooth duct is at P/b = 7.14 and b/D = 0.042. Kumar and Kim [7] numerically investigated the effect of relative roughness widths in discrete multi V-ribs arranged with staggered array on thermal performance of a solar air heater and found that the width ratio of 6.0 gives the highest thermal performance. An experimental investigation on thermal behaviors for wedge-ribs pointing upstream and downstream, triangular and rectangular ribs attached on the two opposite duct walls of a rectangular duct was carried out by Promvonge and Thianpong [8] who indicated that the maximum thermal performance is for the staggered triangular ribs. Also, Thianpong et al. [9] examined thermal characteristics in a rectangular duct roughened by isosceles triangular ribs placed on the two opposite walls and revealed that the staggered rib at P/H = 1.0 and b/H = 0.1 yields the highest thermal performance. Kumar et al. [10] investigated the heat transfer and friction loss in a solar air heater roughened with multiple V-shaped ribs with gaps and pointed that the V-rib at g/b = 1.0, b/D = 0.043, P/b = 8, W/w = 6 and β = 60° gives the highest Nusselt number and friction factor. Skullong [11] studied thermal and flow resistance characteristics in a solar air heater duct fitted with inclined-rib vortex generators placed on the absorber plate and found that the heat transfer rate is around 3.7–4.7 times above the flat/smooth duct while the friction factor increases around 24.4–70.7 times. Sriromreun et al. [12] experimentally and numerically studied fluid flow and heat transfer behaviors in a channel with Z-shaped baffles and showed that the in-phase Z-baffle with larger b/H provides higher heat transfer and friction loss than that with smaller one. The heat transfer improvement in a solar air heater with multiple V-baffles was examined numerically and experimentally by Tamna et al. [13]. Gawande et al. [14] investigated experimentally and numerically the convection heat transfer in a solar air heater duct with reverse L-shaped ribs and indicated that the maximum Nusselt number is about 2.827 times over the smooth duct. Extensive literature reviews over hundred references on rib roughness were reported by Chamoli et al. [15] and Gawande et al. [16].

Most of previous investigations on turbulent flow have considered thermal characteristics for various rib shapes, height and pitch ratios for solid or thick rib, multiple V- or Z-shaped baffles. In fact, the use of thin ribs (called baffle) provides the thermal performance higher than that of the thick ribs, confirmed by Promvonge et al. [17] for numerical work and Skullong et al. [18] for experimental work. Therefore, the study on V-baffles mounted on the absorber plate has rarely been reported. In the present work, the experiment of turbulent air flow over the 60° V-baffles mounted repeatedly on the absorber of a solar air heater is carried out with a view to exploring the heat transfer and flow resistance characteristics. With the measured data obtained, correlations for Nusselt number (Nu) and friction factor (f) have also been determined.

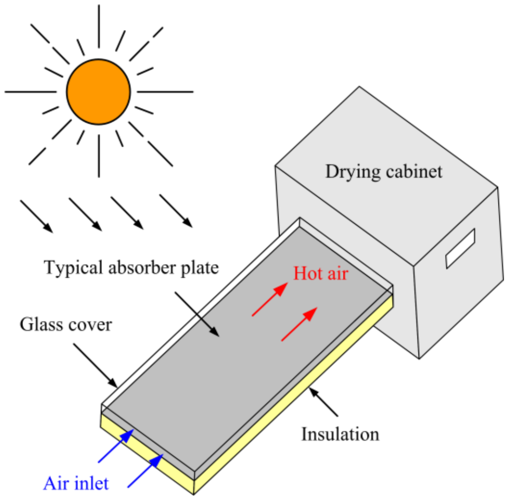


Fig. 1. Schematic diagram of conventional absorber plate of solar air heater.

2. EXPERIMENTATION

The rectangular duct in a solar air heater having dimensions of 3000 mm in total length (L_t), 300 mm in width (W) and 30 mm in height (H) for the smooth or flat-plate duct as well as roughened collectors has been designed and fabricated. The schematic diagram of the experimental system is shown in Fig. 2. It consists of a calm section, a test section, an exit section, a flow meter, a high-pressure blower, a temperature and electrical power input, a volumetric flow meter and devices to measure the pressure. The lengths (L) of the test, calm, and exit sections were 800, 2000, and 200 mm, respectively. To provide a constant surface heat-flux (q'' = 600 W/m^2), an electrical heater sheet was placed on the absorber plate. The solar air heater duct was connected to the discharge side of a 1.9 kW high pressure blower. The air entered the duct at a constant temperature (room temperature setting at 27 °C) and was measured by a volumetric flow-meter. The rate of airflow through the duct was measured by means of a calibrated orifice plate mounted on the flow pipeline and connected to an inclined manometer, while the flow was controlled by an inverter. Twenty-four thermocouples (type-T) were employed for measuring the surface temperature along the absorber plate at certain locations while two RTD-typed temperature sensors were applied for measuring the temperature of outlet air leaving the mixing section. One RTD-typed temperature sensor was used for measuring the temperature of inlet air. The accuracy of such thermocouples was within ±0.1°C. More design details of experimental system can be found in Ref. [19]. A data acquisition system connected to a laptop was employed for recording the output signals of all thermocouples. The pressure drop across the test duct was measured from two static pressure taps located at the entry and the exit on the top duct wall by a digital differential pressure device. Uncertainties of the measured results were calculated based upon the analysis of errors in measurements [20]. The uncertainty for experimental heat balance was within ±5% and the maximum error of the airflow rate was within ±3%.

In the experiment, V-baffles were arranged by its V-tip pointing upstream with the attack angle (or half V-apex angle), β = 60° as depicted in Fig. 3. The V-baffles made of a 1.0-mm thick aluminum strip were 3, 6 and 9 mm high (b), equivalent to three baffle- to duct-height ratios (called "relative roughness height"), $R_B = b/H$ = 0.1, 0.2 and 0.3, and were placed repeatedly on the absorber plate with three baffle-pitch to duct-height ratios (called "relative roughness pitch"), $R_P = P/H$ = 0.5, 1.0 and 1.5, respectively. For comparison, results of the smooth or flat plate duct operating under similar flow conditions were also reported.

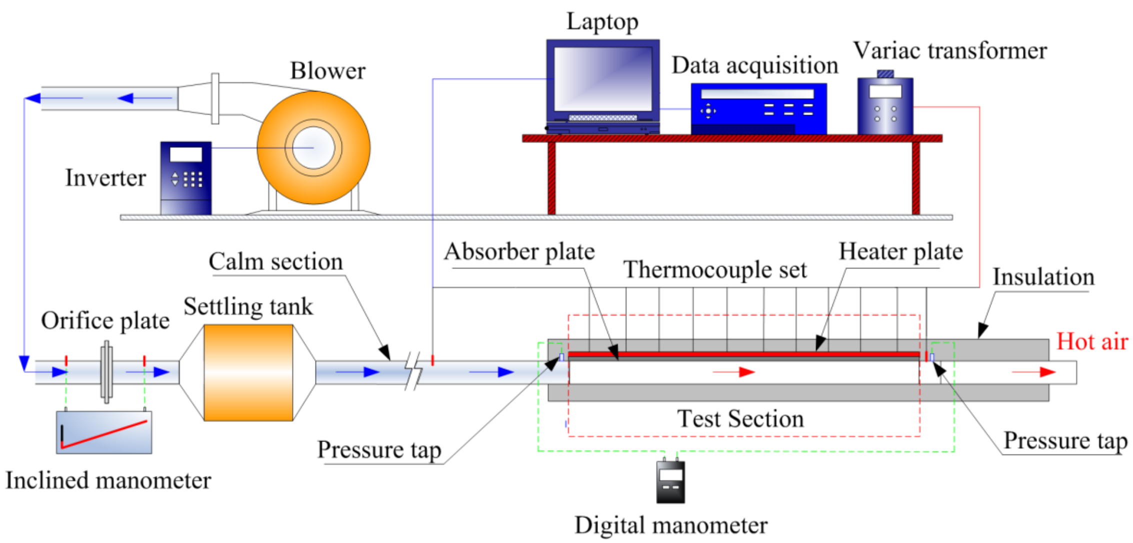


Fig. 2. Schematic sketch of experimental system.

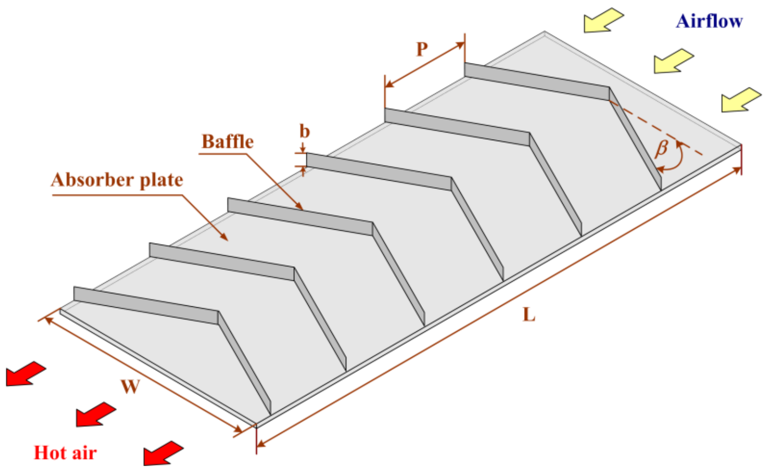


Fig. 3. Schematic layout of absorber plate with V-baffle.

3. Theoretical Analysis

The present experiment was conducted to investigate the heat transfer rate (Nusselt number, Nu), pressure loss (friction factor, *f*), flow parameter (Reynolds number, Re) and thermal performance (thermal enhancement factor, TEF) in a solar air heater duct roughened artificially with V-baffles.

In the current experiment, when air enters the test-duct at steady flow condition, the heat transfer is considered to be equal to the convection heat loss in the duct and expressed as:

$$Q_{\mathrm{a}} = Q_{\mathrm{conv}} \tag{1}$$

where

$$Q_{\mathrm{a}} = \dot{m} C_{\mathrm{p,a}} \left(T_{\mathrm{o}} - T_{\mathrm{i}}\right) \tag{2}$$

The convection heat transfer in the duct is calculated by

$$Q_{\mathrm{conv}} = hA\left(\tilde{T}_{\mathrm{w}} - T_{\mathrm{b}}\right) \tag{3}$$

in which

$$T_{\mathrm{b}} = (T_{\mathrm{o}} + T_{\mathrm{i}})/2 \tag{4}$$

and

$$\tilde{T}_{\mathrm{w}} = \sum T_{\mathrm{w}} / 24 \tag{5}$$

where T_w is the local wall temperature along the duct. The average wall temperature, $\tilde{T}_{\mathrm{w}}$, is evaluated from 24 points of the wall temperatures lined equally along the duct. The average heat transfer coefficient (*h*) and average Nusselt number (Nu) are written as

$$h = \dot{m} C_{\mathrm{p,a}} \left(T_{\mathrm{o}} - T_{\mathrm{i}}\right) / A\left(\tilde{T}_{\mathrm{w}} - T_{\mathrm{b}}\right) \tag{6}$$

The average Nu is calculated from the heat transfer coefficient (*h*) using the following relations:

$$\mathrm{Nu} = hD/k \tag{7}$$

where, D is the duct hydraulic diameter, calculated as:

$$D = 4\times W\times H/2\times(W+H) \tag{8}$$

The Nusselt number ratio ($\mathrm{Nu_R}$) is defined as Nu of the V-baffled duct divided by that of flat-plate duct.

$$\mathrm{Nu_R} = \frac{\mathrm{Nu}}{\mathrm{Nu_f}} = \frac{\text{Nu of duct with baffle}}{\text{Nu of flat plate duct}} \tag{9}$$

The flow in terms of Reynolds numbers (Re) based on hydraulic diameter (D) of the duct is calculated from

$$\mathrm{Re} = UD/\nu \tag{10}$$

The friction factor (f) computed by pressure drop (ΔP) in the test duct is written as

$$f = \frac{2}{(L/D)}\frac{\Delta P}{\rho U^2} \tag{11}$$

in which U is the mean air velocity in duct.
The friction factor ratio (f_R) is defined as f of the V-baffled duct divided by that of flat-plate duct.

$$f_\mathrm{R} = \frac{f}{f_\mathrm{f}} = \frac{f\text{ of duct with baffle}}{f\text{ of flat plate duct}} \tag{12}$$

All of thermo-physical properties of air are determined at overall bulk air temperature (T_b) of Eq. (4).
Thermal enhancement factor (TEF) is given by

$$\mathrm{TEF} = \left(\frac{\mathrm{Nu}}{\mathrm{Nu_f}}\right)\left(\frac{f}{f_\mathrm{f}}\right)^{-1/3} \tag{13}$$

where Nu_f and Nu are for the smooth and the V-baffled ducts, respectively.

4. Results and Discussion

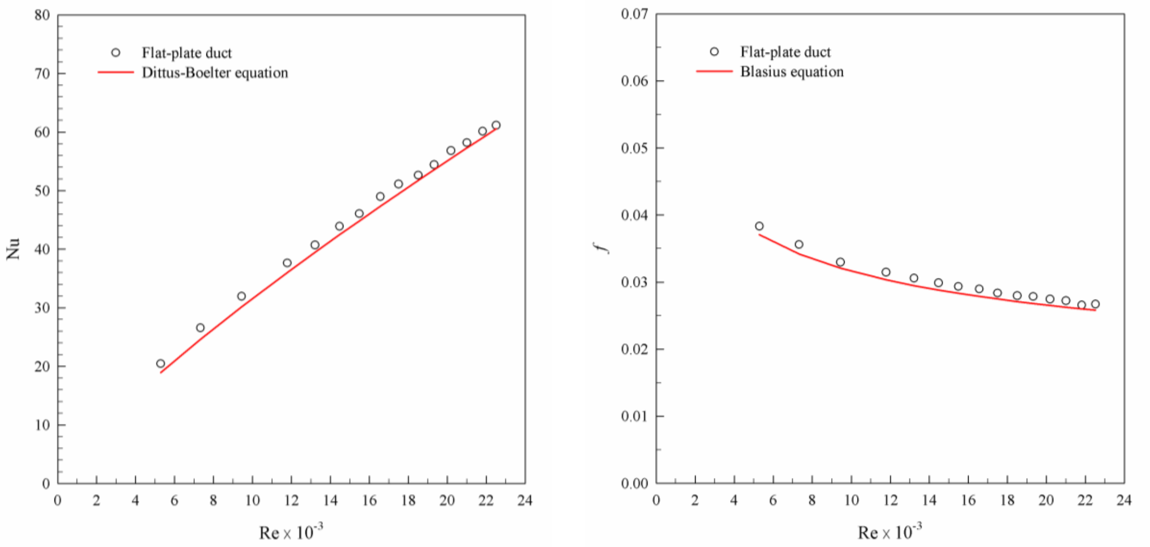
4.1 Validation test

A comparison of experimental $\mathrm{Nu_0}$ and f_0 data from the flat/smooth duct in the present work and those from the standard equations [21] of Dittus-Boelter and Blasius, as given in Eqs. (14) and (15), is depicted in Fig. 4(a) and 4(b). In the figure, the measured data are in good agreement with equations' data. The average deviations of the measured are within 4.6% and 6.3% for Nu and f, respectively.
Dittus-Boelter equation:

$$\mathrm{Nu} = 0.023\mathrm{Re}^{0.8}\mathrm{Pr}^{0.4} \tag{14}$$

Blasius equation:

$$f = 0.316\,\mathrm{Re}^{-0.25} \tag{15}$$

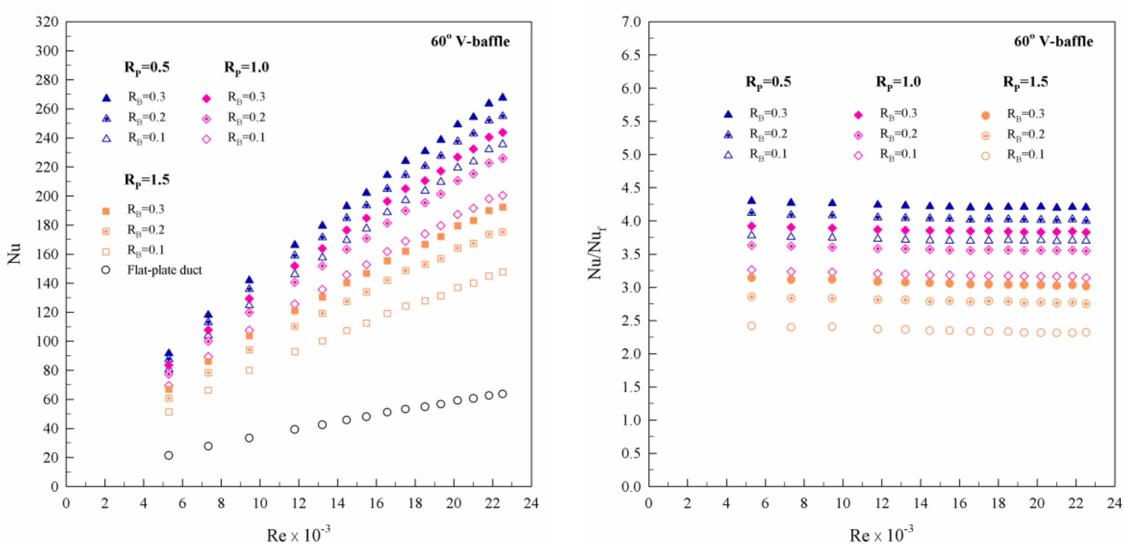


(a) (b)

Fig. 4. Verifications of experimental with published equations of (a) Nu and (b) *f* for flat-plate duct.

4.2 Heat transfer characteristics

In Fig. 5(a), it is clear that heat transfer in the form of Nu increases with increasing Re. The duct with V-baffles on the absorber provides considerably higher Nu than the flat-plate duct alone. Nu tends to increase with increasing R_B. The increment of Nu is due to the reduction of thermal boundary layer thickness on the absorber with V-baffles that help create the vortex-flows for prolonging the residence time of flow mixing. The heat transfer of the absorber plate with V-baffle is seen to be higher than that of the flat-plate duct around 65–77%, and the maximum heat transfer is at R_B = 0.3 and R_P = 0.5.



(a) (b)

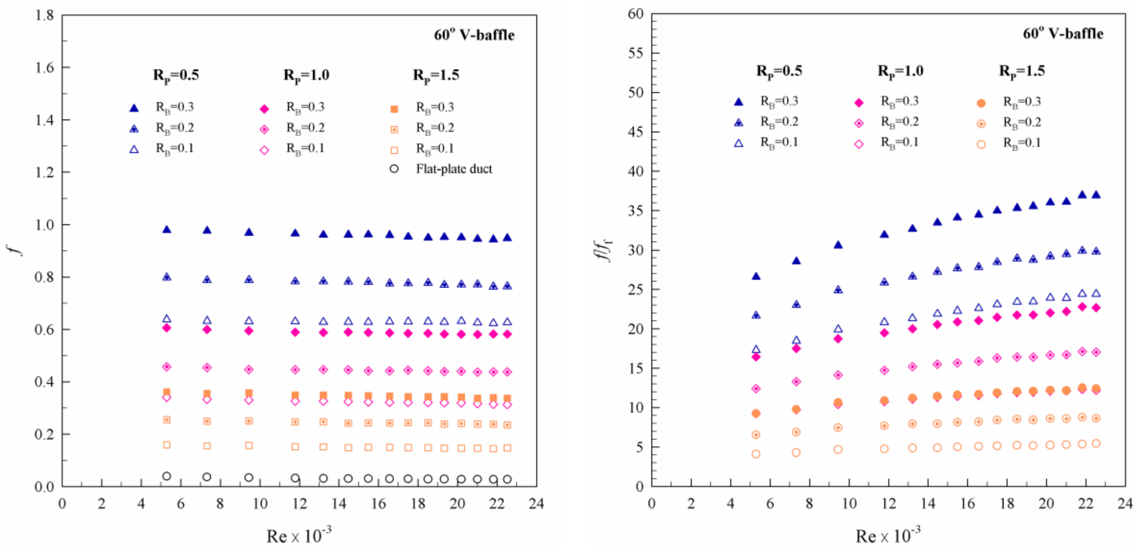
Fig. 5. Variations of (a) Nu and (b) Nu/Nu_f with Re for V-baffles.

Figure 5(b) depicts the ratio of augmented Nu of the V-baffled absorber to Nu of the flat-plate duct, (Nu/Nu_f) with Re at various R_B and R_P. In the figure, Nu/Nu_f tends to be nearly uniform with the rise of Re for all cases. The smaller R_P leads to the increase in Nu/Nu_f while the smaller R_B provides the reversing trend. The V-baffle at R_B = 0.3 and R_P = 0.5 gives much higher Nu/Nu_f than the others for augmented heat transfer. This is caused by using the V-baffle at R_B = 0.3 that can give stronger vortex strength and disrupt the boundary layer at the absorber surface leading to higher rate of heat transfer between the absorber surface and air. At R_B = 0.3, 0.2 and 0.1, the increases in Nu/Nu_f for R_P = 0.5, 1.0 and 1.5 are, respectively, about 4.2–4.3, 4–4.12 and 3.7–3.78; 3.83–3.92, 3.55–3.63 and 3.14–3.26; and 3.02–3.15, 2.75–2.86 and 2.32–2.42 times, depending on Re. This implies that the V-baffle with larger R_B and smaller R_P yields considerably higher heat transfer. However, at this condition, it comes together with extremely higher friction factor, as can be seen in the next section.

4.3 Friction characteristics

The *f* for the V-baffled absorber plate is analyzed and compared with the flat-plate duct. Figure 6(a) presents the influence of R_B and R_P on *f*. As expected, the V-baffle vortex generators can cause a significant friction loss in comparison with the flat-plate duct. *f* tends to increase with rising R_B but shows the downtrend with increasing R_P. This is because of higher blockage of flow especially at larger R_B apart from larger frontal and surface areas while the strength of longitudinal vortex is amplified as the relative height of baffles is increased, which leads to larger form drag. Thus, the flow blockage from the appearance of V-baffle including higher level of turbulence degree in the central core region is a vital factor to cause an extreme friction loss, especially for large R_B and small R_P. Compared with the flat-plate duct at the Re range investigated, *f* of the V-baffles is increased from 4.08–36.93 times.

Figure 6(b) displays the variation of f/f_f with Re for various R_B and R_P values. In the figure, it is noted that f/f_f shows the uptrend with increasing R_B while exhibits the reversing trend with the increment of R_P. The average f/f_f values for R_B = 0.3, 0.2 and 0.1 are, respectively, about 33.61, 27.3 and 22.1; 20.63, 15.57 and 11.33; and 11.46, 8.05 and 4.96 at R_P = 0.5, 1.0 and 1.5. This means that at larger R_B, the application of small R_P should be avoided, for example, the case of R_B = 0.3 and R_P = 0.5 should not be offered in the present work.



(a) (b)

Fig. 6. Variations of (a) *f* and (b) f/f_f with Re for V-baffles.

4.4 Thermal enhancement factor

In real applications, the potential assessment of the V-baffles is essential and displayed in terms of thermal enhancement factor (TEF) that is plotted versus Re for different R_B and R_P as exhibited in Fig. 7. This result is directly relevant to the trade-off between the enhanced Nu and the increased friction loss penalty. As shown in the figure, TEF shows the tendency to decrease with the rise of Re for all baffles and its maximum around 1.57 is at R_B = 0.2 and R_P = 1.0. TEF values at R_P = 1.0 are, respectively, found to be in the range of 1.36–1.56, 1.38–1.57 and 1.35–1.54 for R_B = 0.1, 0.2 and 0.3. Therefore, the best choice of this V-baffle roughness is at R_B = 0.2 and R_P = 1.0 to achieve the superior thermal performance.

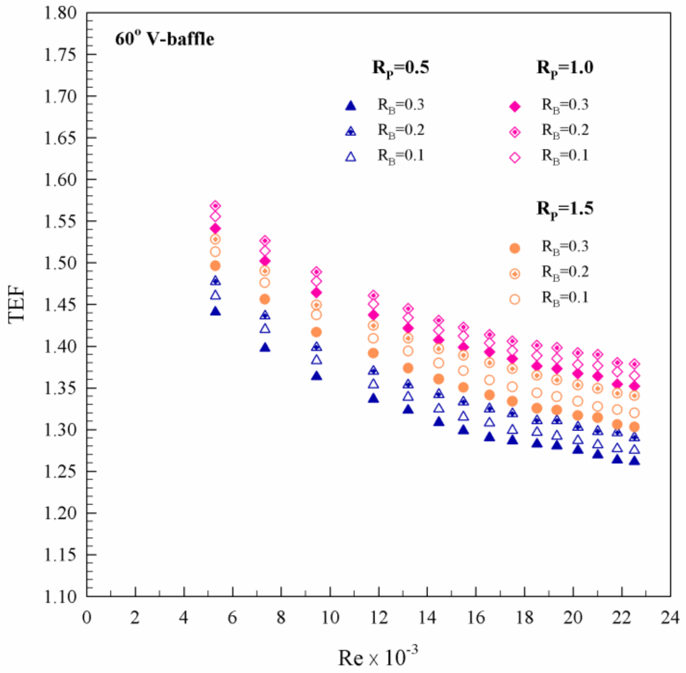


Fig. 7. Variation of TEF with Re for various V-baffles.

5. EMPIRICAL CORRELATIONS

In the current work, the values of Nu and *f* are developed in the form of empirical correlations for the 60° V-baffles and correlated as given in equations (16) and (17). The Nu and *f* correlations are displayed as the function of Re, R_P, R_B and Prandtl number (Pr) whereas *f* is free from Pr. More details for the establishment of empirical correlations of Nu and *f* can be found in Ref. [10]. The correlations of the present work are valid for R_P in the range of 0.5–1.5, R_B ranging from 0.1 to 0.3 and Re from 5300 to 22,600.

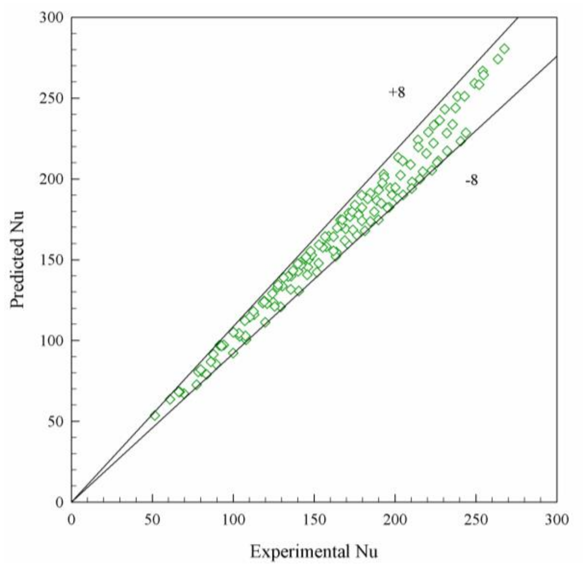
Nusselt number for 60° V-baffles;

$$\mathrm{Nu} = 0.208\mathrm{Re}^{0.733}\mathrm{Pr}^{0.4}\mathrm{R}_\mathrm{B}^{0.177}\mathrm{R}_\mathrm{P}^{-0.33} \tag{16}$$

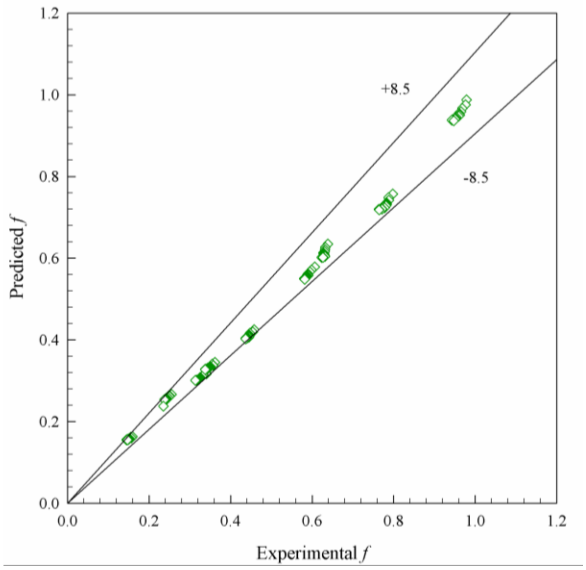
Friction factor for 60° V-baffles;

$$f = 1.419\mathrm{Re}^{-0.037}\mathrm{R}_\mathrm{B}^{0.554}\mathrm{R}_\mathrm{P}^{-1.066} \tag{17}$$

To validate the correlation reliability, Nu and *f* values obtained by the prediction from the correlations were plotted against the measured data as depicted in Fig. 8(a) and (b), respectively. It is observed in the figure that most measured data are within ±8% and ±8.5% of the predicted Nu and *f*, respectively.



(a)



(b)

Fig. 8. Comparison of experimental with predicted values for (a) Nu and (b) *f*.

6. CONCLUSIONS

Effects of artificial roughness in the form of V-shaped baffles placed on the absorber plate on friction and heat transfer behaviors in a solar air heater duct have been investigated experimentally. The effects of Re, R_B and R_P on thermal characteristics are introduced in the current study. Key findings from the study are as follows:

- The use of V-baffles yields a significant effect to flow behaviors in the duct leading to enhancing the heat transfer as well as friction loss.
- The maximum Nu on average for the V-baffle is found to be 4.23 times above the flat-plate duct while the highest *f* is around 33.61 times at $R_P = 0.5$ and $R_B = 0.3$.
- At thermal performance comparison, the highest TEF of about 1.57 is at $R_B = 0.2$, $R_P = 1.0$ and Re = 5300. Thus, the V-baffle artificial roughness is a promising method for performance improvement of a solar thermal system.
- Comparison with previous correlation shows an excellent agreement.

NOMENCLATURE

A	[m^2]	absorber plate area
b	[m]	height of baffle/rib
R_B	[=b/H]	relative height of baffle
R_P	[=P/H]	relative pitch of baffle
$C_{p,a}$	[J/kg K]	specific heat of air
D	[m]	hydraulic diameter of duct
f	[-]	friction factor of roughened duct
f_f	[-]	friction factor of flat-plate duct
g	[m]	gap distance
H	[m]	duct height
k	[W/m K]	thermal conductivity of air
$\dot{m}$	[kg/s]	mass flow rate of air
L	[m]	length of test section
L_t	[m]	total length of test section
Nu	[-]	Nusselt number of roughened duct
Nu_f	[-]	Nusselt number of flat-plate duct
P	[m]	pitch length between baffles
Q	[W]	heat transfer rate

Pr	[-]	Prandtl number
R_P	[$=P/H$]	relative longitudinal pitch
Re	[-]	Reynolds number
T_i	[ºC]	inlet air temperature
T_o	[ºC]	outlet air temperature
T_b	[ºC]	bulk mean air temperature
TEF	[-]	thermal enhancement factor
W	[m]	duct width

Greek symbols

μ	[kg/m s]	dynamic viscosity
β	[degree]	inclination or attack angle

Subscripts

a		air
w		wall

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