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Collector efficiency of upward-type double-pass solar air heaters with fins attached ☆

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ABSTRACT

The collector efficiency of upward-type double-pass flat plate solar air heaters with fins attached and external recycle is investigated theoretically. The double-pass device was constructed by inserting the absorbing plate into the air conduit to divide it into two channels (the upper and lower channels). The double-pass device introduced here was designed for creating a solar collector with heat transfer area double as well as the extended area of fins between the absorbing plate and heated air. Moreover, the advantage of external recycle application to solar air heaters is the enhancement of forced heat convection strength, resulting in considerable device heat transfer performance improvement. This advantage may compensate for the remixing at the inlet which decreases the heat transfer transfer-driving force decrement (temperature difference).

1. Introduction

A flat plate solar air heater differs from more conventional heat exchangers in several respects. The latter usually employ a fluid to exchange high heat transfer rates using conduction and convection. In solar air heaters, energy is transferred from a distant source of radiant energy directly into air [1,2]. The heat may then be utilized by passing air through a conduit system located between the bottom and absorbing plate. The heated air is subsequently used for space heating and drying [3,4]. In its simplest form, a flat plate solar air heater consists of one or more sheets of glass or transparent material situated above an absorbing plate with the ambient air flowing either over or under the absorbing [5], so it acts as a black body to absorb heat. The sun's rays pass through the glass and are trapped in the space between the covers and plate or absorbed into the black body. Except for the glass covers, all parts of a solar air heater are well insulated to make the energy loss as small as possible. The glass covers are employed to reduce convection and radiation losses into the atmosphere.

In addition to the essential effects of free and forced convection [6], there are many ways to achieve considerable improvement in collection efficiency by increasing the transfer area with internal fins attached [7,8], creating turbulence inside the flow channel using baffles [9,10] or designing corrugated surfaces [11,12], and enhancing the convective transfer rate [13].

Recycle-effect applications in the design and operation of equipment with external or internal reflux can effectively enhance the heat and mass transfer rate, leading to improved performance such as air lift reactors, were confirmed by many investigators [14,15], loop reactors [16], draft-tube bubble columns [17] and heat and mass exchangers [18–20], which are widely applied to absorption, fermentation, polymerization, and heat and mass transfer operations. Two conflicting effects exist in a recycle operation. The first is increasing fluid velocity, resulting in convective heat or mass transfer enhancement. The second is the decrease in driving force (temperature or concentration difference) due to remixing. Considerable improvement in collector efficiency of solar air heaters with fins attached is obtained by employing such a double-pass device due to the convective heat transfer rate increment as compensation for the driving force decrement, instead of using a single-pass device and operating at the same total flow rate. The purpose of this study is to investigate the influence of the external-recycling effect on the performance in an upward-type flat plate solar air heater with fins attached.

2. Mathematical model

The structure of an upward-type flat plate solar air heater with internal fins attached and operated with external recycle is illustrated by the schematic diagram of Figs. 1 and 2. A black absorbing plate was welded at the center of the collector with air flow channels above and below the metal absorber plate. The designed solar air heater consists of two glass covers, an absorbing plate, a recycling channel with well insulation, and a recycle device was situated at the end of upper flow channel. Before entering the upper channel, the fluid of the mass flow rate m and inlet temperature $T_{f,i}$ mix with the fluid exiting from the upper channel of the mass flow rate Rm regulated by a blower situated at the end of the upper channel. The overall heat loss coefficient, in which the edge and bottom heat losses were neglected, was estimated by an empirical correlation from an absorbing plate across a static space between two glass covers to ambient surrounding. The steady-state energy balance will be taken under the following assumptions: the

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Nomenclature

A_c	surface area of the absorbing plate, m^2
A_f	total surface area of fins, m^2
$A_{f,b}$	total cross-sectional area of fins, m^2
B	width of the air tunnel in the solar collector, m
C_p	specific heat of air at constant pressure, J/kg K
D_e	equivalent diameter of the air tunnel, m
F'	efficiency factor of the solar air heater, defined in Eq. (7)
F_R	heat-removal factor for the solar air heater
H	height of the air tunnel in the solar collector, m
h_1	convective heat transfer coefficient for fluid flowing over a flat plate, W/m^2
h_w	convective heat transfer coefficient between glass cover and ambient, W/m^2
I	improvement in collector efficiency
I_0	solar radiation incident, W/m^2
k	thermal conductivity of air, W/m^{2}K
k_s	thermal conductivity of absorbing plate, W/m^{2}K
L	collector length, m
$\dot{m}$	mass flow rate of air, kg/s
n	number of fins attached
Nu	Nusselt number
Q_u	useful gain of energy carried away by air per unit time, W
R	reflux ratio
Re	Reynolds number
T_a	ambient temperature, K
T_f	temperature distribution for the air flow, K
$T_{f,i}$	inlet air flow temperature, K
$T_{f,o}$	outlet air flow temperature, K
$T^0_{f,i}$	mixed air flow temperature at the beginning of the upper channel, defined in Eq. (18), K
$T_{f,m}$	mean air flow temperature, K
T_p	absorbing plate temperature, K
$T_{p,m}$	mean absorbing plate temperature, K
t	thickness of the fin, m
U_t	loss coefficient from the top of absorbing plate to the ambient, W/m^{2}K
u	mean air flow velocity, m/s
V	wind velocity, m/s
w_1	distance between fins, m
w_2	height of the fins, m
z	axis along the flow direction, m

Greek letters

α	absorptivity of the absorbing plate
ϵ_g	Emissivity of glass cover
ϵ_p	Emissivity of absorbing plate
η	Collector efficiency
η_f	fin efficiency
η_0	η obtained without recycle
μ	air viscosity, kg/s m
ρ	air density, kg/m^3
σ	the Stefan Boltzmann constant, W/m^{2}K^4
τ	transmittance of glass cover
ϕ	dimensionless quantity defined by Eq. (3)

Subscript

a	ambient
g	glass cover
f	fluid
i	inlet
o	outlet
p	absorbing plate

Superscript

o	mixed

temperature of the absorbing plate, bottom plate and bulk fluids are functions of the flow direction only, and both the glass covers and fluids do not absorb radiant energy. Except for the glass covers, all parts of the solar air collector outside surface and the recycling flow channel are well insulated.

2.1. Temperature distribution for the fluid in the flow direction

The steady-state energy balance for differential sections of the absorbing plate, bottom plate and fluid are, respectively.

$$I_0\tau^2\alpha = h\phi\left(T_p - T_f\right) + U_t\left(T_p - T_a\right) \tag{1}$$

$$\left[\dot{m}(1+R)C_p\right]\frac{dT_f}{dz} = h_1\phi B\left(T_p - T_f\right) \tag{2}$$

where the dimensionless quantity ϕ and fin efficiency η_f are defined with collector surface area A_c and total surface area of fins A_f as

$$\phi = 1 + \left(A_f\eta_f - A_{f,b}\right)/A_c \tag{3}$$

$$\eta_f = tanh(Mw_2)/Mw_2 \tag{4}$$

in which $M = \sqrt{h_1(2L+2t)/k_sLt}$, from Eq. (1)

$$T_p - T_f = \left[I_0\tau^2\alpha - U_t\left(T_f - T_a\right)\right]/(h_1\phi + U_t) \tag{5}$$

Substituting Eq. (5) into Eq. (2), one has

$$\frac{dT_f}{dz} + \frac{F'B\left[U_t\left(T_f - T_a\right) - I_0\tau^2\alpha\right]}{\left[\dot{m}(1+R)C_p\right]} = 0 \tag{6}$$

Where

$$F' = h_1\phi/(h_1\phi + U_t) \tag{7}$$

Eq. (6) can be easily integrated for the boundary condition

$$T_f = T^o_{f,i} \quad at \quad z = 0 \tag{8}$$

The result is

$$\frac{T_f - T_a - I_0\tau^2\alpha/U_t}{T^o_{f,i} - T_a - I_0\tau^2\alpha/U_t} = exp\left[-\frac{F'U_tBz}{\dot{m}(1+R)C_p}\right] \tag{9}$$

Eq. (9) is the temperature distribution of the bulk fluid along the flow direction. Thus, the fluid temperature at the outlet is readily obtained from Eq. (9) by substituting the condition: $T_f = T_{f,o}$ at $z = L$. The result is

$$\frac{T_{f,o} - T_a - I_0\tau^2\alpha/U_t}{T^o_{f,i} - T_a - I_0\tau^2\alpha/U_t} = exp\left[-\frac{F'U_tA_c}{\dot{m}(1+R)C_p}\right] \tag{10}$$

where $A_c = BL$, surface area of the absorbing plate.