PERGAMON

Available online at www.sciencedirect.com

SCIENCE DIRECT

Renewable Energy 29 (2004) 211–222

RENEWABLE ENERGY

www.elsevier.com/locate/renene

Natural convective heat transfer in trapezoidal enclosure of box-type solar cooker

Subodh Kumar *

Centre for Energy Studies, Indian Institute of Technology, Hauz Khas, New Delhi 110016, India

Received 12 February 2003; accepted 30 May 2003

Abstract

This paper presents simple thermal analysis to evaluate the natural convective heat transfer coefficient, h_{c12} for a trapezoidal absorber plate-inner glass cover enclosure of a double-glazed box-type solar cooker. Several indoor simulation experiments in steady state conditions have been performed to measure the temperatures of absorber plate, inner and outer glass covers, ambient air, electrical input supply and wind speed. The experimental data has been correlated by an equation of the form, $\mathrm{Nu} = C\mathrm{Ra}^n$. The values of the constants C and n, obtained by linear regression analysis are used to calculate the convective heat transfer coefficient. The heat transfer analysis predicts that h_{c12} varies from 4.84 to 6.23 W m^{-2} °C^{-1} for the absorber plate temperature from 54 to 141 °C. The results of h_{c12} are compared with those of rectangular enclosure for the same absorber-inner glass cover temperatures and gap spacing. The study reveals that the values of convective heat transfer coefficient and top heat loss coefficient for rectangular enclosure are lower by 31–35% and 7% respectively.

Keywords: Box-type solar cooker; Convective heat transfer coefficient; Trapezium enclosure; Top heat loss coefficient

1. Introduction

The box-type solar cookers are constructed usually either with a rectangular or trapezium absorber plate-glass cover enclosure. The rectangular enclosure is simpler to construct and design and therefore more research work has been concentrated towards the experimentation and thermal modeling. The maximum plate temperature

* Tel.: +91-11-659-6409; fax: +91-11-6855-227.
E-mail address: subodh@ces.iitd.ernet.in (S. Kumar).

0960-1481/$ - see front matter © 2003 Elsevier Ltd. All rights reserved.
doi:10.1016/S0960-1481(03)00193-9

Nomenclature

A	aperture area of cooker (m^2)
C, n	constants in Nusselt Eq. (4)
h_{c12}	convective heat transfer coefficient between absorber plate and inner glass cover ($W\ m^{-2}\ K^{-1}$)
h_{r12}	radiative heat transfer coefficient between absorber plate and inner glass cover ($W\ m^{-2}\ K^{-1}$)
k	thermal conductivity of air ($W\ m^{-1}\ K^{-1}$)
L_{12}	characteristic length (m)
T_1	plate temperature (K)
T_2	inner glass temperature (K)
T_{m12}	mean temperature of absorber plate and inner glass cover (K)
Q_{net}	net upward heat flow (W)
U_t	top heat loss coefficient ($W\ m^{-2}\ K^{-1}$)

Greek letters

ϵ_p	emissivity of absorber plate (0.90)
ϵ_g	emissivity of glass cover (0.88)
σ	Stefan-Boltzman constant ($5.67 \times x10^{-8}\ W\ m^{-2}\ K^{-4}$)
β	coefficient of volumetric of expansion (K^{-1})
ν	kinematic viscosity of air ($m^2\ s^{-1}$)
μ	dynamic viscosity ($kg\ m^{-1}s^{-1}$)

Dimensionless terms

Gr	Grashof number ($g\beta(T_1-T_a)L_{12}{}^3/\nu^2$)
Nu	Nusselt number ($h_{c12}L_{12}/k$)
Pr	Prandtl Number ($c\mu/k$)
Ra	Rayleigh number (Gr Pr)

attained in any given solar cooker for a given climatic conditions such as intensity of solar radiation, wind speed and ambient air temperature depends on overall thermal losses from the cooker. The top heat losses that constitute the major losses have a strong influence on the thermal performance. The top heat loss coefficient U_t for a box-type solar cooker with double glazing is similar to that for a double-glazed flat-plate solar collector. The top heat loss coefficient U_t can be written in terms of individual heat transfer coefficient as:

$$U_t^{-1} = (h_{c12} + h_{r12})^{-1} + (h_{c23} + h_{r23})^{-1} + (h_w + h_{r3a})^{-1} \tag{1}$$

The computation of U_t requires the knowledge of radiative and convective heat transfer for the various components of solar cooker. The radiative heat transfer coefficients between absorber plate and inner glass cover (h_{r12}), within the glass covers (h_{r23}) and outer glass cover and surrounding air (h_{r3a}) can easily be calculated by using the standard relations [1]. Many correlations for wind heat transfer coefficient (h_w) as a function of wind speed are also available in the literature [2–5]. There also exist experimental and analytical studies on natural convective heat transfer within the rectangular enclosures. Buchberg et al. and Hollands et al. [6–7] proposed the correlation for the Nusselt number for the natural convective heat transfer coefficient between two parallel rectangular plates. Recently, Samdarshi and Mullick [8] suggested an approximate equation for the convective heat transfer coefficient for horizontal rectangular enclosure in doubled-glazed flat plate solar collector.

The equation is

$$h_{c12} = \frac{5.78[(T_1 - T_2)]^{0.27}}{T_{m12}^{0.31} L_{12}^{0.21}} \tag{2}$$

where T_1 and T_2 are the absorber plate temperature and inner glass cover temperature respectively, T_{m12} is the mean temperature of plate and inner glass temperature and L_{12} is the air gap between the plate and inner glass cover.

The computation of natural convective heat transfer within the trapezoidal enclosure is quite complex due to irregular geometry of the enclosure. It has been noticed from the literature review that recently, Mullick et al. [9] conducted indoor experiments on the double-glazed box-type solar cooker with trapezoidal enclosure and suggested a set of equations for individual heat transfer coefficients to correlate top heat loss coefficient. The approximate equation for convective heat transfer coefficient for absorber plate-inner glass trapezoidal enclosure is:

$$h_{c12} = \frac{15.4[(T_1 - T_2)]^{0.285}}{T_{m12}^{0.34}} \tag{3}$$

The paper presents a comparison of results for computed values of h_{c12} obtained from the proposed correlation and Eq. (3) for the same dimensions of the cooker and the temperature difference between absorber and inner glass cover. It also reports difference in the values of U_t obtained by using correlations of h_{c12} for trapezium and rectangular enclosures.

2. Experimental arrangement and procedure

A commercially available double-glazed box type solar cooker of fiberglass body (specifications in Appendix A) has been used for the indoor experiment. Fig. 1 shows the experimental test set-up. The dull black painted aluminum trapezoidal shaped tray with an aperture of 0.245 m^2 was fitted inside the cooker. Glasswool insulation was used on bottom and sides of the cooker to minimize the thermal losses through conduction. The tray (0.380×0.380 m in size at the bottom) was heated by five plate-

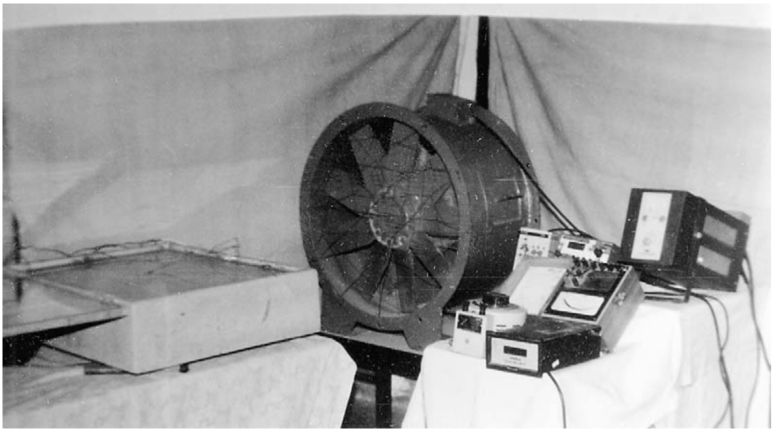

Fig. 1. Indoor experimental arrangement.

type electric heaters (each 0.380 m long) of total capacity 250 W. These were fixed on the underside of the aluminum tray to provide uniform heating through a servo-stabilized electric power supply. The four inclined sides of the tray (120° to horizontal) were also individually heated by another set of four identical plate heaters of total capacity 100 W made to exact size of the sides as shown in Fig. 2. All the experiments were performed by adjusting the heat flux through side heaters to 50% of the heat flux provided by the bottom heaters. This condition simulates the situation when the sun is directly overhead (hence there is complete thermal symmetry). The plate and sides were heated to different temperatures by adjusting the power input supply through separate variac. The power input to each heater was measured by a portable single phase wattmeter (accuracy within ±0.5% of full scale value 150 W). All the temperatures were measured by calibrated copper-constantan thermocouples

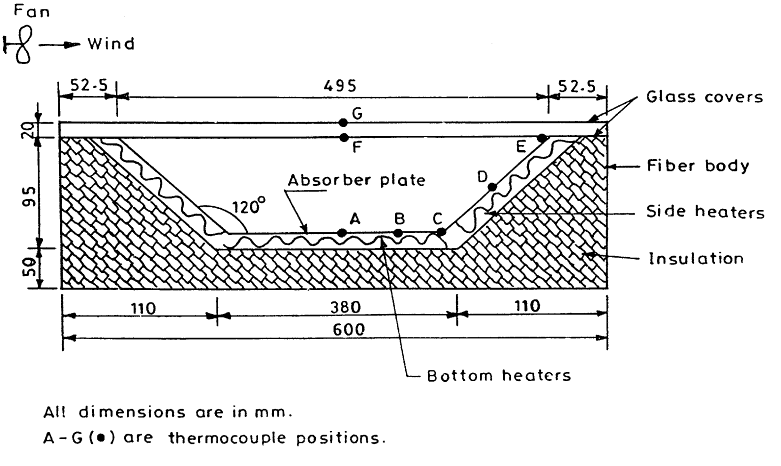

Fig. 2. Schematic diagram of a box-type solar cooker in indoor condition.

with the help of digital micro-voltmeter (with sensitivity of 0.01 °C). Calibrated copper–constantan thermocouples were fixed at different locations on the absorber plate as well as in the centre of the inner glass cover. Another thermocouple was embedded in the centre of the outer glass cover (by cutting a fine groove). A schematic diagram of the solar cooker showing locations of thermocouples is presented in Fig. 2. The plate temperature and side temperature of the cooker were determined as the average of three temperatures of thermocouples A, B, C and C, D, E respectively. The ambient air temperature was also measured through a separate thermocouple along with these observations. Forced airflow over the outer glass cover of the cooker was produced by a 1 hp, 3-phase tube axial-flow fan. Different wind speeds were produced by controlling the air supply on the suction side of the fan. A 3-cup anemometer using chopper type sensor was used for measurement of wind speed (least count 0.1 m s^{-1}).

Losses by conduction through glasswool below and at the sides of the cooker plate (Q_b) were subtracted from the sum of electric power input (Q_{in}) supplied to the bottom and sides heaters to obtain the net upward heat flow (Q_{net}). Several experiments were conducted to obtain different values of Q_{net} over a range of plate temperature from 54 to 141 °C (by adjusting the power input to the heaters) and wind speed from 1.1 to 2.6 m s^{-1}. Each observation was finalized in 6–8 h to obtain a very good steady state. The experimental observations are presented in Table 1.

3. Thermal analysis

The natural convective heat transfer coefficient, h_{c12} for the absorber plate-glass enclosure can be computed using the expression for the Nusselt number as [10]:

$$\mathrm{Nu} = \frac{h_{c12}L_{12}}{k} = C(\mathrm{GrPr})^n \tag{4}$$

or

$$h_{c12} = \frac{k}{L_{12}}C(\mathrm{GrPr})^n \tag{5}$$

where L_{12} is the characteristic length (the distance between the absorber plate and inner glass cover) and k is the thermal conductivity of the air at the mean temperature of the plate-glass enclosure.

Under steady state condition, the net input electrical energy (Q_{net}) is equal to the rate of heat loss (or net upward heat flow) from the plate to the inner glass cover. The upward heat loss from the cooker absorber plate at an average temperature T_1 to the inner glass cover at an average temperature T_2 is given by

$$Q_{net} = A(h_{c12} + h_{r12})(T_1 - T_2) \tag{6}$$

where A the aperture area of solar cooker and h_{c12} and h_{r12} are the natural convective and radiative heat transfer coefficients respectively between plate and inner glass

Table 1
Indoor experimental measurements and results for Ra and h_{c12}

Absorber plate temperature T_1 (°C)	Inner glass temperature T_2 (°C)	Outer glass temperature T_3 (°C)	Ambient air temperature T_a (°C)	Total electric supply to heaters Q_{in} (W)	Rayleigh number (Ra)	Natural convective heat transfer coefficient h_{c12} (W/m^2 °C)
Wind speed,V = 1.1 m/s						
58.40	43.37	24.00	16.90	68.1	788,130	4.88
67.03	49.00	25.60	16.09	85.0	858,314	5.17
75.70	53.90	26.84	15.90	106.4	948,378	5.49
85.60	61.20	29.55	16.20	124.8	949,375	5.61
98.90	70.30	32.50	16.40	154.2	966,174	5.81
109.03	77.00	34.90	15.80	178.9	975,879	5.95
117.9	83.46	37.95	16.90	197.9	957,057	6.00
Wind speed, V = 1.8m/s						
59.30	45.00	25.25	18.90	68.5	736,984	4.87
76.26	55.95	28.65	18.90	99.8	868,561	5.40
87.87	63.65	31.25	18.80	124.8	914,386	5.65
99.52	71.68	34.00	18.90	150.0	928,910	5.82
110.15	78.82	36.24	18.45	175.9	937,714	5.97
120.03	85.80	38.30	18.60	201.4	926,366	6.05
141.06	100.20	42.60	18.80	255.3	901,324	6.23
Wind speed, V = 2.6 m/s						
54.25	39.18	19.93	14.70	67.3	837,027	4.84
70.73	49.50	22.90	14.70	100.1	982,666	5.33
82.65	57.90	24.70	14.70	126.1	1,002,531	5.51
93.23	64.10	27.00	14.70	150.2	1,059,988	5.75
104.30	71.20	29.44	14.80	176.1	1,063,258	5.88
114.36	77.90	31.70	15.00	201.4	1,064,072	5.99
133.60	91.05	34.90	14.80	254.3	1,064,790	6.21

cover. The radiative heat transfer coefficient h_{r12} is computed using the following standard relation:

$$h_{r12} = \frac{\sigma(T_1^2 + T_2^2)(T_1 + T_2)}{1/\varepsilon_p + 1/\varepsilon_g - 1} \tag{7}$$

Substituting h_{c12} from Eq. (5), Eq. (6) becomes

$$\left[\left(\frac{Q_{net}}{A(T_1 - T_2)} - h_{r12}\right)\left(\frac{L_{12}}{k}\right)\right] = C(\mathrm{GrPr})^n \tag{8}$$

Putting

$$\left[\left(\frac{Q_{net}}{A(T_1 - T_2)} - h_{r12}\right)\left(\frac{L_{12}}{k}\right)\right] = Z \tag{9}$$

Eq. (8) becomes $Z = C(\mathrm{GrPr})^n$.

Taking the logarithm of both sides of the above equation, one can write

$$\ln Z = \ln C + n\ln(\mathrm{GrPr}) \tag{10}$$

Eq. (10) represents an equation of a straight line in the following form i.e.

$$Y = pX + q \tag{11}$$

where $Y = \ln Z$, $p = n$, $X = \ln[\mathrm{GrPr}]$ and $q = \ln C$.

The values of X and Y in Eq. (11) are calculated from the experimental data of the absorber plate and inner glass temperatures and upward heat flow values for different wind speeds, presented in Table 1. The different physical properties of air used for the computation of the Grashof number (Gr) have been evaluated at mean fluid temperature T_{m12} [8]. $k = 0.000206 T_{m12}^{0.85}$, $\nu = 9.0 \times 10^{-10} T_{m12}^{1.72}$, and $\beta = 1/T_{m12}$, where $T_{m12} = (T_1 + T_2)/2$.

The values of the constants p and q of Eq. (11) are obtained by using the following linear regression formulae.

$$p = \frac{N\sum XY - \sum X \sum Y}{N\sum X^2 - \left(\sum X\right)^2} \tag{12}$$

and

$$q = \frac{\sum X^2 \sum Y - \sum X \sum XY}{N\sum X^2 - \left(\sum X\right)^2} \tag{13}$$

where N is number of observations in each set.

Having found p and q, the values of C and n in Eq. (4) can be determined as follows:

$$C = e^q \text{ and } n = p$$

4. Results and discussion

The experimental data for absorber plate temperature, inner glass temperature obtained for different wind speeds at steady state condition have been used for determining the physical properties of the air which in turn are used for computing the values of the Grashof number. The constants C and n in Eq. (4) are obtained by linear regression analysis. The following correlations for the Nusselt number are proposed:

$$\mathrm{Nu} = 0.0427\mathrm{Ra}^{0.439} \text{ for wind speed} = 1.1\,\mathrm{m/s} \tag{14a}$$

$$\mathrm{Nu} = 0.0647\mathrm{Ra}^{0.410} \text{ for wind speed} = 1.8\,\mathrm{m/s} \tag{14b}$$

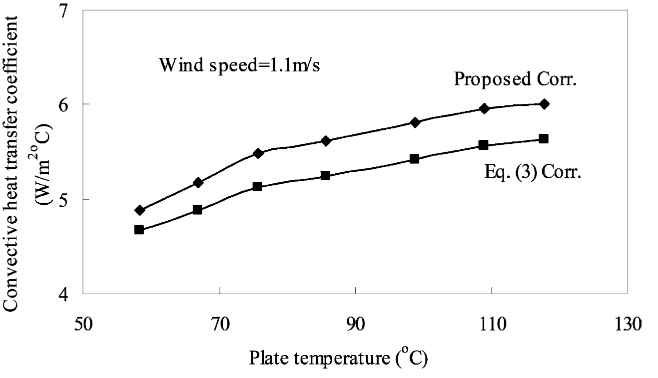

Fig. 3. Comparison of results for the convective heat transfer coefficient using proposed and Eq. (3) correlations for trapezium enclosure for a wind speed of 1.1 m s^{-1}.

$$\mathrm{Nu} = 0.0932\mathrm{Ra}^{0.380} \text{ for wind speed } = 2.6 \text{ m/s} \tag{14c}$$

The values of convective heat transfer coefficient, h_{c12} as a function of absorber plate temperature is evaluated by using the proposed correlations (Eqs.14(a–c)) for different wind speeds and the results are presented in Table1. The comparison of results for h_{c12} using the proposed correlations and Eq. (3) is depicted in Figs. 3–5. As expected, h_{c12} increases linearly with absorber-plate temperature for all values of wind speeds. It can also be noticed that the values of h_{c12} using the present correlations are in fairly good agreement with those obtained from Eq. (3). The percentage variation is found to vary between 3.1 and 6.7% for the wide range of absorber plate temperature from 54 to 141 °C and wind speed 1.1 to 2.6 m s^{-1}. The close agreement in results shows that the proposed correlations can predict natural convective heat transfer in the trapezium enclosure for absorber-plate temperatures of interest in a double-glazed box type solar cooker. Fig. 6 shows the effect of wind speed on the

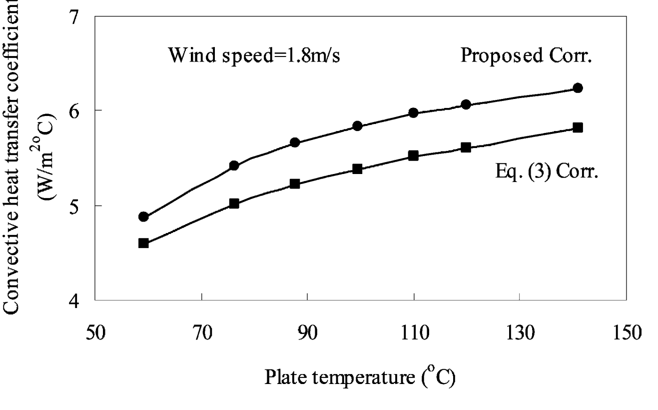

Fig. 4. Comparison of results for convective heat transfer coefficient using proposed and Eq. (3) correlations for trapezium enclosure for wind speed of 1.8 m s^{-1}.

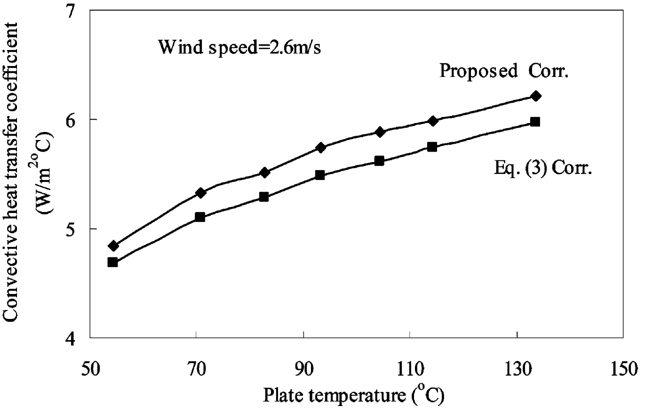

Fig. 5. Comparison of results for convective heat transfer coefficient using proposed and Eq. (3) correlations for trapezium enclosure for wind speed of 2.6 m s^{-1}.

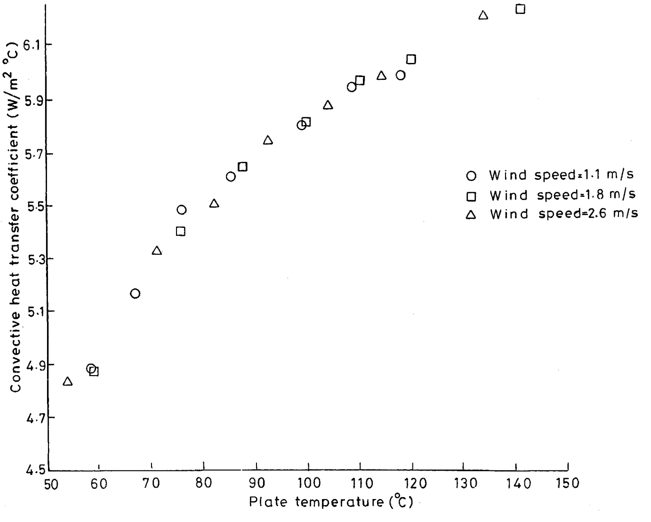

Fig. 6. Effect of wind speed on convective heat transfer coefficient within trapezium enclosure.

convective heat transfer coefficient. It can be observed that the variation in h_{c12} due to a change in wind speed is negligible due to insignificant change in the overall heat transfer coefficient for a double-glazed solar cooker. For the purpose of comparison, h_{c12} has also been computed for a rectangular enclosure using Eq. (2) for the same absorber plate and inner glass cover temperatures and gap spacing. Figs. 7–9 show that the values of h_{c12} for a rectangular enclosure are 31–35% lower than those computed for the trapezium enclosure. The standard relations of Buchberg et al. and Hollands et al. [6–7] for a rectangular enclosure are also used to compute h_{c12}. These computations also result in lower values of h_{c12} as compared to those obtained for

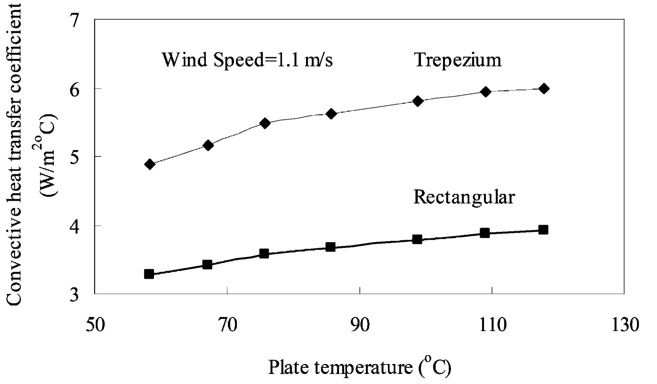

Fig. 7. Variation of convective heat transfer coefficient with absorber plate temperature for trapezium and rectangular enclosures for wind speed of 1.1 m s^{-1}.

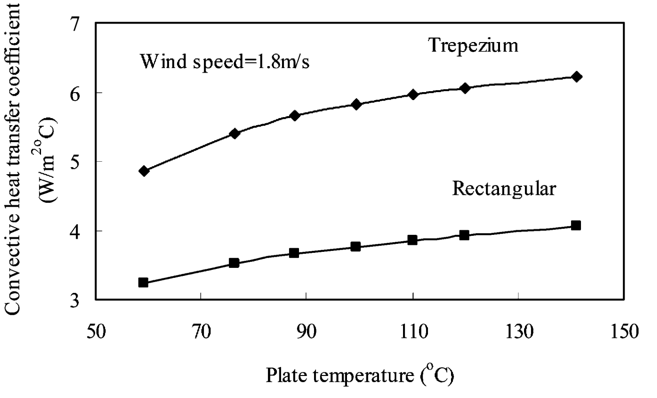

Fig. 8. Variation of convective heat transfer coefficient with absorber plate temperature for trapezium and rectangular enclosures for wind speed of 1.8 m s^{-1}.

the trapezium enclosure. Therefore greater care is required while using the relations for a rectangular enclosure in computing the convective heat transfer coefficient within the trapezium enclosure. The variation in the top heat loss coefficient U_t with absorber plate temperature for a trapezium and rectangular enclosures is shown in Fig. 10. It can be noticed that U_t for a rectangular enclosure is lower by 5–7%.

5. Conclusions

The natural convective heat transfer in a trapezium enclosure of a box-type solar cooker can be predicted by using the simple proposed correlations for a wide range of absorber-plate temperature. It is unreasonable to use the correlation(s) for rectangular enclosure in computing h_{c12} for trapezium enclosure due to under-estimated results.

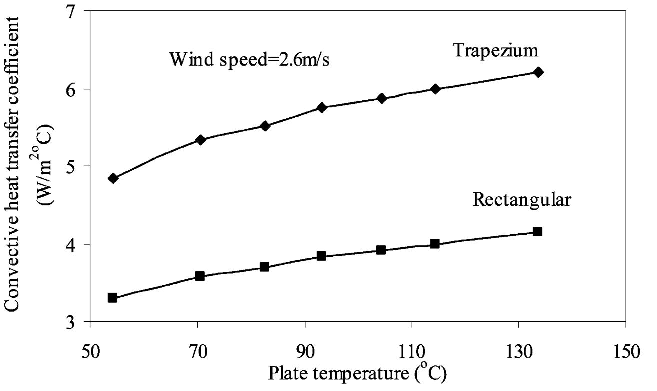

Fig. 9. Variation of convective heat transfer coefficient with absorber plate temperature for trapezium and rectangular enclosures coefficient for wind speed of 2.6 m s^{-1}.

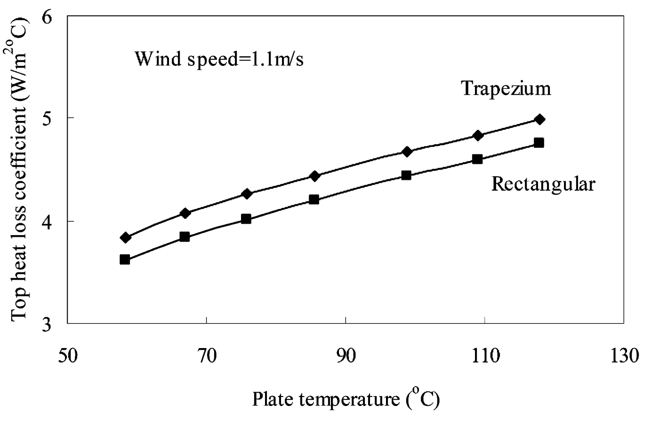

Fig. 10. Variation of top heat loss coefficient with absorber plate temperature for trapezium and rectangular enclosures.

The top heat loss coefficient U_t is found to be lower by 7% in the case of a rectangular enclosure. The major advantage of using a trapezoidal shaped tray in a box-type solar cooker is the absorption of a higher fraction of incident solar radiation falling on the aperture at larger incidence angles, due to a more exposed surface area. In addition, less absorber material is required, thus resulting in lower fabrication cost.

Appendix A

A commercially available box-type solar cooker having the following specifications has been used for experimentation:

Overall dimensions of cooker	$600 \times 600 \times 165$ mm
Material for outer casing of the cooker	Fiberglass
Dimension of tray shaped absorber plate	
Top	495×495 mm
Bottom	380×380 mm
Depth of the tray	95 mm
Emissivity of the absorber plate	0.90
Material for absorber plate material	Aluminum
Specific heat of absorber plate material	895 J kg^{-1} ^{o}C^{-1}
Thickness absorber plate	0.55 mm
Mass of absorber plate	0.325 kg
Thickness of glass covers	
Inner glass cover	3 mm
Outer glass cover	2 mm
Spacing between the glass covers	15 mm
Emissivity of glass cover	0.88
Type of insulation	glasswool
Thermal conductivity of insulation	0.05 W m^{-1} ^{o}C^{-1}
Thickness of insulation at the bottom	50 mm

References

[1] Duffie JA, Beckman WA, editors. Solar engineering of thermal processes. New York: John Wiley & Sons; 1991.
[2] McAdams WH. Heat transmission. New York: McGraw-Hill, 1954.
[3] Watmuff JH, Charters WWS, Proctor D. Solar and wind induced external coefficients for solar collectors. COMPLES 2:56
[4] Test FL, Lessmann RC, Johary A. Heat transfer during wind flow over rectangular bodies in natural environment. Transaction of ASME Journal of Heat Transfer 1980;103:262–7.
[5] Kumar S, Sharma VB, Kandpal TC, Mullick SC. Wind induced heat losses from outer cover of solar collectors. Renewable Energy 1997;10(4):613–6.
[6] Buchberg H, Catton I, Edwards DK. Natural convection in enclosed space: a review of application to solar energy collection. ASME Journal of Heat Transfer 1976;98(2):182–8.
[7] Hollands KGT, Unny TE, Raithby GD, Konicek L. Free convective heat transfer across inclined air layers. ASME Journal of Heat Transfer 1976;98(2):189–93.
[8] Samdarshi SK, Mullick SC. Analytical equation for the top heat loss factor of a flat-plate collector with double glazing. Journal of Solar Energy Engineering 1991;113:117–22.
[9] Mullick SC, Kandpal TC, Kumar S. Top heat loss factor of double-glazed box-type solar cooker from indoor experiments. Energy 1997;22(6):559–65.
[10] Cengel YA. Heat transfer: a practical approach. New York: WCB/McGraw-Hill, 1998.