Energy Conversion and Management 51 (2010) 329–337

Contents lists available at ScienceDirect

Energy Conversion and Management

journal homepage: www.elsevier.com/locate/enconman

Heat loss study of trapezoidal cavity absorbers for linear solar concentrating collector

Panna Lal Singh [a,*], R.M. Sarviya [b], J.L. Bhagoria [b]

[a] Central Institute of Agricultural Engineering, Berasia Road, Bhopal 462 038, MP, India
[b] Maulana Azad National Institute of Technology, Bhopal 462 007, MP, India

ARTICLE INFO

Article history:
Received 4 July 2008
Received in revised form 5 March 2009
Accepted 30 September 2009
Available online 24 October 2009

Keywords:
Overall heat loss coefficient
Trapezoidal cavity absorber
Rectangular pipe
Round tube
Linear Fresnel solar concentrating collector

ABSTRACT

There should be minimum heat loss from the absorber to achieve better efficiency of the solar collector. Overall heat loss coefficients of the trapezoidal cavity absorber with rectangular and round pipe were studied in the laboratory. Two identical rectangular pipe absorbers (section size: 100 × 23 mm, thickness: 2.5 mm and length 2170 mm) and two round pipe absorbers (a set of six mild steel round tubes of 16 mm diameter and 2.5 mm thickness brazed together in single layer making 100 mm width) were fabricated. A rectangular and a round pipe were painted with ordinary mat black paint (emissivity at 100 °C = 0.91) and one pipe of each type was coated with black nickel selective surface (emissivity at 100 °C = 0.17). Overall heat loss coefficient of the absorber was studied by circulating hot oil through it at different temperatures. The heat loss coefficient was increased with the absorber temperature. The heat loss coefficients for ordinary black coated and selective surface coated round pipe absorbers were varied from 3.5 to 7.5 W/m^2/°C and 2.7–5.8 W/m^2/°C respectively. The rectangular pipe section has marginally higher heat loss coefficients as compared to round pipe absorber. Selective surface coating on the absorbers reduced heat loss coefficient significantly by 20–30% as compared to ordinary black coating. The double glass cover also reduced heat loss coefficient by 10–15% as compared to single glass cover. The overall heat loss coefficients were also estimated analytically by parallel plate correlation and cavity correlations. The trend of variation of estimated heat loss coefficients by both methods was similar to experimental values. However, estimated values by cavity correlation were closure and uniformly distributed at all temperature range.

1. Introduction

The linear Fresnel solar concentrating device can be used for medium temperature (80–250 °C) applications [1,2]. A typical linear Fresnel reflector consists of a long narrow flat mirror elements fixed on a horizontal base. Each mirror element is tilted at an angle such that all incident solar rays falling on them are reflected to a common focus. Linear focus type concentrators tracked about the linear receiver or absorber. The absorber is normally a tube or series of tubes which contain a heat transfer fluid. Absorber of the solar concentrating device, play very important roll in collection of solar energy. To achieve higher efficiency of the solar collector, there should be minimum thermal losses from the absorber. The overall heat loss coefficient of the absorber includes convection, radiation and conduction heat losses [2–4]. Therefore, the overall heat loss coefficient is of great importance to the solar absorber. Heat loss coefficient is expected to change with surface coating, glass cover, shape and size of the absorber surface, temperature of the absorber, etc. and it could affect thermal performance of the absorber for solar concentrating device. Contact area of the fluid with the absorber surface plays important roll for heat transfer to the fluid. The glass cover reduces heat loss from the absorber due to wind effect [5]. The double glass cover with air trapped between the glasses act as transparent insulator and reduces the convection heat loss [2]. Unlike flat plate collectors, the concentrating collector have many different configurations, each case should be analysed specially [6].

Considerable research effort has gone into the understanding collection of solar radiation and heat transfer mechanism to the absorber. Negi et al. [7] and Khan [8] studied overall heat loss coefficient of the concentrically glass covered round tube absorber by circulating hot oil through the absorber. However, their study was limited up to 120 °C absorber temperature. The non-evacuated solar absorber painted ordinary black paint and covered with glass yield poor performance [1]. There is scope to reduce the heat loss coefficient in the trapezoidal cavity absorber. The trapezoidal cavity absorber can be insulated from three sides which does not receive the sun rays from the solar collector to minimize thermal losses. There is limited literature available on study of trapezoidal

* Corresponding author. Tel.: +91 755 2521127; fax: +91 755 2734016.
E-mail address: plsingh@ciae.res.in (P.L. Singh).

0196-8904/$ - see front matter © 2009 Elsevier Ltd. All rights reserved.
doi:10.1016/j.enconman.2009.09.029

Nomenclature

A_{abs}	surface area of the absorber pipe, m^2
A_p	surface area of the absorber pipe/plate facing cavity, m^2
A_{cg}	surface area of the glass cover, m^2
C_p	specific heat of fluid, kJ/kg/°C
C_{pa}	specific heat of the fluid (air), kJ/kg K
d	depth of the cavity, m
g	acceleration due to gravity, m^2/s
Gr	Grashof number
h_{cp}	convection heat transfer coefficient from absorber pipe/plate, W/m^2 K
h_{co}	convection heat transfer coefficient from outer glass cover to air, W/m^2 K
h_{rp}	radiation heat transfer coefficient from absorber pipe/plate to outer cover, W/m^2 K
h_{ro}	radiation heat transfer coefficients from outer glass cover to air, W/m^2 K
I	intensity of solar beam radiation, W/m^2
k_a	thermal conductivity of fluid (air), W/m^2
K_{gw}	thermal conductivity of the glass wool, W/mK
L_c	characteristic length of the absorber cavity (depth), m
L_{co}	characteristics length of outer glass cover, m
m	mass of fluid flow rate, kg/s
Nu	Nusselt number
Nu_{co}	Nusselt number for convection heat transfer from the outer glass cover surface
Nu_{cp}	Nusselt number for convection heat transfer between absorber plate and glass cover
N_{Rc}	radiation conduction interaction parameter, $\sigma\ (T_h)^4\ d/k(T_h\text{-}T_C)$
Pr	Prandtl number
Pr_{cp}	Prandtl number for heat transfer between absorber plate and glass cover
Pr_{co}	Prandtl number for heat transfer between outer glass cover and surrounding ambient
Re	Reynold number
T_i	inlet fluid temperature, °C
T_0	outlet fluid temperature, °C
T_a	ambient temperature, °C
T_{abs}	absorber temperature, °C
T_p	average absorber pipe/plate temperature, °C
T_h	temperature of the hot pipe/plate, K
T_C	temperature of the cold plate/ glass cover, K
T_R	temperature ratio, T_h/T_U
U_L	overall heat loss coefficient of absorber, $W/m^2/°C$
U_{L1}	convection and radiation heat loss coefficients from the absorber plate (through glass cover at bottom portion), $W/m^2/°C$
U_{L2}	conduction heat loss coefficients from insulated sides, $W/m^2/°C$
v	air velocity, m/s

Greek symbols

σ	Stefan–Boltzman constant, $5.6697 \times 10^{-8}\ W/m^2/K^4$
ΔT	temperature difference between hot plate and cold plate, K
ε	emissivity of the surface
ε_H	emissivity of the hot plate surface of the cavity
ε_C	emissivity of the cold plate surface of the cavity
ε_B	emissivity of wall surface at right side of the cavity
ε_T	emissivity of wall surface at left side of the cavity
ρ	density of the fluid (air), kg/m^3
μ	viscosity of the fluid (air), kg/m s

cavity absorber. Reynolds et al. [4] used flow visualization technique to capture the heat flow patterns within the trapezoidal cavity with hot plate to investigate the heat losses from absorber experimentally. The cavity was modeled using a commercial computational fluid dynamic (CFD) software package. Only reasonable agreement was found between computational and experimental heat transfer rate. Heat loss predicted by CFD model (623 W/m^2) was highly under-predicted (about 40%) as compared to the experimental results (1040 W/m^2) and the discrepancies could not be explained. In a study of the trapezoidal cavity, Natarajan et al. [9] found non-uniform heating of the bottom wall produced greater heat transfer rate as compared to uniform heating case for all Rayleigh numbers. The convection heat loss between absorber pipe and inner glass surface can be estimated by considering heat loss between two horizontal parallel plates (hot plate up and cold plate at bottom) [10]. Radiation heat transfer coefficient between hot absorber plate and glass cover of the cavity can be calculated considering radiative heat transfer modeled as that between two parallel planes [11]. Considering emissivities of the hot and cold plates and interaction of surface radiation with free convection, Balaji and Venkatesan [12] developed correlations for square cavity with air ($Pr = 0.71$) as the intervening medium.

The present study envisaged the overall heat loss coefficient study of the trapezoidal cavity absorber with rectangular pipe and round pipe with ordinary black coating and selective surface coatings at different absorber temperatures (up to 175 °C) and development of correlations for the heat loss coefficient. The study was aimed to assess overall heat loss coefficient for trapezoidal cavity absorber with change in the shape of the absorber pipe, surface area, surface coating, number of glass cover and absorber temperature experimentally. Heat loss in the trapezoidal cavity absorber was also analyzed critically and estimated analytically by parallel plate correlation and cavity correlation and compared with the experimental results. As there is very limited literature available on heat loss study on trapezoidal cavity absorber, the present study would be very useful and provides significant contribution to the field.

2. Material and methods

2.1. Experimental set-up and determination of overall heat loss coefficient

The schematic of the experimental set up is shown in Fig. 1. It consisted of two tanks, absorber set up, flow regulator, oil re-circulating pump, electrical heating system, thermocouples, etc. The Hytherm-500 oil (properties of oil given in Appendix A) was filled in the tank. The tank-A was kept at elevated position. The oil was heated with help of the electrical heaters in the tank-A and passed through the absorber at a desired flow rate and fall in the tank-B. The absorber pipe was connected with tanks with galvanized iron (GI) pipe of 19 mm ID. A flow regulator valve was fitted to control the oil flow through the absorber pipe. A small pump (50 W capacity) was used to re-circulate oil from tank-B to tank-A. An electrical heater of 2 kW capacity was placed below the tank-A to heat oil and provision was also made for an additional immersion electrical heater of 1.5 kW capacity for faster heating. A stirrer with 50 W electrical motor was used to mix the heated oil in the tank-A. There was an axial fan (50 W capacity) to flow the wind along with glass cover.