Influence of Channel Depth on the Performance of Wavy Fin Absorber Solar Air Heater

Abhishek Priyam, Prabha Chand

Abstract—Channel depth is an important design parameter to be fixed in designing a solar air heater. In this paper, a mathematical model has been developed to study the influence of channel duct on the thermal performance of solar air heaters. The channel depth has been varied from 1.5 cm to 3.5 cm for the mass flow range 0.01 to 0.11 kg/s. Based on first law of thermodynamics, the channel depth of 1.5 cm shows better thermal performance for all the mass flow range. Also, better thermohydraulic performance has been found up to 0.05 kg/s, and beyond this, thermohydraulic efficiency starts decreasing. It has been seen that, with the increase in the mass flow rate, the difference between thermal and thermohydraulic efficiency increases because of the increase in pressure drop. At lower mass flow rate, 0.01 kg/s, the thermal and thermohydraulic efficiencies for respective channel depth remain the same.

Keywords—Channel depth, thermal efficiency, wavy fin, thermohydraulic efficiency.

I. INTRODUCTION

SOLAR air heaters, usually have flat absorber plate for the collection of solar energy. Due to their simple design and little maintenance, it is used for drying and space heating applications. However, due to high thermal losses from the absorber plate, they generally exhibit low thermal performance. Hence, there is a need for the development of an efficient flat plate solar air heater. In the previous studies, the concept of roughness and packed beds are commonly used to achieve enhanced heat transfer rate in convective heat transfer. Studies show that appropriate design of channel geometry can improve the performance of solar air heaters. The present study is an investigation of influence of channel depth on the performance of wavy fin absorber solar air heater. Vijaya et al. [1] reviewed 12 different solar air heaters that were fabricated and used for drying various products, namely, onions, fruits, and so on. Their review revealed that exploitation of solar air heaters is appropriate for drying purposes. Also, the quality of products dried by those solar air heaters is higher in comparison with that of dried by fossil fuels facilities.

Kumar et al. [2] reviewed heat transfer and friction characteristics of a multiplicity of artificially roughened solar air heaters. They expressed different ways that could be used to improve heat transfer and as a result performance enhancement of these solar systems. Tyagi et al. [3] compared energy and exergy efficiencies of three different solar air heater collector arrangements with and without thermal energy storage materials, using the first and second laws of thermodynamics. Their results showed that utilization of thermal energy storage materials such as paraffin wax and hytherm oil can soar both energy and exergy efficiencies of their systems simultaneously. Additionally, they mentioned that these efficiencies are slightly higher for the system used paraffin wax, compared to the one that exploited hytherm oil. Singh et al. [4] conducted a research to illuminate the exergy efficiency of a solar air heater that had discrete V-down rib roughness on its absorber plate. They perceived that not only did their system performed better from energy and exergy points of view, but also it is extremely appropriate to be used for the condition that Reynolds number is lower than 18000. However, using their system for higher Reynolds numbers will lead to more pump power consumption that is unfavorable. Bouadila et al. [5] manufactured a novel solar air heater with packed-bed latent storage energy to evaluate the nocturnal performance of their apparatus. They reported that the daily energy efficiency varied between 32% and 45%, while the daily exergy efficiency ranged from 13% to 25%. Sabzpooshani et al. [6] investigated the exergetic performance of a single pass baffled solar air heater. They realized that exergy efficiency was the superior criterion for performance evaluation. Yet another, the more intense the solar radiation is, the higher exergy efficiency will be. Bayrek et al. [7] fabricated five different solar air heaters with and without porous baffles to assess the energy and exergy efficiencies. Their experiments anticipated that using a solar air heater with a thickness of 6mm and an air mass flow rate of 0.025 kg/s had the maximum efficiencies, while the one where no baffles were inserted in had the lowest efficiencies. Furthermore, the energy and exergy efficiencies for their five various solar systems varied between 39.35%–77.57% and 21.55%–54.54%. Bahrehmand and Ameri [8] mathematically studied the energy and exergy efficiencies of two different solar air heaters. They deduced that collector that triangular fins were inserted in was more efficacious than that of rectangular fins from energy and exergy perspectives. Yadav et al. [9] analytically appraised the exergetic performance of a solar air heater provided with protrusions arranged in arc fashion on its absorber plate. They figured out that using such solar facility is appropriate for Reynolds number less than 20 000, while for Reynolds number greater than 20 000, the exergetic efficiency will decline due to considerable increase in pumping power required by fan.

Nwosu [10] through analytical calculations has attempted to determine the minimal entropy generation condition using

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exergy optimization technique in a pin finned solar air heater. Much later, Donggen Peng et al. [11] through experimental analysis have shown that the use of pin fin arrays on the absorber plate can significantly increase the thermal efficiency as compared to the smooth duct. Priyam and Chand [12] investigated on the thermal and thermohydraulic performance of wavy fin absorber solar air heater to study the influence of fin spacing. In another work by Chand and Priyam [13], thermal performance was investigated to study the aspect ratio of wavy fin solar air heater. However, their work did not consider the variation of channel depth by the introduction of wavy fins in the air path. Based on the above detailed literature reviews, the main objective of this analysis is to investigate the effect of channel depth on the thermal and thermohydraulic performance of the wavy fin absorber air heater system. A composite parameter called effective efficiency which considers the effective thermal gain after accounting for the pressure losses in the flow is used to determine the worthiness of wavy fin array as thermal performance enhancement device.

II. ANALYSIS

It is well known fact that one possible way to maximize useful solar heat gain is to employ higher mass flow rates. For the systems with certain limitations on collector length and pumping power, however this may be achieved by making the flow channel deeper (higher channel depth to collector length ratio) so that flow resistance across the air channel is relatively low. The collector under consideration consists of a wavy finned absorber plate with insulated bottom plate and a glass cover. A passage between absorber plate and bottom plate has been made through which the air is to be heated as shown in Fig. 1 and the geometrical construction is shown in Fig. 2. The heat gain by air may be calculated using the following equations:

$$Q_u = I A_c (\tau\alpha)_e - U_L A_c (T_p - T_a) = A_c \left[I(\tau\alpha)_e - U_L (T_p - T_a) \right] \quad (1)$$

$$Q_u = \dot{m} C_p (T_o - T_i) \quad (2)$$

$$q_u = F_R \left[I(\tau\alpha)_e - U_L (T_{fi} - T_a) \right] \quad (3)$$

where F_R is collector heat removal factor and given by

$$F_R = \left(\frac{\dot{m} C_p}{A_C U_L} \right) \left[1 - \exp \left(\frac{-F' A_C U_L}{\dot{m} C_p} \right) \right] \quad (4)$$

The collector efficiency F′ can be given by

$$F' = \frac{h_e}{h_e + U_L} \quad (5)$$

and the equivalent heat transfer coefficient, h_e is

$$h_e = h_{fp} + \frac{2 h_f . \Phi_f . \beta . h_{ff}}{w} + \frac{h_r . h_{fb}}{h_r + h_{fb}} \quad (6)$$

The fin efficiency, Φ_f may be computed as

$$\Phi_f = \frac{\tanh \ mh_f}{mh_f} \quad (7)$$

where,

$$m = \left[\frac{2 h_{ff}}{k_a \delta_f} \right]^{1/2} \quad (8)$$

The mean absorber plate temperature from (1) and (3) is given by

$$T_{pm} = T_i + \frac{Q_u}{A_c F_R U_L} (1 - F_R) \quad (9)$$

and, the mean fluid temperature, T_{fm} is given by

$$T_{fm} = \frac{1}{L} \int_0^L T_f dx = T_i + \frac{Q_u}{A_c F_R U_L} \left(1 - \frac{F_R}{F'} \right) \quad (10)$$
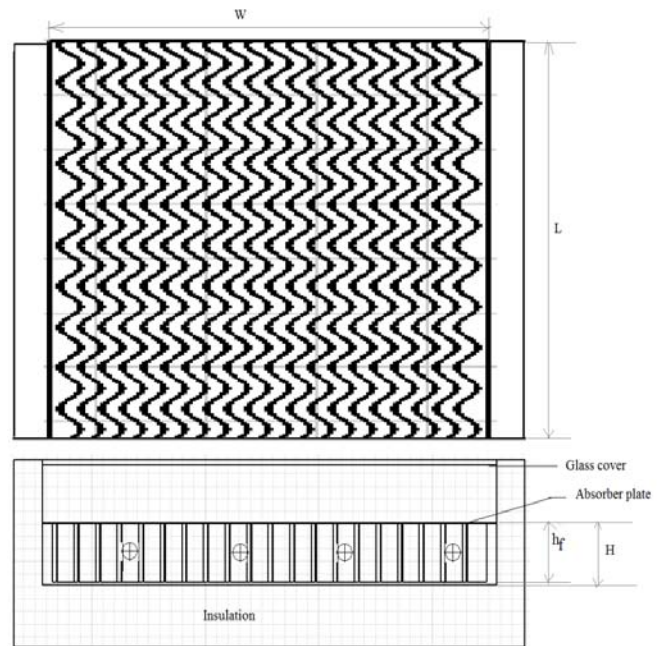

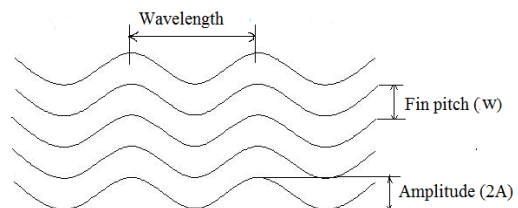Fig. 1 Schematic diagram of wavy finned solar air heater

Fig. 2 Geometrical structure of wavy fin

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First, an assumption of mean plate temperature is made from which U_L is calculated, with approximate values of F_R, F' and Q_u, a new value of mean plate temperature is obtained from (9) and used to calculate a new value of top loss coefficient, and this is repeated until the accuracy of 0.01% is achieved.

The thermal efficiency may be calculated using (3) as

$$\eta = \frac{q_u}{I} \qquad (11)$$

III. Heat Transfer and Pressure Drop

The amount of heat loss from the collector in terms of an overall loss coefficient, U_L, as

$$Q_L = U_L A_c (T_p - T_a) \qquad (12)$$

where

$$U_L = U_t + U_b + U_s \qquad (13)$$

The energy loss through the side is much smaller than the top loss coefficient and bottom loss coefficient. It can be expressed as

$$U_s = \frac{(L + W) H k_i}{L W \delta_s} \qquad (14)$$

where L= length of collector, W= width of collector, H= duct height, k_i= thermal conductivity of insulation, δ_s = thickness of side insulation.

The energy loss through the bottom of the collector is sum of conduction, convection, and radiation to the environment. The magnitude of convection and radiation term is negligibly small as compared to the conduction term. Therefore, generally bottom loss coefficient U_b, is approximately expressed as;

$$U_b = \frac{k_i}{\delta_b} \qquad (15)$$

where k_i = thermal conductivity of the insulation, δ_b = thickness of bottom insulation.

The top loss coefficient from the collector is evaluated by considering both convection and radiation from the absorber plate to ambient. An empirical relation for calculation of top loss coefficient is given by Klein [14]

$$U_t = \left[\frac{N_{gc}}{\left(\dfrac{C}{T_{pm}}\right)\left(\dfrac{T_{pm} - T_a}{N_{gc} + f'}\right)^{0.33}} + \frac{1}{h_w} \right]^{-1}$$
$$+ \left[\frac{\sigma(T_{pm}^2 + T_a^2)(T_{pm} + T_a)}{\dfrac{1}{\varepsilon_p + 0.05\,N_{gc}(1 - \varepsilon_p)} + \dfrac{2N_{gc} + f' + 1}{\varepsilon_c} - N_{gc}} \right] \qquad (16)$$

where

$$f' = (1 + 0.04\,h_w + 0.0005\,h_w^2)(1 + 0.091\,N_{gc})$$

$$C = 365.9\,(1 - 0.00883\,\theta + 0.0001298\,\theta^2)$$

$$h_w = 5.7 + 3.8 V_w$$

where h_w is the wind heat transfer co-efficient, and V_w is the wind velocity (m/s).

The radiative heat transfer coefficient h_r is calculated by [14]

$$h_r = \sigma \varepsilon_p (T_{pm}^2 + T_{sky}^2)(T_{pm} + T_{sky}) \qquad (17)$$

where

$$T_{sky} = 0.0552 T_a^{1.5}$$

and for calculating the Nusselt number, the correlation of the colburn factor (j) is recommended by Ismail et al. [15] and may be used for wavy fin.

$$j = 2.348\,Re^{0.786}\,(h_f/w)^{0.312}\,(2A/w)^{0.192}\,(\lambda/2A)^{0.432}$$
$$(\text{for } 100 < Re < 800) \qquad (18)$$

and

$$j = 0.242\,Re^{0.375}\,(h_f/w)^{0.235}\,(2A/w)^{0.288}\,(\lambda/2A)^{0.553}$$
$$(\text{for } 1000 < Re < 15000) \qquad (19)$$

where,

$$j = \frac{Nu}{Re\,Pr^{1/3}} \qquad (20)$$

and

$$Re = \frac{G D_h}{\mu} \qquad (21)$$

and the expression for fanning friction factor developed by Ismail et al. [15] and may be used for wavy fin

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$$f = 9.827 \, \mathrm{Re}^{0.705} (h_f / w)^{0.322} (2A / w)^{0.394} (\lambda / 2A)^{0.603}$$
$$\text{(for } 100 < \mathrm{Re} < 800) \qquad (22)$$

and

$$f = 10.628 \, \mathrm{Re}^{0.359} (h_f / w)^{0.264} (2A / w)^{0.848} (\lambda / 2A)^{1.931}$$
$$\text{(for } 1000 < \mathrm{Re} < 15000) \qquad (23)$$

Total net energy gain, Q_{net}, of the wavy fin solar air heater may be expressed as:

$$Q_{net} = Q_u - \frac{P_m}{C} \qquad (24)$$

where 'P_m' is the mechanical energy consumption to overcome the friction and it is computed by the relation

$$P_m = \frac{\dot{m} \Delta P}{\rho} \qquad (25)$$

'C' is the conversion factor representing conversion from thermal energy to compression energy of the fan / blower imparted to air [12] and given as

$$C = \eta_{Th} \eta_{fr} \eta_m \eta_f \qquad (26)$$

The thermohydraulic performance parameter (effective efficiency) can now be written using above equations as;

$$\eta_{eff} = \frac{G C_p [(T_{fo} - T_{fi}) - C_m]}{I \mathrm{x} A_c} \qquad (27)$$

where C_m is the equivalent temperature drop due to friction and is given by

$$C_m = \frac{\Delta P}{C.\rho.Cp} \qquad (28)$$

IV. RESULTS AND DISCUSSION

In order to obtain results numerically, codes were developed in C++ using the parameters listed in Table I.

Fig. 3 shows the variation of total loss coefficient for different channel depth and mass flow rates. It can be seen from plot that the loss coefficient increases with the increase in channel depth and for a specific mass flow rate. Also, for the entire range of channel depth, the total loss coefficient decreases with the increase in mass flow rate. This is because the increase in channel depth leads to increase in mean plate temperature which in turn increases the total loss coefficient. Also, by increasing the mass flow rate, mean temperature of absorber plate decreases, and the total loss coefficient decreases. The maximum value of loss coefficient is obtained as 6.52 W/m²-K for the minimum mass flow rate of 0.01 kg/s and channel depth of 3.5 cm.

TABLE I
VALUES OF PARAMETERS USED IN ANALYSIS

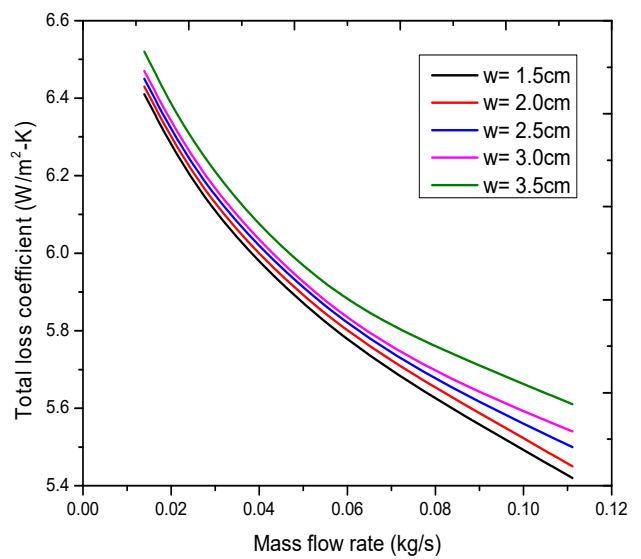| Parameter	Range/value	Parameter	Range/value
A_c	1.2 m²	k_a	0.029 W/m K
L	1.2 m	δ_{ins}	5 cm
W	1 m	2A	15 mm
I	900 W/m²	λ	70 mm
H	1.5 - 3.5 cm	L'	1.551 m
N_{gc}	1	δ_{gc}	4 mm
h_f	1.2 - 3.2 cm	T_a	29 °C
δ_f	1 mm	T_{fi}	30 °C
k_{ins}	0.1 W/m K	ε_p	0.95
$(\tau\alpha)_e$	0.85	ε_b	0.95
w	1 cm	ε_c	0.88
$\dot{m}$	0.01- 0.11 kg/s	V_w	2.5 m/s

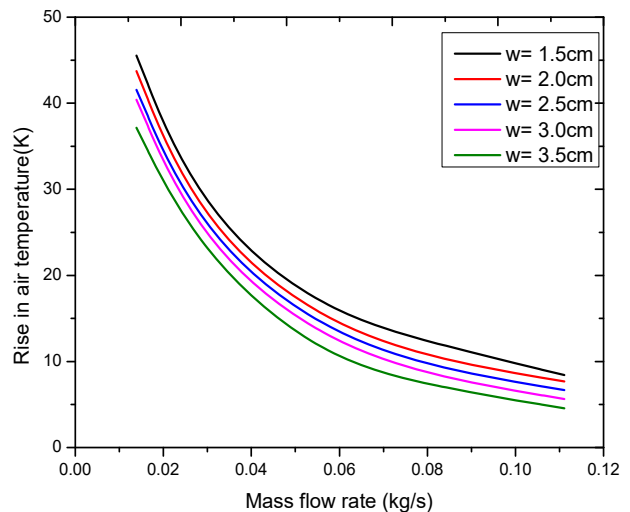Fig. 3 Variation of total loss coefficient for different channel depth

Fig. 4 Variation of air temperature rise for different channel depth

Fig. 4 shows the variation of air temperature rise for various channel depth and entire range of mass flow rate. For all channel depth, air temperature rise decreases with increase in

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mass flow rate. The higher mass flow rate show a lesser steeper fall because of high rate of energy collection that results in higher heat transfer rates and relatively lesser thermal losses. Also, increase in channel depth decreases the air temperature rise. This is because at higher channel depth the rate of heat collection is high that leads to higher heat transfer rates and lesser thermal losses. A maximum air temperature rise has been achieved as 45.6 K for the channel depth of 1.5 cm.

Fig. 5 shows the variation of thermal efficiency as a function of mass flow rate for various values of channel depth. The inlet air temperature is one degree higher than the ambient temperature. It is evident from Fig. 5 that the thermal efficiency is high at the lower value of channel depth, regardless of other parameters and for higher thermal efficiency the channel depth should be low. This is because the convective heat transfer coefficient increases with the decrease in channel depth to enhance the heat transfer between the absorber plate and flowing air in the channel.
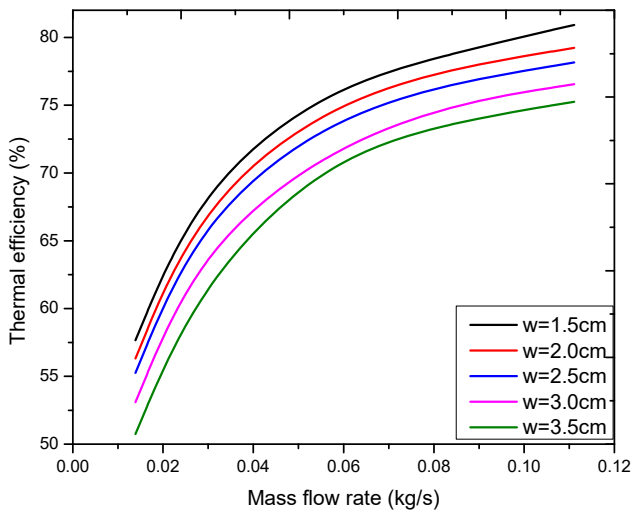

Fig. 5 Variation of thermal efficiency for different channel depth

Fig. 6 shows the thermal efficiency as a function of temperature rise parameter for various channel depth. It can be seen that the lower channel duct improves the thermal efficiency significantly. This can be attributed to the increased velocity of air through the channel duct. The channel depth of 1.5 cm shows the higher thermal efficiency of 81.6 %. When the channel depth approaches the minimum value, both heat transfer rate and fan power consumption significantly intensifies. In the other words, both heat transfer rate and fan power consumption decrease as the channel depth increases. From the plot, a gradual drop in thermal efficiency at any mass flow rate implies that the effect of heat transfer rate is dominant in the case of thermal efficiency.

Fig. 7 shows the plot for pressure drop along the collector as a function of mass flow rate for various channel depth. It can be seen from plot that the pressure drop increases with the increase in mass flow rate for all the channel depth. Maximum pressure drop has been shown with the lower channel depth.

For the channel depth as narrow as 1.5 cm, the flow friction along the channel length increases significantly and it results in great pressure drop at higher mass flow rate.
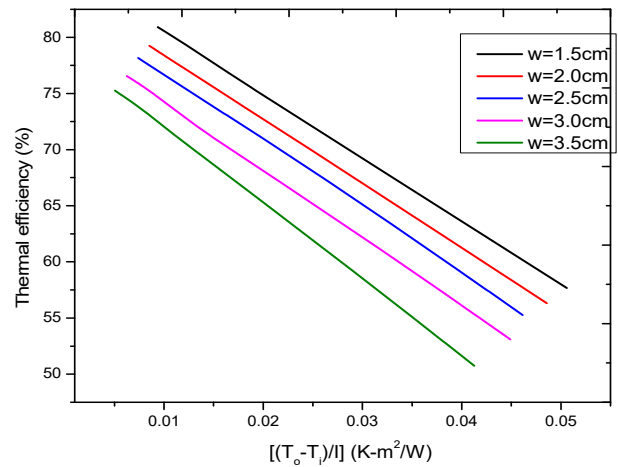

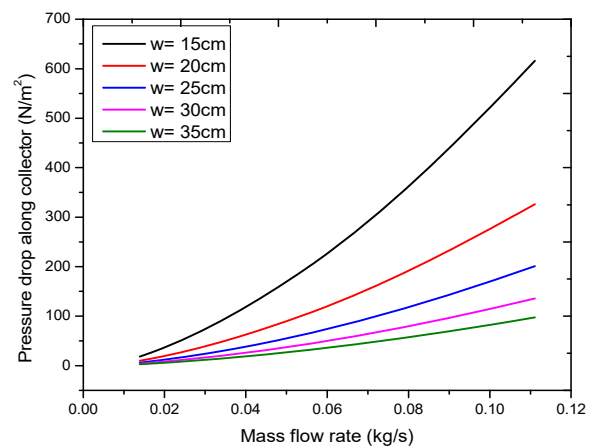Fig. 6 Variation of thermal performance for different channel depth

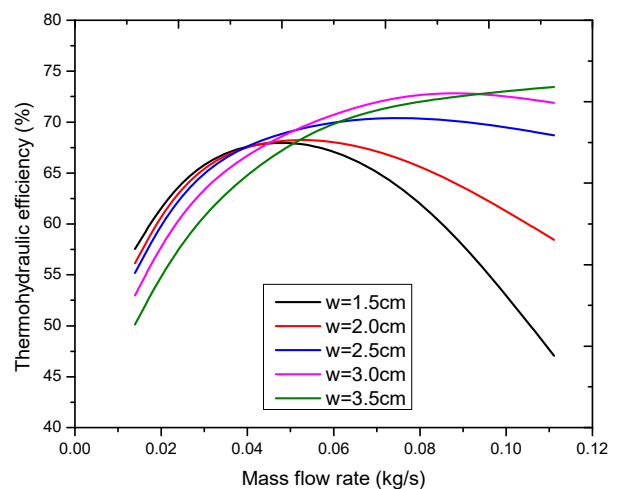Fig. 7 Variation of pressure drop across collector for different channel depth

Fig. 8 Variation of thermohydraulic efficiency for different channel depth

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Fig. 8 shows the thermohydraulic efficiency as a function of mass flow rate for various channel depth. It can be seen from the plots in Fig. 8 that the thermohydraulic efficiency decreases drastically when the flow rate is increased. The thermohydraulic efficiency at mass flow rates greater than 0.045 kg/s is better for channel depth of 3.0 cm followed by 3.5 cm, 2.5 cm, 2.0 cm, and 1.5 cm. Below this mass flow rate, reverse trend is observed.

V. CONCLUSION

The influence of the channel depth on the thermal and thermohydraulic performance of wavy fin absorber solar air heater has been studied using a mathematical model for the range of design and operating parameters. The important findings of the study are:

1. The maximum value of loss coefficient is obtained as 6.403 and 6.52 W/m^2-K for the minimum mass flow rate of 0.01 kg/s and channel depth of 1.5 and 3.5 cm, respectively.

2. A maximum air temperature rise has been achieved as 45.6 and 37.5 K for the channel depth of 1.5 and 3.5 cm, respectively.

3. The thermal efficiency is high at the lower value of channel depth, regardless of other parameters and for higher thermal efficiency the channel depth should be low.

4. Narrower channel depth shows higher flow friction along the channel length and it results in great pressure drop at higher mass flow rate.

5. Based on thermal analysis, the narrower channel shows better thermal efficiency irrespective of all mass flow rates but based on thermohydraulic analysis, narrower channel depth showed better thermohydraulic efficiency up to the mass flow rate of 0.041 kg/s but at higher mass flow rate (> 0.045 kg/s), reverse trends starts.

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