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COMPARISON OF THE PERFORMANCE OF ENHANCED SOLAR AIR HEATERS HAVING TURBULENCE PROMOTERS

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Abstract A solar air heater is a simple device to heat air by utilizing solar energy. The main drawback of a solar air heater is that the heat transfer coefficient between the absorber plate and the air stream is low, which results in a lower thermal efficiency of the heater. To overcome the low value of heat transfer between the absorber plate and the air stream, the underside of the absorber plate is roughened with artificial roughness elements, which act as turbulence promoters. Use of these turbulence promoters improves the convective heat transfer by creating turbulence in the flow. However, it would result in an increase in friction losses and hence greater power requirements from a fan or blower. In this study a comparison of two solar air heaters having expanded metal mesh and transverse wire roughness is made on the basis of reduced heat transfer surface for fixed heat duty and pressure drop, increased heat duty for fixed surface area, and reduced pumping power for fixed heat duty and surface area.

Keywords: Solar air heaters, heat transfer enhancement, artificial roughness

INTRODUCTION

Flat plate solar air heaters have been extensively employed to deliver air at low temperatures for crop drying, seasoning of timber, and curing industrial products. Conventional solar air heater as shown in Fig. 1 mainly consists of an absorber plate with a parallel plate forming a passage for air flow. A glass or plastic cover is fixed above the absorber plate and the system is insulated thermally from the back and from its sides. Solar air heaters have a number of advantages over solar water heaters, which include reduced risks of corrosion, low maintenance and low costs. The efficiency of flat plate solar air heaters has been found to be low because of low heat transfer coefficient between the absorber plate and the flowing air, which increases the absorber plate temperatures, leading to higher heat losses to the environment. Hence, different modifications are suggested and applied to improve the heat transfer coefficient between the absorber plate and air. One of the methods is installation of turbulence promoters in the form of artificial roughness on the underside of the absorber plate. Such methods can produce significant heat transfer enhancement, but there is also an increased pressure drop penalty. The application of artificial roughness has been recommended to enhance heat transfer coefficient by several investigators like [Cortes and Piacentini, 1990], [Gupta, 1993], [Dipprey and Sabersky, 1963], [Qayoum and Saini, 2000], [Webb and Eckert, 1972], [Webb et al., 1971] and [Prasad, 1991]. They used different geometries and orientations of these roughness

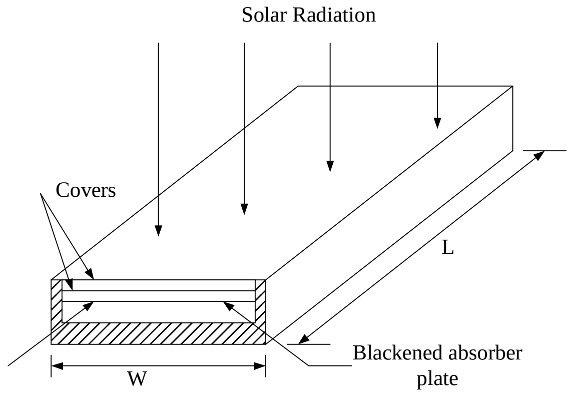

Fig. 1 Conventional Flat Plate Solar Collector

elements and arrived at various correlations for heat transfer coefficient and friction factors. Our primary geometries. The primary interest is to determine how

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the enhanced surface will affect the performance of the air heater. In this study computer codes have been developed using the existing correlation for different surfaces for evaluating the performance of such type of air heaters. A quantitative method is required to evaluate the performance improvement by given enhancement. Normally, one compares the performance of an enhanced surface with that of the corresponding smooth surface. The preferred evaluation method sets a performance objective and calculates the performance improvement relative to reference design (smooth surface) for a given set of operating conditions and design constraints. Hence this method defines the improvement of the objective function relative to the smooth surface. In this work the performance evaluation criteria proposed by [Webb, 1981] have been used. The performance objectives may be defined such that the enhanced surface is required to do better job than the reference smooth surface for established constraints. There are three design objectives and they are:

i) reduced heat transfer surface material for equal pumping power and heat duty,
ii) increased heat duty for fixed surface, and
iii) reduced pumping power for equal heat duty and surface area.

NOMENCLATURE

A	Collector area, m^2
A/A_s	Relative heat transfer area
A_c	Duct cross-section area, m^2
c_p	Specific heat of air, $Jkg^{-1}K^{-1}$
e	Roughness height, m
e/D	Relative roughness height
e^+	Roughness Reynolds Number, $(e/D)Re\sqrt{f/2}$
f	Friction factor
G	mass velocity, ρA_c
G^*	Relative mass velocity, G_s/G
h	Convective heat transfer coefficient,
I	Intensity of solar radiation, Wm^{-2}
K	Overall heat conductance, hA
k	Thermal conductivity of air, $Wm^{-1}K^{-1}$
K/K_s	Relative heat conductance
L/e	Relative shortway length
Nu	Nusselt number, hL/k
P	pitch, m
P	Pumping power, W
p/e	Relative roughness pitch
P/IA	Pumping power per insolation
P/P_s	Relative pumping power
Pr	Prandtl Number
Q_u	Useful heat gain per insolation
Re	Reynolds number
S/e	Relative longway length
St	Stanton Number
T_a	Air exit temperature, ^oC or K
T_a	Ambient temperature of air. ^oC or K
T_i	Air inlet temperature, ^oC or K
T_m	Bulk mean temperature of air, ^oC or K
T_p	Mean plate temperature, ^oC or K
V	Velocity of air in the duct, ms^{-1}
	$W m^{-2}K^{-1}$
$\dot{m}$	Mass flow rate of air, kgs^{-1}
$\Delta T/I$	Temperature rise parameter, $^oCW^{-1}m^{-2}$
Δp	Pressure drop, Nm^{-2}
$\langle\tau\alpha\rangle$	Transmittance-absorptance product of absorber cover combination
η_{th}	Thermal efficiency, Q_u/IA

Subscripts

Unsubscripted variables refer to rough surfaces
Subscripted s refer to smooth surface

CORRELATION FOR HEAT TRANSFER COEFFIENT AND FRICTION FACTOR

Solar air heaters having underside of the absorber plate roughened with expanded metal roughness and transverse wire rib roughness are used. Fig. 2 shows an absorber plate having roughness elements. The parameters for absorber plates having expanded and transverse wire rib roughness are shown in Fig. 3

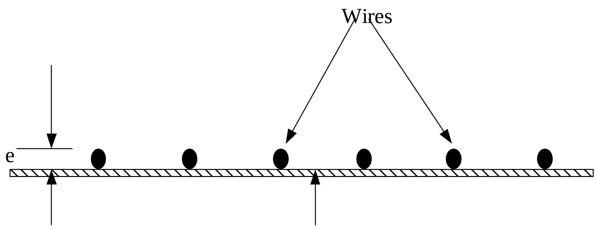

Fig. 2 Absorber plate showing roughness elements

The correlations for predicting heat transfer coefficient and friction factor for expanded metal mesh roughness have been developed by [Saini and Saini, 1997] and are

$$Nu = 4.0x10^{-4}\ \mathrm{Re}^{1.22}(e/D)^{0.625}(S/10e)^{2.22} \\ x[Exp\{-1.25(\ln(S/10e))^2\}(L/10e)^{2.66}] \\ x[\mathrm{Exp}\{-0.824(\ln(L/10e))^2\}] \quad (1)$$

$$f = 0.815\,\mathrm{Re}^{-0.361}(L/e)^{0.266} \\ x(S/10e)^{-0.190}(10e/D)^{0.591} \quad (2)$$

The correlations for predicting heat transfer coefficient and friction factor for transverse wire roughness have been developed by [Gupta et al., 1993] and are

For $e^+ < 35$

$$Nu = 0.000824(e/D)^{-0.178}(W/H)^{0.288}(\mathrm{Re})^{1.062} \quad (3)$$

And for $e^+ \geq 35$

$$Nu = 0.00307(e/D)^{-0.469}(W/H)^{0.245}(\mathrm{Re})^{0.812} \quad (4)$$

$$f = 0.06412\,(e/D)^{0.019}\,(W/H)^{0.237}\,(\mathrm{Re})^{-0.185} \quad (5)$$

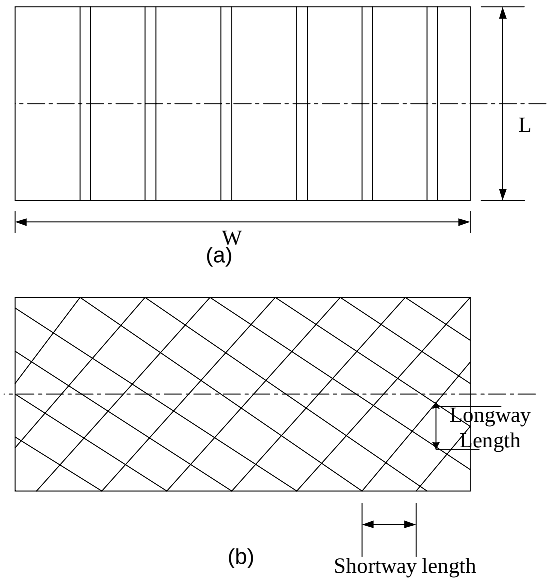

Fig. 3 Absorber plate having
(a) Transverse wire roughness
(b) Expanded metal mesh roughness

The correlations for predicting heat transfer coefficient and friction factor for smooth absorber plate are due Blausius and Dittus-Boelter and are given in [Incropera and Dewitt, 1996], and are

$$f_s = 0.079(\mathrm{Re})^{-0.25} \quad (7)$$

$$Nu_s = 0.23\,\mathrm{Re}^{0.8}\,\mathrm{Pr}^{0.6} \quad (8)$$

Stanton Number is calculated by

$$St = Nu / \mathrm{Re}\,\mathrm{Pr} \quad (9)$$

SOLUTION PROCEDURE

Case A (Reduced surface area)

To obtain reduced heat transfer surface area for equal heat transfer exchange and friction power the steps are:

The relative mass velocity, G^* as given by [Webb,1981],

$$G^* = \left[(f/St)/(f_s/St_s)\right]^{\frac{1}{3}} \quad (10)$$

Starting with arbitrary values of G*, f/St is calculated for known values of Re_s and Pr.

From Eqs. (1) to (9), relative roughness height (e/D), friction factor (f) and Stanton Number (St) are calculated for absorber plates having expanded metal mesh or transverse wire roughness.

The relative heat transfer surface area, A/A_s as proposed by [Webb, 1981] is then calculated as

$$A/A_s = (f/f_s)^{\frac{1}{2}} \Big/ (St/St_s)^{\frac{3}{2}} \quad (11)$$

Case B (Increased heat transfer)

To obtain increased heat exchange capacity for equal surface area and friction power the steps are:

The friction factor of the roughened absorber plate is calculated for the above-mentioned condition using arbitrary values of $G^*$and known values of Re_s and Pr using

$$f = f_s (G^*)^3 \quad (12)$$

Using Eqs. (2) and (5), the value of e/D is calculated which will give the above value of friction factor and using Eqs. (1), (3), (4) and (9) the value of Stanton Number for roughened absorber plate is calculated. The relative conductance of the roughened and smooth absorber plates is K/K_s and is calculated as proposed by [Webb, 1981] as

$$K/K_s = (St/St_s) \Big/ (f/f_s)^{\frac{1}{2}} \quad (13)$$

Case C (Reduced friction power)

To reduce the friction power expenditure for equal heat transfer surface area the steps are:

The Stanton Number for the above-mentioned condition for a solar heater having roughened absorber plate is calculated for arbitrary value of G^* and known values of Re_s and Pr as

$$St = St_s G^* \quad (14)$$

Using Eqs. (1), (3), (4), and (9) the value of e/D is calculated corresponding to the value of Stanton Number predicted from Eq. (15).

The friction factor is calculated from Eqs. (2) and (7) for the corresponding value of e/D.

The relative friction power P/P_s is calculated as proposed by [Webb, 1981] as

$$P/P_s = (f/f_s)/(St/St_s)^3 \quad (15)$$

RESULTS AND DISCUSSIONS

Case A (Reduced surface area)

Fig. 4 and Fig. 5 shows the graphical results for Case A in the form A/A_s vs. G^* for Re_s=14000 and Re_s =18000 for expanded metal mesh and transverse wire roughness. From these curves it is quite clear that as G^* increases A/A_s decreases. This is because with the increase in the value of G^* the mass flow in the roughened duct decreases and obviously less surface area is needed for the fixed heat duty at lower mass flows. Also for expanded metal mesh roughness the reduction in the surface area is more pronounced for the same heat exchange and friction power.

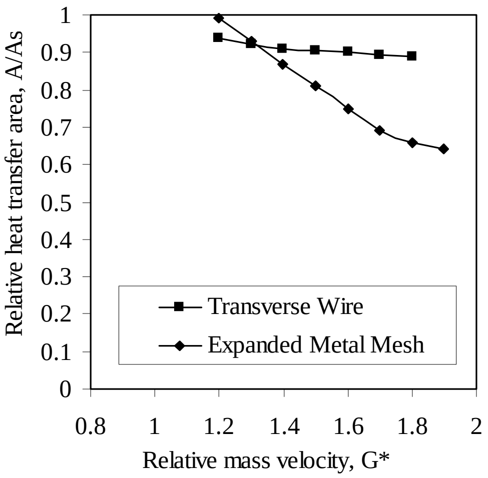

Fig. 4 Effect of Relative mass velocity on Relative heat transfer area at Reynolds Number of 14000 for smooth absorber plate

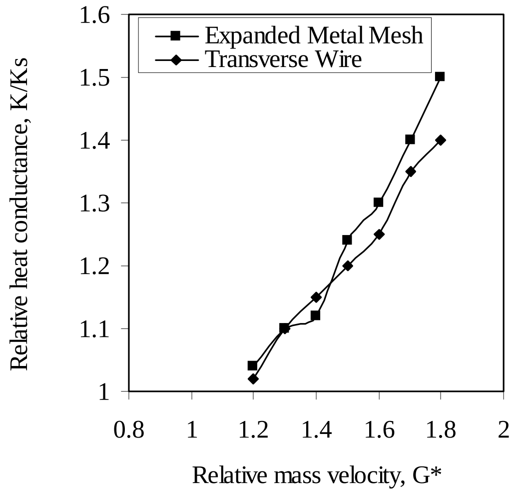

Fig. 6 Effect of Relative mass velocity on Relative heat conductance Vs at Reynolds Number of 14000 for smooth absorber plate

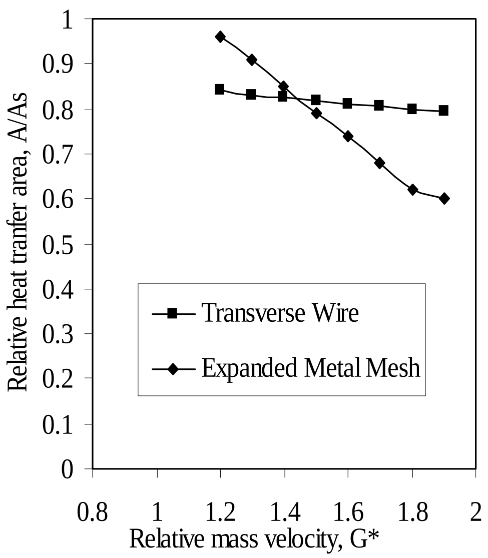

Fig. 5 Effect of Relative mass velocity on Relative heat transfer area at Reynolds Number of 18000 for smooth absorber plate.

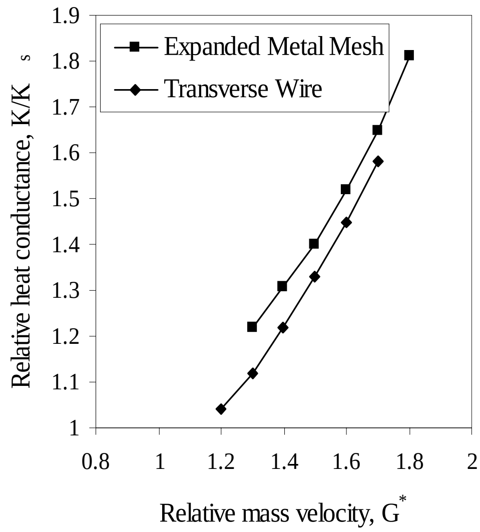

Fig. 7 Effect of Relative mass velocity on Relative heat conductance Vs at Reynolds Number of 18000 for smooth absorber plate.

Case B (Increased heat Transfer)

Fig. 6 and Fig. 7 shows the graphical results for case B in the form of K/K_s vs. G^* for Re_s =14000 and Re_s =18000 for expanded metal mesh and transverse wire mesh roughness. The relative conductance increases with the increase in G^*. Also relative conductance of expanded metal mesh roughness is more than that of expanded metal mesh roughness for the same surface area and friction power.

Case C (Reduced pumping power)

Fig. 8 and Fig. 9 shows the graphical results for Case C in the form of P/P_s vs. G^* for Re_s =14000 and Re_s =18000 for expanded metal mesh and transverse wire mesh roughness. The relative pumping power decreases with the increase in G^* which is quite obvious as less pumping power is needed at lower mass flows. For expanded metal mesh roughness the reduction in relative pumping power is more as is obvious from the graph for the same heat duty and same heat transfer area. The effect of Reynolds number is very small.

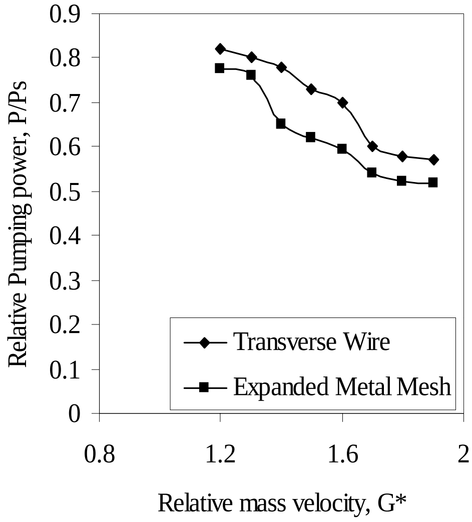

Fig. 8 Effect of Relative mass velocity on Relative pumping power at Reynolds Number of 14000 for smooth absorber plate

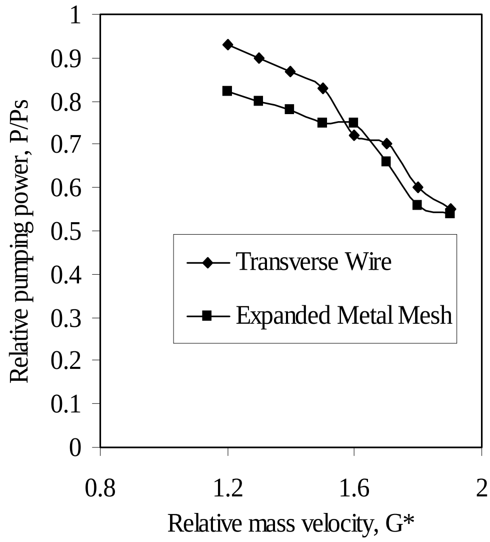

Fig. 9 Effect of Relative mass velocity on Relative pumping power at Reynolds Number of 18000 for smooth absorber plate

CONCLUSIONS

From this study the following conclusions can be drawn:

1. Solar air heaters having absorber plates roughened with expanded metal mesh roughness show better performance in all the three cases.
2. It is observed that at very small mass flow rates the performance of solar air heaters having underside of the absorber plate roughened is better in all the three cases.

REFERENCES

Cortes, A. and Piacentini, R. "Improvement of the Efficiency of a Bare Solar Collector by Means of Turbulence Promoters", Applied Energy, Vol. 36, 253-261, 1990.

Dipprey, D.F., and Sabersky, R.H. "Heat and momentum transfer in smooth and rough tubes at various Prandtl numbers", Int. J. Heat Mass Transfer, Vol.6, 329-353, 1963.

Gupta, D. "Investigations on fluid flow and heat transfer in solar air heaters with roughened absorbers", Ph.D. Thesis, University of Roorkee, June 1993.

Gupta, D., Solanki, S.C. and Saini, J. S. "Thermo-hydraulic Performance of Solar Air heaters with Roughened Absorber Plates", Solar Energy, Vol.61, No.1, 33-42, 1997.

Gupta, D., Solanki, S.C., and Saini, J.S. "Heat and fluid flow in rectangular solar air heater ducts having transverse rib roughness on absorber plates" , Solar Energy, Vol. 51, No.1, 31-37, 1993

Incropera, F.P. and DeWitt, D. P. "Fundamentals of Heat and Mass Transfer", John Wiley & Sons. Inc New York, pp 420-461 (1996)

Prasad, B.N., and Saini, J.S. "Artificially roughened solar air heaters," Solar Energy, Vol. 47, No. 2, 91-96, 1991.

Qayoum, A. and Saini, J.S. "Optimal design of Artificially Roughened Solar Air-Heaters", 15th National and 4th ISHMT/ASME Heat and Mass Transfer Conference at Pune, Jan., 12-14, 2000, pp 913-918

Saini, R.P. and Saini, J.S. "Heat Transfer and friction correlations for artificially roughened ducts with expanded metal mesh as roughness element", Int. J. of Heat Mass transfer, Vol.40, No.4, 973-986, 1997.

Webb, R.L. "Performance Evaluation Criteria of Enhanced Heat Transfer Surfaces in Heat Exchanger Design", Int. J. Heat Mass transfer, Vol.4, 715-726, 1981.

Webb, R.L. and Eckert, E.R.G. "Application of Rough Surfaces to Heat Exchange Design", Int. J. Heat Mass transfer, Vol.15, 1647-1658, 1972.

Webb, R.L., Eckert, E.R.G., and Goldstein, R.G. "Heat Transfer and friction in tubes with repeated-rib-roughness", Int. J. Heat Mass transfer, Vol.14, 601-617, 1971.