Abstract

In this paper, an analytical steady state model is developed to study the thermal performance of an individual vacuum tube solar collector with coaxial piping (direct flow type) incorporating both single and two-phase flows. A system of equations which describe the different heat transfer mechanisms and flow conditions was established, discretised, and solved in an iterative manner. For the case of good vacuum condition ($10^{-5}$ mb) the calculated efficiency curve for single phase flow deviates significantly from the experiments with increasing collector temperature, but agrees well for the case of gas conduction inside the glass envelope at very low pressure ($\ll 1$ mb) due to the corresponding increase in overall heat loss coefficient ($U$-value).

For two-phase flow, the occurrence and propagation of flow boiling and condensation inside the collector piping under saturated condition is hypothesized. The modeling results indicate that for all-liquid-single-phase fluid flow, the collector efficiency decreases with decreasing mass flow rate. Once the fluid reaches the boiling point at a certain mass flow rate, no significant reduction in efficiency is observed anymore, which is in accordance with the experimental study.

Keywords: Vacuum tube collector; Single phase flow; Two-phase flow; Boiling and condensation; Collector efficiency

1. Introduction

For certain constant input conditions, as the mass flow rate through an individual direct flow-type vacuum tube collector in a collector panel is reduced the fluid can reach the boiling point at the corresponding pressure, which results in a two-phase fluid flow. This phenomenon is also known as “partial stagnation”. Solar collectors used in high temperature application such as solar cooling, are particularly prone to the occurrence of partial stagnation. Typically vacuum tube collectors are used. So, although there are a large number of research papers regarding the performance of solar collectors including, flat plates, vacuum tubes, heat pipes, concentrating type, etc., but here a selected literature review only related to direct flow type vacuum tube collectors is presented.

Various researchers have analyzed the thermal performance of variety of configurations of vacuum tube solar collectors to estimate the heat extracted and fluid temperature in the collector. Hsieh (1981) developed mathematical formulation to perform a thermal analysis of compound-parabolic concentrator (CPC) having a concentric evacuated double pipe as a heat receiver. Due to the use of selective surface and the vacuum between envelope and receiver, collector shows a very slight drop of efficiency at high operating temperatures. Estrada-Gasca et al. (1992) developed a steady state one-dimensional mathematical model to estimate the theoretical efficiency of an all glass evacuated tube solar collector with an internal or external absorber film deposited on the inner or outer surface of the inner glass cover, respectively. They expressed the variation of
Nomenclature

\( A_{\text{in}} \) outside surface area of the inner pipe (m\(^2\))
\( A_{\text{out}} \) outside surface area of the outer pipe (m\(^2\))
\( A_{\text{ab}} \) absorber area (m\(^2\))
\( A_{\text{ap}} \) aperture area (m\(^2\))
\( A_g \) glass cover area (m\(^2\))
\( A_x \) flow cross sectional area (m\(^2\))
\( B_0 \) boiling number (–)
\( C_{pf} \) specific heat of liquid (J/kg K)
\( C_0 \) convective number (–)
\( D_o \) outside diameter of outer pipe (mm)
\( D_i \) inside diameter of outer pipe (mm)
\( d_o \) outside diameter of inner pipe (mm)
\( d_i \) inside diameter of inner pipe (mm)
\( DH_{,\text{hp}} \) hydraulic diameter based on heated perimeter (m)
\( DH_{,\text{wp}} \) hydraulic diameter based on wetted perimeter (m)
\( F \) fin efficiency factor (–)
\( F_{\text{p-g}} \) view factor from absorber to glass (–)
\( F_c \) collector efficiency factor (–)
\( F_{\text{rf}} \) Froude number (–)
\( G \) mass velocity (kg/m\(^2\) s)
\( G_{\text{av}} \) available solar irradiance (W/m\(^2\))
\( h_{\text{r-t/b-pg}} \) radiative heat transfer coefficient from absorber to glass cover from top or bottom (W/m\(^2\) K)
\( h_{\text{r-t/b-ga}} \) radiative heat transfer coefficient from glass cover to ambient from top or bottom (W/m\(^2\) K)
\( h_{\text{c-t/b-pg}} \) convective heat transfer coefficient from absorber to glass cover from top or bottom (W/m\(^2\) K)
\( h_{\text{c-t/b-ga}} \) convective heat transfer coefficient from glass cover to ambient from top or bottom (W/m\(^2\) K)
\( h_{\text{f,i}} \) single phase all-liquid-flow heat transfer coefficient in the inner pipe (W/m\(^2\) K)
\( h_{\text{fo}} \) single phase all-liquid-flow heat transfer coefficient at the inner wall of annulus or outer wall of inner pipe (W/m\(^2\) K)
\( h_{\text{f,i,outer}} \) single phase all-liquid-flow heat transfer coefficient at the outer wall of annulus or outer wall of inner pipe (W/m\(^2\) K)
\( h_g \) enthalpy of vaporization (J/kg)
\( h_f \) enthalpy of saturated fluid (J/kg)
\( h_c \) convective heat transfer coefficient (Eq. (31)) (W/m\(^2\) K)
\( h_{\text{s,i,outer}} \) superficial heat transfer coefficient (W/m\(^2\) K)
\( h_{\text{Nc,B}} \) nucleate boiling heat transfer coefficient (W/m\(^2\) K)
\( h_{\text{p,bol}} \) boiling heat transfer coefficient (W/m\(^2\) K)
\( h_{\text{p,cond}} \) condensation heat transfer coefficient (W/m\(^2\) K)
\( k \) thermal conductivity of copper (W/m K)
\( k_f \) thermal conductivity of fluid (W/m K)
\( L_{\text{ab}} \) length of absorber sheet (mm)
\( m \) mass flow rate of fluid (kg/s)
\( M \) specific mass flow rate (kg/m\(^2\) h)
\( Nu \) Nusselt no.
\( P_{\text{gas}} \) gas pressure inside glass envelope (mb)
\( P_{\text{heated}} \) heated perimeter (m)
\( Pr_f \) Prandtl no. of liquid (–)
\( Q_{\text{f,outer}} \) heat gain of the fluid in the outer pipe (W)
\( Q_{\text{f,inner}} \) heat gain of the fluid in the inner pipe (W)
\( Q_{\text{abs}} \) heat absorbed on the absorber plate (W)
\( Q_{\text{f-net}} \) overall net heat gain of fluid (W)
\( Re \) Reynolds number (–)
\( Re_e \) equivalent Reynolds no. (Eq. (38)) (–)
\( T_{\text{fi}} \) temperature of fluid at inlet to collector (K)
\( T_{\text{fo}} \) temperature of fluid at outlet of collector (K)
\( T_{\text{rpm}} \) mean absorber plate temperature (K)
\( T_{\text{f,i,outer/inner}} \) bulk mean temperature of the fluid in the outer or inner pipe (K)
\( T_{\text{m}} \) arithmetic mean of the fluid temperature at the inlet and outlet of the collector tube (K)
\( T_{\text{a}} \) ambient temperature (K)
\( T_{\text{g,t/b}} \) glass cover temperature of the top or bottom portion above or below the absorber (K)
\( T_{\text{sat}} \) saturated temperature of fluid (K)
\( T_{\text{f,i,outer/inner-i}} \) fluid temperature at the inlet of outer or inner pipe element \( i (i = 1, 2, 3, \ldots, n) \) (K)
\( t_{\text{ab}} \) thickness of absorber sheet (mm)
\( U \) overall heat loss coefficient (W/m\(^2\) K)
\( U_t \) top heat loss coefficient (W/m\(^2\) K)
\( U_b \) bottom heat loss coefficient (W/m\(^2\) K)
\( U_o \) overall heat transfer coefficient (W/m\(^2\) K)
\( V_{\text{air}} \) velocity of air (m/s)
\( W \) width of absorber (mm)
\( x \) vapor quality (–)
\( Y \) area correction factor (Eq. (6)) (–)
\( \Delta T_m \) mean fluid temperature difference in the outer and inner pipe (K)

Greek symbols

\( \Phi \) surface heat flux (W/m\(^2\))
\( \varepsilon_g \) emissivity of the glass (–)
\( \varepsilon_p \) emissivity of the absorber surface coating (–)
\( \varepsilon_c \) emissivity of the copper (polished) (–)
\( \eta_c \) collector efficiency based on aperture area (–)
\( \varepsilon_{\text{ta}} \) effective transmittance-absorptance product (–)
\( \sigma \) Stephen Boltzmann constant (W/m\(^2\) K\(^4\))
\( \rho_f \) density of liquid (kg/m\(^3\))
\( \rho_g \) density of vapor (kg/m\(^3\))
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