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# **Performance Analysis of Solar Air Heater Using V Shaped Ribs on Absorber Plate**

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**ABSTRACT**: The heat transmission and thermal efficiency of a solar air heater with a V-shaped rib were studied statistically. Using a solar air heater with artificial roughness in the form of 300 incline ribs, thermal efficiency has been estimated. On the upper side of the absorber surface, 15 mm and 20 mm pitches of inclined ribs were used to construct. The rib pitch values of 20 mm, and heat transfer data were examined for the comparison. Various geometrical features of the solar duct were numerically calculated with an eye toward increasing the maximum heat transfer rate. The model with 15 mm pitch has a maximum efficiency of 43.8 percent, while the model with 20 mm pitch has a maximum efficiency of 40.7 percent. When compared to findings for a smooth absorber surface, the efficiency of both models increases by 4.26 percent. Model-I reaches its maximum glazing glass cover and absorber surface temperatures at sunlight intensities of 1180-1250 W/m2 at the 12:30-1:30 a.m. time slot. A comparison of the exergy analysis demonstrates that the exergy value increases as the time of day increases in solar intensity. The CFD analysis is performed to verify the experiment results and to examine flow phenomena above the absorber surface.

KEYWORDS: Solar Intensity, Thermal efficiency, Relative pitch roughness, exergy analysis, solar air heater.

#### I. INTRODUCTION

In order to accomplish anything, you are going to require a steady supply of energy. Over 80 percent of the world's overall energy requirements are met by the burning of fossil fuels. On the other hand, based on the rate at which fossil fuels are being consumed at the moment, it is anticipated that their supply would run out within a few years[1]–[3]. As a result, it is essential for us to investigate and develop alternatives to fossil fuels in order to reduce the amount of reliance we have on these fuels. In regions with colder temperatures, it is standard practice to heat buildings with air heaters that are also used for drying crops and ageing lumber. Solar energy, being a source of energy that is continually replenished, may be used in air warmers. The air that is drawn into the solar air heater is warmed as it passes over the absorber surface. As a consequence of the low heat transfer coefficient between surface of the absorber and the air, the performance of a solar-powered air heater is subpar[2], [4]. This is as a result of the formation of a laminar sub layer in between them. As a consequence of this, researchers have played with a variety of artificial roughness geometries in the hope of shattering this sub layer and improving the performance of the material. Some of these experiments have produced the expected results, and those results are currently being put to use in order to satisfy the existing need for energy[5]–[9].

Yadav et. al. [10] It was observed that a solar air heater with roughened absorber surface wires was around 8 to 10 percent more efficient than a solar air heater with smooth wires when compared to another solar air heater with smooth wires. An investigation showed that roughened solar air heaters are not effective for thermo hydraulic efficiency when the Reynolds number is more than 29000. Wang et. al.[11] In order to boost the effectiveness of the solar air collector, S-shaped ribs with gaps were placed on the absorber surface. Under controlled testing settings, it was possible to achieve an increase in thermal efficiency of between 13 and 48 percent. Anil et. al.[12] developed a relationship between the Nusselt number and the absorber surface, as well as generated gaps on the ribs of numerous V-shaped artificial roughness on the absorber surface. It was discovered that the value of the Nusselt number increased by 6.74 times while utilising this configuration, but the friction factor also increased by 6.37 times in comparison to when it was used with a smooth surface. Patel et. al.[13] The influence of NACA 0040 profile ribs arranged in reverse sequence as a roughness element in a solar air heater was tested, and the results of the experiment showed that the thermo hydraulic performance was 2.53 at a Reynolds number of 6000, which is an increased performance. Poongavanam et. al.[14] experiments showed that using absorber surfaces with shot-blasted V-corrugations improved heat transfer on solar air heaters, and the Nusselt number obtained was higher than that of conventional solar air heaters while the friction factor was slightly higher. Anup kumar et. al.[15] Statistical analysis of a solar air heater's absorber surface was used for numerical examination. Relative pitch roughness of 8, rib orientation angle of 60o, and twist ratio of 3 were

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used to provide the greatest gain in thermal and exergetic efficiency over a smooth surface. Sahu et. al.[16] This study compared the exergy efficiency of a solar air heater with one with an arc-shaped wire roughened absorber surface with that of a similar solar air heater with an absorber surface that was smooth, and it found the rough absorber surface to be 56 percent more efficient. Singh et al[17] studied SAH utilising two different roughness surfaces, one with many broken transverse ribs and the other with square wave shaped ribs, for both CFD and experimental purposes. For square-shaped rib roughened absorber surfaces, the maximum thermal hydraulic efficiency was more than double that of a plane absorber surface, while the pumping power required was three times that of the plane absorber surface. There was a threefold increase in pumping power and absorber surface efficiency while treating SAH patients with numerous fractured ribs. For this investigation, researchers used ribs with a pitch to height ratio of 10 and a height to diameter ratio of 0.043. Gawande et al.[18] A reverse L-shaped ribbed absorber surface was used to test the SAH's performance. These measurements were conducted at a relative roughness pitch of seven, a Reynolds number of 3800 to 18000, and at a constant relative roughness height of 0.042. The thermo-hydraulic performance of 1.90 was reached at a high heat transfer rate.

#### II. EXPERIMENTAL SET UP

The experimental setup comprises of a flat surface solar air heater. The diagrammatic representation of the flat surface solar heater is illustrated in figure 1. The rectangular part is made using aluminum which is integrated for absorption. The interior dimension of the duct utilized in agreement is 1000 mm x 600 mm x 50 mm where 1.0 m is the length of the rectangular section of the duct, 0.6 m is the breath and 0.15 m is height. The entry and exit length of the experimental setup are of equal dimension. Hence length at the entry as well as exit is .4 m. The exit part of the duct is linked to a .053 m diameter pipe. The air heater is constructed in such a manner that it can offer maximum heat flux up to 1250 W/m2.

A large rectangular piece makes up the solar heater's main body. It features a glass cover on the upper side, and an absorber surface on the lower side that absorbs solar light. At the setup's intake, a 2 kW centrifugal blower and a 3 phases 220 Volt motor draw in ambient air to power the system's setup. The solar heater's air flow may be adjusted using gate valves located at the heater's inlet and exit. Figure 1 shows the schematic diagram of a rectangular solar air heater.V-shaped 30 degree inclined ribs are used to create artificial roughness on absorber surfaces. The rib pitch on one surface is 15 millimeters, while the pitch on the opposite surface is 20 millimeters. Figure 2 shows the roughness geometry of both surfaces. The glass and absorber surfaces are separated by an air gap.

Every reading session began with a check to make sure that the setup arrangement was working properly and that no leaks were present at any of the joints. In order to achieve steady state between readings, the set up was left on for at least an hour between each set. Air temperature input and exit sections, solar radiation, absorber surface temperature, glass surface temperature on average, and pressure drop throughout a section are all recorded for each experiment in the series. The turbulence in the rectangular duct has increased to the point that it can be observed. At the rectangular duct portion's panel, no slippage is expected.





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Fig. 1.Schematic diagram of rectangular solar air heater

#### **III. DATA REDUCTION**

#### **3.1 EFFICIENCY ANALYSIS**

Nusselt number and friction factor for V-shaped 30 degree inclined ribs are employed in the current experimentation. As a consequence, the solar air heater's design parameters are optimised. According to the method outlined below, the thermal efficiency has been calculated.

Using the relationship shown below, we may estimate the net gain in thermal energy for practical purposes,

$$Q_{u1} = [I(\tau \alpha) - U_L(T_{pm} - T_a)]A_p$$
 3.1.1

The following equation anticipates the total heat loss U<sub>L</sub>.

$$U_{\rm L} = U_{\rm t} + U_{\rm b} + U_{\rm s}$$

 $U_{t_s}$ ,  $U_b$  and  $U_s$  are the role of Solar air heater performance factor and their values are intended by utilizing following relations.

$$U_{b} = \frac{k_{b}}{X_{b}}$$
 3.1.3

$$U_{s} = \frac{(L+W)t_{e}K_{i}}{LWt_{i}}$$
3.1.4

The empirical equation used for determining by using the top loss coefficient U<sub>t</sub>, is

$$U_{t}^{-1} = \left[ \frac{\sigma(T_{pm}^{2} - T_{g}^{2})(T_{pm} - T_{g})}{\left(\frac{1}{\epsilon_{p}} + \frac{1}{\epsilon_{g}} - 1\right)} + \left(\frac{K_{a}N_{u}}{L_{g}}\right) \right]^{-1} + \left[\sigma \in_{g} (T_{g}^{2} - T_{a}^{2})(T_{g} - T_{a}) + h_{a}\right]^{-1} + \frac{t_{g}}{K_{g}}$$
3.1.5

$$T_{g} = \frac{F_{1}T_{pm} + C T_{a}}{1 + F_{1}}$$
3.1.6

F<sub>1</sub>

$${}^{F_{1}} = \frac{\left(12x10^{-8}(T_{a} + 0.2T_{pm})^{3} + h_{a}\right)^{-1} + 0.3t_{g}}{\left(6x10^{-8}(\epsilon_{p} + 0.028)(T_{pm} + 0.5T_{a})^{3} + 0.6L_{g}^{-0.2}\{(T_{pm} - T_{a})\cos\beta\}^{0.25}\right)^{-1}}$$

$$C = \left[\frac{\left(\frac{T_{s}}{T_{a}}\right) + \left(\frac{h_{a}}{3.5}\right)}{1 + \left(\frac{h_{a}}{3.5}\right)}\right]$$
3.1.7

To calculate the Nusselt number the below equation is given by Singh[21]

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$$Nu = 0.00236 \times \text{Re}^{0.9} \times \left(\frac{p}{e}\right)^{3.5} \times \left(\frac{\alpha}{60}\right)^{-0.023} \times \left(\frac{d}{w}\right)^{-0.043} \times \left(\frac{g}{e}\right)^{-0.014} \times \left(\frac{e}{D_{h}}\right)^{0.27}$$

$$\times \exp\left(-0.084 \left(\ln\left(\frac{p}{e}\right)\right)^{2}\right) \times \exp\left(-0.72 \left(\ln\left(\frac{\alpha}{60}\right)\right)^{2}\right)$$

$$\times \exp\left(-0.05 \left(\ln\left(\frac{d}{w}\right)\right)^{2}\right) \times \exp\left(-0.15 \left(\ln\left(\frac{g}{e}\right)\right)^{2}\right)$$
3.1.9

For calculating mass flow rate, an equation like this is used,

$$\dot{m} = \frac{Q_{u1}}{C_{pa}\Delta T}$$
 3.1.10

Utilization below equation yields the flow Reynolds number,

$$R_{e} = \frac{\rho V D_{H}}{\mu_{a}} \qquad 3.1.11$$

Nusselt number-based convective heat transfer coefficients [22],

$$h = \frac{N_u k_a}{D_H}$$
3.1.12

The surface efficiency factor  $F^\prime$  is calculated by the equation,

$$F' = {n \over h + U_L}$$
 3.1.13

Then the valuable heat gain of solar air heater is given by the equation,

$$Q_{u2} = F_r [I(\tau \alpha) - U_L (T_{pm} - T_a)] A_p$$
 3.1.14

$$F_{r} = \frac{\dot{m}C_{pair}}{A_{p}U_{L}} \left[ 1 - \exp\left\{\frac{-F'U_{L}A_{p}}{\dot{m}C_{pair}}\right\} \right]$$
3.1.15

The thermal efficiency of the arrangement is calculated by the relation,

$$\eta_{th} = \frac{\text{Useful heat gain}}{\text{Solar intensity radiation falling on the glass surface}}$$

$$\eta_{\rm th} = \frac{Q_{\rm u}}{A_{\rm p}I_{\rm T}}$$
 3.1. 16

#### 3.2 EXERGY ANALYSIS

Exergy efficiency of the solar air heater is calculated by the equation,

$$\eta_{ex} = 1 - \frac{\sum \dot{E}_{loss}}{\sum \dot{E}_{inlet}}$$
 3.2. 1

The net exergy energy rate supplied by the solar air heater is specified by,

$$\sum \dot{E}_{inlet} = IA \left[ 1 - \frac{4}{3} \left( \frac{T_a}{T_{sun}} \right) + \frac{1}{3} \left( \frac{T_a}{T_{sun}} \right)^4 \right]$$
 3.2.2

The summation of exergy destruction losses are stated as

$$\sum \dot{E}_{loss} = (\dot{E}_{opt-loss}) + (\dot{E}_{q-loss}) + (\dot{E}_{fr-loss}) + (\dot{E}_{sun,Tp-loss}) + (\dot{E}_{Ta,Tp-loss})$$
 3.2. 3

Figure 10 shows how the exterior optical loss of a heater may be calculated using the bellow affiliation,

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$$\dot{E}_{opt-loss} = IA(1-\tau\alpha) \left[ 1 - \frac{4}{3} \left( \frac{T_a}{T_{sun}} \right) + \frac{1}{3} \left( \frac{T_a}{T_{sun}} \right)^4 \right]$$
**3.2.4**

A solar air heater's exergy loss as a result of heat transmission to the atmosphere,

$$\dot{E}_{q-loss} = U_L A (T_p - T_a) \left[ 1 - \left(\frac{T_a}{T_p}\right) \right]$$
3.2.5

Exergy losses in solar air heaters are calculated using frictional exergy losses,

$$\dot{E}_{fr-loss} = \frac{T_m m \Delta p}{\rho f T_a}$$
3.2.6

The exergy destruction finite differential between fluid and absorber surface is given by,

$$\dot{E}_{Ta,Tp-loss} = IA\eta_{\rm th} T_{\rm a} \left[ \left( \frac{1}{T_{\rm a}} \right) - \left( \frac{1}{T_{\rm p}} \right) \right]$$
3.2.7

Solar intensity radiation absorbed by the absorber surface results in exergy destruction losses.

$$\dot{E}_{sun,Tp-loss} = IA\tau\alpha \left[ 1 - \frac{4}{3} \left( \frac{T_a}{T_{sun}} \right) + \frac{1}{3} \left( \frac{T_a}{T_{sun}} \right)^4 - \left( \frac{T_a}{T_p} \right) \right]$$
3.2.8

# **3.3 CFD SIMULATION**

The flow conditions and temperature differences between the input and exit of the solar air heater have been clearly depicted in experimental data, which has improved the overall efficiency of the device. Computational fluid dynamics simulations in ANSYS FLUENT 14.5 and SOLIDWORKS 18.2 were used to frame the model and better understand the air flow phenomena between the v-shaped ribs roughened absorber surface and the air entering the test section. A simpler rectangular duct with a v-shaped 300 inclined roughness geometry in its bottom has been devised. The model's p/e ratio is set at 2.5. A 1 m/sec inlet velocity is used in this analysis. Figure 2 depicts a combined model of the duct with bottom-side roughness. According to Jaipur's location, For 18 June 2020 at 13 p.m. in Jaipur, a constant solar intensity value of 885W/m<sup>2</sup> is available.Fluid flow is assumed to be stable, incompressible, and turbulent during the CFD setup. There is no variation in the fluid properties across the channel. Atmospheric inlet temperature of 300K is used for the active fluid.

Figure 3 shows the meshing process after a model has been imported into ANSYSFLUENT. A horizontal rectangular duct is used as a computational space. A constant mesh size of 89766 nodes and 184727 elements has been chosen. Discrete quadrilateral grids are used in the computation domain. Relevance is set at 40, and fine sizing is chosen as an option.



Fig. 2. Solar air heater CFD model



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Fig. 3. Meshing of solar air heater

In order to get a more accurate result. Series combination is used to achieve double precision. The absorber's potential materials The surface is made of aluminium, and it has the same properties as the other materials in the library. Energy equation, viscous flow with realizable k- with enhanced wall fin, and radiation equation with direct transfer DTRM solar loading are all considered in the CFD solver. Glass has a density of 2500 kg/m3, a specific heat capacity of 750 J/kg-k, and a thermal conductivity of 0.7 W/m-k. As a result of the way the computational domain is constructed, air enters the duct at a specific velocity that is dependent on Reynolds number. Therefore, the velocity boundary at the duct's inlet is defined as 1m/s. Assumption: The outlet is a pressure outlet, with a gauze pressure value of zero. In contrast to the duct's other sides, the duct's upper side is heated by a constant heat flux of 800-1250w/m<sup>2</sup>.

#### **IV. EXPERIMENTAL RESULTS**

As seen in Figure 4, the effectiveness of a solar air heater can vary depending on the time of the day. The experimental results show that both surfaces, model I and model II, reach their peak efficiency and performance at 01:00 PM. Model-I represents a 15 mm pitch surface, whereas Model-II represents a 20 mm pitch surface with a V-shaped 300 inclined rib roughness on the absorber surfaces. 39.51 percent, 41.81 percent, 42.05 percent, 42.05 percent, 42.5 percent, 43.1 percent, 43.8 percent, 42.68 percent, and 41.39 percent are the thermal efficiency numbers for Model–I and Model–II, respectively. In the morning, the amount of solar intensity radiation is modest, but as the day passes, it rises. Peak values of 1250 W/m<sup>2</sup>.



Fig. 4. Comparison between Thermal Efficiency of SAH with Model-I and Model-II vs. Time of Day

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For Model-I and II, Figure 5 shows the time-of-day variation of glazing temperature and absorber surface temperature. The temperature of the absorber surface and the glazing is calculated using the equation provided in the calculation section. The inlet and outlet air temperatures, as well as the radiation intensity, affect the glazing and absorber surface temperature values. The graph shows that Model-I has a higher temperature than Model-II. As a result of the close spacing of the artificial rib roughness in Model-I, the absorber surface and air are able to transfer heat more quickly and with a larger surface area. Temperature differences between inlet and outlet sections (T0-Ti) change as the day progresses, as shown in Figure 6. The graph demonstrates that low radiation intensity results in a low (T0-Ti) value. Based on the time of day and radiation intensity, (T0-ti) has a different value each time.



Fig. 5. Variation of Glazing, Mean Absorber Surface Temperature vs. Time of day of SAH with Model-I and Model-II



Fig. 6. Variation of Temperature Difference  $(T_0-T_i)$  Vs. Time of day

As shown in Figure 7, time of day and radiation intensity are related. Radiation intensity is at its highest between the hours of 12:30 and 13 p.m. Radiation intensity can reach 1250 W/m2 at its maximum. At 9 a.m., the lowest radiation intensity is 800 W/m2. The graph depicts an increase in radiation intensity until 1:00 p.m., at which point it begins to decline. As can be seen, both the amount of solar radiation and the surrounding temperature are subject to change over time.



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Fig. 7 . Variation in Radiation Intensity and Time of Day

Solar air heater models I and II are shown side-by-by-side in Figure 8 to illustrate how their energy efficiency varies with the time of day. My observations show that efficiency in both situations diminishes with time, and for a brief period of time the efficiency value is dependent on the flow of air above the absorber surface due to heat transfer. Since the absorber surface may be made artificially rougher, it improves heat transfer rates and hence improves heat removal rates. The intake and exit temperatures have an effect on exergy efficiency. Both models' exergy efficiency rises till 13:00. After the 13:00 hour, it begins to decline because of lower outlet temperatures and other exergy losses.



Fig. 8. Comparison of SAH Exergy Efficiency (%) with Model-I and Model-II vs. Time of Day

For model-I and model-II, a separate type of exergy loss vs. time of day is shown in Figures 9 and 10. The temperature differential between the absorber surface and the sun is the most critical element in the loss of exergy. Other sorts of

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losses are given the greatest values by this method. During periods of low solar intensity, its value is lower, while during higher solar intensity, it reaches its peak. The solar air heater's inlet and output sections have a temperature differential that affects all exergy destruction losses. The turbulence intensity and pressure drop of the flowing air causes friction exergy destruction. As the convection heat transfer coefficient increases, the temperature differential between the flowing air and the absorber surface decreases. As a result, the highest solar intensity exergy efficiency value is high.



Fig. 9. Exergy Losses Variation vs. Time of Day for Model-I



Fig. 10. Exergy Losses Variation vs. Time of Day for Model-II

Comparison of the exergy efficiency provided to the solar heater for models I and II can be seen in Figure 11. Temperatures at the intake and outlet affect exergy efficiency.  $1250 \text{ W/m}^2$  of radiation is the greatest value of radiation intensity during the hours of 12:30 and 13 hours. The greater exergy energy is given to the solar air heater at this solar intensity.



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Fig. 11. Exergy Energy Supplied by SAH with Models I and II vs. Time of Day

# CFD SIMULATION RESULT

Solar air heater temperature distribution is shown in Figure 12 as a changing contour map. There is a healthy differential in temperature between the inlet and exit temperatures of 328K.



Fig. 12. Total Temperature Distribution within Solar Air Heater as a Contour Plot

The variability of the streamline flow pattern of air velocity within the solar air heater is depicted in Figure 13. It displays the air flow turbulence on the roughness surface of the absorber. This enhances the heat transfer rate from the roughness surface to the flowing fluid when there is a lot of turbulence. As a result, vortices form behind the ribs, resulting in a high temperature during departure. The pressure difference between the input and output air is also 0.635 Pa. This shows that the pumping power required for this design is also lower. As a result, the solar air heater's losses are reduced.



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Fig. 13. Inside the Solar Air Heater, there is a smooth flow of air.

# V. CONCLUSION

Although raising the rib pitch from 20 mm improves heat transmission, it also raises friction losses, which lowers overall efficiency, according to the current experimental data. For the V-shaped rib roughened solar heater, the following conclusions may be drawn from the results of both experimental and numerical analysis:

- For Model-I, 43.8 percent and for Model-II, 40.7 percent, the highest efficiency is found in the time span of 13 hours, i.e. from 9AM to 5PM.
- Temperature rises as a result of an increase in solar radiation intensity.
- Artificial rib roughness of 300 inclined V-shaped increases solar air heater efficiency to 4.26 percent on average.
- Model-glazing I's glass cover temperature and absorber surface temperature are at their highest at sunlight intensities of 1180-1250 W/m2 between the hours of 12:30-12:30.
- In both models, the exergy value rises with increasing sun intensity, according to the exergy study.
- The temperature differential between the input and output is 22 K, according to the CFD results. This adds credence to the findings of the experiments.
- For improved temperature differences between the intake and outflow, CFD simulations suggest that V-shaped roughness with rib angles of 30 degrees is most suited. Improved pumping efficiency and reduced waste.

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