Development and Improvement in Heat Transfer and Friction Behaviour of Solar Air Heater

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Abstract: As cheap fossil fuels become scarce, global temperatures will rise. In the future, renewable energy will be both the primary and secondary solution. It is possible to lessen the demand for primary energy resources while saving money by using solar energy for passive cooling and heating in a number of applications. A manufactured roughness on the underside of the absorber plate can improve the thermal performance of a solar air heater. This procedure is both efficient and economical. Several experimental investigations using various types of roughness components have been carried out to increase heat transmission from the absorber plate to the air moving via solar air heaters. These findings show that multiple V-rib roughness has a significant impact on heat transfer coefficient and friction factor in a solar water heater duct with artificially roughened surfaces. For all pitch values and 3 mm roughness height, the best Nusselt number enhancement is at Reynolds number 6000. With 3-mm and 3.5-mm roughness, the highest increases in friction factor occur at all pitch values from 10-60 mm. A link between the Nusselt number and friction factor has been developed using these experimental results.

Keywords: Solar Energy, Solar Air Heater, Nusselt Number, Fiction Factor, Reynolds Number, Pitch, Roughness Height.

1. Introduction

Energy drives economic growth. Demand for energy rises with population growth, industrialization, and increased residential use. However, energy consumption in emerging countries is currently based on fossil fuels (petroleum, coal, and natural gas). Worldwide, new solar energy technologies (solar thermal and photovoltaic) are being developed. Solar energy can be used for passive cooling and heating, sparing primary energy resources. The solar air heater is the simplest and most extensively utilized heat exchanger in solar energy applications. A solar air heater is a flat plate collector that consists of an absorber plate and a parallel plate below it that forms an air channel. Due to the limited heat transfer properties of air, a solar air heater's thermal efficiency is lower than a solar water heater. The second reason for poor performance is the formation of a laminar sublayer near the absorber plate surface. So it's vital to disrupt this layer and make the flow turbulent. Solar air heaters must improve their thermal efficiency to be commercially viable. Extended surfaces like fins or V corrugations on the absorber plate aid in two ways: enhancing turbulence and boosting heat transfer area. This improves heat transfer. So, repeating ribs are used to interrupt the laminar sublayer, improving heat transfer and efficiency [1, 2, and 3].

Roughness geometry was discussed in detail by Saini and Singhal [4], who also discussed the effects of various factors. It is the development of two flow separation regions on either side of a rib that has the most significant impact on the flow patterns produced by its presence. In this study, it was discovered that vortices are responsible for turbulence and that as a result, there is an increase in both heat transfer and friction losses. Muluwork et al. [5] investigated the thermal performance of staggered discrete V apex up and down rib matching transverse friction staggered discrete ribs and made comparisons between the two types. When compared to v-up and transverse discrete ribs, it is discovered that V-down discrete ribs perform significantly better. Prasad and Saini [6] concentrated their attention on the influence of relative roughness height (e/Dh) and relative roughness pitch (p/e) on heat transmission and friction factor, respectively. Using a square channel with two opposite rib roughened walls that were in line with each other, Han and Zhang [7] studied the influence of the angle of orientation on the local heat transfer distribution and pressure drop in the channel. For e/D=0.0625 and p/e=10, the Reynolds number was altered from 15x103 to 90x103, with the range being 15x103 to 90x103. The rib configuration includes 90°, 60°, and 45° parallel ribs, 60° and 45°crossed ribs, 60°and 45°V-shaped ribs, and 60-degree and 45-degree ribs. In the experiments, it was discovered that the V-shaped rib outperforms the angled ribs due to the production of two vortex cells in the V-shaped rib as opposed to only one vortex cell in the inclined ribs. The ribs' top was chamfered having chamfer angles of 5°,12°,15°,18°,22° and 30°, while the relative roughness pitch (P/e) and relative roughness height (e/Dh) of the ribs were kept constant having values of 10 and 0.03 respectively. The ebb and flow of the Reynolds number of the duct ranged between approximately 3000 and 21,000, with the highest value reported to be best suitable for solar air heaters by Layek et al. [8]. The flow through a rectangular duct with multi-gap V-down ribs combined with staggered ribs was studied experimentally for heat transfer, friction factor, and thermo-hydraulic performance. The rectangular duct utilized had an aspect ratio of 12 and the Reynolds number ranged from 4000 to 12,000. The angle of attack (a) was varied from 40 to 80 degrees, rib pitch-to-height (P/e) ratio was varied from 4 to 14, rib height-to-hydraulic diameter (e/Dh) ratio was varied from 0.026 to 0.057, and rib pitch-to-height (P/e) ratio was varied from 4 to 14. The highest Nusselt numbers were obtained for P/e values of 6 and 12 and a drop in Nusselt numbers for e/Dh values above 0.044. Singh et al. [9] found that the largest improvement in Nusselt number and thermohydraulic performance was 3.34 and 2.45 times. Karmere and Tikekar [10] investigated heat transmission and fluid movement in a solar air heater roughened with metal grit ribs. This study used a collector plate with a 60-degree inclination to the airflow and metal ribs of circular, square, and triangle cross-sections. The grit rib pieces were staggered on the surface to form a grid. According to Momin et al [11] and Hans et al [12], bending long-angled ribs into a v-shape helps generate two leading ends (where heat transfer is greatest) and a single trailing end (where heat transfer is low). A rectangular duct with V-shaped ribs as artificial roughness is studied numerically in this work. Many tests have been conducted to determine the heat transfer and fluid flow characteristics of rectangular ducts roughened with multiple V-ribs. Finally, evaluated relationships between the Nusselt number and friction factor in terms of roughness and flow characteristics using experimental data.

2. Experimental Work

The rectangular test section has been fabricated to determine the convective heat transfer coefficient for the flow of air in the duct. The value of heat transfer coefficient 'h' has been found at different flow rates for different configurations. The setup consists of a centrifugal blower through an orifice plate along with an inclined U-tube manometer. Experiments were conducted to observe enhancement ratio in terms of Nusselt number (Nu/Nus) and enhancement ratio in terms of friction factor (f/fs) for Reynold Number range 3000 to 30000 at various values of roughness pitch (p) at roughness height (e).

3. Results and Discussions

3.1 Heat Transfer

The Nusselt number enhancement ratio (Nu / Nus) is used to assess heat transfer improvements in ducts with surface roughness compared to smooth ones. Tables 1 to 3 reveal variations in Nu/Nus with Reynolds number, roughness heights, and pitches. Table 1 shows that the most substantial heat transfer enhancement is achieved with a 10 mm pitch and Reynolds numbers between 10,000 and 12,000, particularly for a 1 mm roughness height. Increasing the pitch reduces this enhancement, but the optimal improvement consistently occurs within the 10,000 to 12,000 Reynolds number range at a 1 mm roughness height.

Enhancement ratio = Nu / Nus ----------------(1)

Table 1: Variation in enhancement ratio (Nu/Nus) with Reynold Number for various values of roughness pitch (p) at roughness height (e) of 1mm

	Nu/Nus for various values of roughness pitch (p)					
Re	**p=10**	**p=15**	**p=20**	**p=25**	**p=40**	**p=60**
6000	1.96	1.76	1.70	1.62	1.62	1.50
10000	2.-17	1.93	1.88	1.69	1.63	1.56
12000	2.17	1.99	1.93	1.72	1.64	1.65

Table 2: Variation in enhancement ratio (Nu/Nus) with Reynold Number for various values of roughness pitch (p) at roughness height (e) of 3 mm

	Nu/Nus for various values of roughness pitch (p)					
Re	**p=10**	**p=15**	**p=20**	**p=25**	**p=40**	**p=60**
6000	2.32	2.61	2.27	2.18	2.13	1.44
10000	2.28	2.53	2.14	2.06	2.00	1.38
12000	2.22	2.48	2.04	1.98	1.92	1.32

Table 3: Variation in enhancement ratio (Nu/Nus) with Reynold Number for various values of roughness pitch (p) at roughness height (e) of 3.5 mm

	Nu/Nus for various values of roughness pitch (p)					
Re	**p=10**	**p=15**	**p=20**	**p=25**	**p=40**	**p=60**
6000	2.38	2.71	2.30	2.31	2.14	1.50
10000	2.33	2.61	2.14	2.11	1.98	1.41
12000	2.26	2.55	2.06	2.04	1.93	1.36

Table 2 demonstrates that, for a 3 mm roughness height, the best heat transfer enhancement occurs with a 15 mm pitch at a Reynolds number of 6,000, though higher pitches negatively affect heat transfer. Importantly, data highlights a significant performance decline with greater roughness pitch, even dropping below the Nusselt number of a smooth duct at a 60 mm roughness height. Thus, introducing roughness can adversely affect the duct's thermal performance. Additionally, the optimal enhancement consistently occurs at Reynolds number 6,000 for all pitch values at a 3 mm roughness height in Tables 2 and 3. Overall, higher pitches have an adverse impact on heat transfer, especially at increased roughness heights, suggesting the introduction of roughness may harm the duct's thermal performance.

3.2 Friction Factor

The use of artificial roughness results in the enhancement of heat transfer rate accompanied by an increase in friction factor. The increase in friction factor is also represented by the enhancement ratio which is defined as the ratio of friction factor of roughened duct to that of smooth duct.

Enhancement ratio=f/fs…………………………(2)

Tables 4 to 6 detail the variation of the Enhancement ratio (f/fs) concerning Reynolds number and roughness geometry parameters. In Table 4, maximum enhancements in friction factor are observed at a 15 mm pitch and Reynolds numbers between 10,000 to 12,000, with the highest enhancement occurring at a 20 mm pitch for a 1 mm roughness height. Tables 5 and 6 reveal that the maximum friction factor enhancements are consistent across all pitch values from 10 to 60 mm at Reynolds numbers between 10,000 to 12,000 for roughness heights of 3 and 3.5 mm. Higher roughness heights lead to a significant increase in pumping power requirements.

Table 4: Variation in enhancement ratio (f/fs) with Reynold Number for various values of roughness pitch (p) at roughness height (e) of 1mm

Re	f/fs for various values of roughness pitch (p)					
	p=10	p=15	p=20	p=25	p=40	p=60
6000	1.95	2.28	2.24	2.52	1.99	1.81
10000	2.49	2.73	2.69	2.72	2.37	2.02
12000	2.59	2.88	2.86	2.34	2.37	1.97

Table 5: Variation in enhancement ratio (f/fs) with Reynold Number for various values of roughness pitch (p) at roughness height (e) of 3mm

Re	f/fs for various values of roughness pitch (p)					
	p=10	p=15	p=20	p=25	p=40	p=60
6000	3.61	4.03	3.79	3.49	3.32	3.75
10000	4.02	4.11	3.95	3.85	3.62	4.01
12000	4.00	4.08	3.98	3.79	3.61	4.36

Additionally, it's noted that higher roughness heights and pitches may result in only marginal increases in heat transfer rate or, in some cases, even a decrease compared to a smooth duct. Equation 3 defines the thermo-hydraulic performance.

$$\eta = \frac{Nu/Nus}{f/fs} \text{---------------} - (3)$$

Table 6: Variation in enhancement ratio (f/fs) with Reynold Number for various values of roughness pitch (p) at roughness height (e) of 3.5mm

Re	f/fs for various values of roughness pitch (p)					
	p=10	p=15	p=20	p=25	p=40	p=60
6000	3.61	4.03	3.79	3.49	3.32	3.75
10000	4.02	4.11	3.95	3.85	3.62	4.01
12000	4.00	4.08	3.98	3.79	3.61	4.36

4. Conclusions

Maximum heat transfer enhancement occurs at a 10 mm pitch and a Reynolds number between 10,000 to 12,000 for a roughness height of 1 mm. This enhancement decreases with an increase in pitch. However, at a roughness height of 1 mm, the optimal enhancement occurs at Reynolds number 10,000 to 12,000 for all pitch values. The highest heat transfer enhancement is at a 15 mm pitch and a Reynolds number of 6,000 for roughness heights of 3 and 3.5 mm. However, increasing pitch adversely affects heat transfer. Notably, high roughness pitch can deteriorate duct performance, and at a roughness height of 60 mm, the Nusselt number is lower than in a smooth duct. Additionally, maximum friction factor enhancement occurs at a 20 mm pitch and a Reynolds number between 10,000 to 12,000 for a roughness height of 1 mm, and the same conditions produce maximum friction factor enhancement. At higher roughness heights, pumping power requirements increase significantly, and in some cases, heat transfer rates may decrease, particularly at very high roughness pitches.

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