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Unglazed Transpired Solar Collectors: Heat Loss Theory

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Unglazed transpired solar collectors offer a potentially low cost, high-efficiency option for once-through applications such as preheating air for ventilation, crop drying, and desiccant regeneration. This paper examines the major heat loss mechanisms associated with this concept. Radiation heat loss is determined by considering losses to both the sky and the ground. Convective heat losses are obtained by integrating the product of the temperature and velocity profiles in the boundary layer at the downwind edge of the collector. This convective heat loss is then expressed in terms of the thermal equivalent length of irradiated absorber, and analysis shows that this loss can be very low for large collectors even under windy conditions. These results are incorporated into a simple computer model which predicts collector efficiency as a function of suction velocity, wind speed, ambient temperature, and radiation. Remaining research issues are discussed.

1 Introduction

Unglazed transpired collectors can be used to heat ambient air in once-through solar energy systems. With this type of unglazed collector, air adjacent to the front surface of the absorber is drawn through the perforated absorber so that heat that would otherwise be lost by convection from the absorber is captured by the air flow into the collector (see Fig. 1). The energy in the thermal boundary layer, however, is lost over the edge of the collector. This type of design shows promise for applications such as ventilation preheat, crop drying, and desiccant regeneration. A German patent (Wieneke, 1981) describes an unglazed perforated roof absorber for heating ventilation air. Schulz (1988) describes a fabric absorber used in Germany for crop drying. A U.S./Canadian company is currently manufacturing and marketing unglazed perforated walls for ventilation preheat.

Operating parameters for an unglazed transpired collector will depend on the application. Typical face velocities might range from 0.01 m/s for desiccant regeneration to 0.05 m/s for preheating ventilation air. Example calculations given in the text will be for the ventilation preheat application.

A number of studies have been done on glazed solar collectors utilizing transpired absorbers to heat recirculated air. The primary reason for the use of transpiration was to increase the heat-transfer coefficient between the absorber and the air stream. This paper describes the theory of potentially low-cost, high-efficiency unglazed collectors utilizing a thin transpired absorber in once-through air heating systems. Our objective is to describe the heat losses associated with these designs with special attention paid to the potential effects of wind losses.

Fluid dynamics for boundary layer flow parallel to porous surfaces has been studied for aerodynamic applications such as the use of suction on airplane wings to reduce drag. Heat transfer issues have been addressed in conjunction with injection cooling

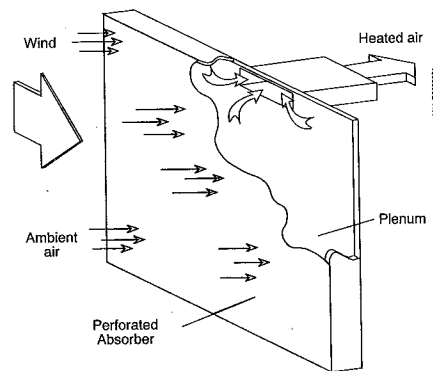


Fig. 1 Unglazed transpired solar collector oriented vertically for building ventilation preheat. Intake air is drawn by the building ventilation fan through the perforated absorber plate and up the plenum between the absorber and the south wall of the building.

for turbine blades and rocket nozzles. Heat transfer with suction, which is the case for an unglazed transpired collector, has not received much attention.

In this paper, we: (1) present an overall heat balance for an unglazed transpired collector; (2) estimate the radiative heat loss term; (3) review the theory on the effects of suction on boundary layer flow and heat transfer for natural and forced convection under laminar and turbulent conditions, and apply this theory to estimate the convective heat loss term; (4) incorporate the radiative and convective loss terms into a simple model for predicting collector thermal performance; and (5) discuss remaining research issues.

2 Overall Heat Balance

The overall heat balance on an unglazed transpired collector is

$$\rho c_p v_o A_c (T_{out} - T_{amb}) = I_c A_c \alpha_c - Q_{rad} - Q_{conv} \quad (1)$$

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The left-hand side of this equation represents the useful energy collected. The first term on the right-hand side is the solar energy absorbed by the absorber and is straightforward to calculate. Note that I_c is the total radiation striking the absorber including direct, diffuse, and reflected. The second and third terms are, respectively, the losses to the environment via radiation and convection. Our focus will be estimating these losses with special attention paid to the convection term since wind loss for an unglazed collector is usually a major concern.

2.1 Radiation Heat Loss. Radiation loss occurs both to the sky and to the ground with the view factors depending on the tilt of the absorber. (We will assume that the wall behind the collector plenum is adiabatic and at a temperature close to the absorber temperature so that radiation loss to this wall is negligible.) Assuming the absorber is gray and diffuse, the radiant heat loss is

$$Q_{\text{rad}} = \epsilon_c \sigma A_c (T_{\text{coll}}^4 - F_{cs} T_{\text{sky}}^4 - F_{cg} T_{\text{gnd}}^4). \quad (2)$$

2.2 Convection Heat Loss. Convective heat loss from the surface will depend to a certain extent on the nature of the surface. A surface with homogeneous suction (versus suction at discrete holes or slots) has the simplest wall boundary condition and can be analyzed in a straightforward manner. We will thus analyze this type of surface and later discuss how the results might be extrapolated to a surface with discrete suction. There is a considerable amount of information in the literature on velocity profiles and, to a lesser extent, temperature profiles for homogeneous suction surfaces. We will review this work and apply it to the problem of convective heat losses for collector applications. Our analysis will be further simplified by considering the local free-stream flow to be parallel to the surface.

3 Laminar Forced Convection

3.1 Velocity Profile. For a nonporous plate subjected to a laminar parallel wind, the velocity boundary layer grows as $x^{1/2}$ (see Fig. 2). The Reynolds number also increases as $x^{1/2}$, and transition to turbulent flow will eventually occur. Once the boundary layer becomes turbulent it grows much more rapidly, increasing as $x^{4/5}$ instead of $x^{1/2}$ (Bejan, 1984).

The effect of homogeneous wall suction on the velocity and thermal boundary layers is shown in Fig. 3. Using scaling arguments, one can argue that for sufficiently large x , $\delta u/\delta x$ goes to zero (since the denominator becomes large). Thus, $u = u(y)$ only and, by continuity, $\delta v/\delta y = 0$. Since $v = -v_0$ everywhere along the wall, we must have for sufficiently large x that $v = -v_0$ everywhere in the flow field. The x -momentum equation becomes linear:

$$-v_0 \frac{du}{dy} = \nu \frac{d^2 u}{dy^2}. \quad (3)$$

This equation is directly integrable. Integrating it twice and applying the boundary conditions that $u = 0$ at the wall and $u = U_\infty$ at infinity, one obtains the following solution:

$$u = U_\infty \left(1 - e^{-\frac{v_0 y}{\nu}}\right). \quad (4)$$

Note that this is an exact solution to the Navier-Stokes equation, and boundary layer approximations are not needed.

3.2 Boundary Layer Thickness. Often the boundary layer thickness is defined as that value of y for which the horizontal velocity is 99 percent of the free stream value, and this is usually written as δ_{99} . The selection of 99 percent is completely arbitrary, however, and for reasons which will we will explain later, it is more convenient for us to use a definition based on 86 percent of the free-stream value. Putting δ_{86} in place of y in the above equation and setting $u = .86U_\infty$ we can solve for δ_{86} to obtain:

$$\delta_{86} = 2.0 \frac{\nu}{v_0} \quad (5)$$

Unlike the no-suction profile, the boundary layer thickness for suction is a constant independent of distance along the plate, and it does not depend on the free-stream velocity. This solution is not valid for the starting length in which du/dx and dv/dy are not zero, but is approached asymptotically and so is called the asymptotic solution. *For a typical ventilation suction velocity of .05 m/s, the velocity boundary layer thickness is only 0.6 mm.*

As with the case of the no-suction profile, the asymptotic suction will be stable only for Reynolds numbers (based on boundary layer thickness) below a certain critical value. Schlichting (1979) points out that suction provides a great deal of stability to flow over a flat plate, raising the critical Reynolds number by a factor

Nomenclature

A_c = collector area (m ²)	u = velocity parallel to the wall (m/s)
c_p = specific heat at constant pressure (J/kg °C)	U_∞ = free stream velocity (m/s)
C_f = friction factor	v = velocity normal to the wall (m/s)
F_{cg} = collector-to-ground view factor	v_0 = suction velocity (m/s)
F_{cs} = collector-to-sky view factor	W = collector width (m)
g = acceleration of gravity (m/s ²)	x = distance along the wall (m)
I_c = solar insolation incident on the collector (W/m ²)	y = distance from the wall (m)
h = convective heat-transfer coefficient (W/m ² °C)	Nu = hx/k , Nusselt number
k = thermal conductivity (W/m °C)	Pr = ν/α , Prandtl number
L_e = equivalent convective heat loss length (m)	Re = $U_\infty vx/\nu$, Reynolds number
L_s = boundary layer starting length (m)	St = $h/U_\infty \rho c$, Stanton number
q'' = net wall heat flux: insolation-infrared losses (W/m ²)	α_c = collector absorptance
Q_{conv} = collector convective heat loss (W)	α = thermal diffusivity (m ² /s)
Q_{rad} = collector radiant heat loss (W)	β = thermal expansion coefficient (1/K)
T_{amb} = ambient temperature [= T_∞] (°C)	Δ_{86} = thermal boundary layer thickness (m)
T_{coll} = collector temperature [= T_0] (°C)	δ_{86} = velocity boundary layer thickness (m)
T_{out} = collector output temperature (°C)	ϵ_c = absorber surface emissivity
T_0 = wall temperature [= T_{coll}] (°C)	ϵ_{hx} = absorber heat exchange effectiveness
T_∞ = free stream temperature [= T_{amb}] (°C)	η = collector efficiency
T = local mean temperature in turbulent boundary layer (°C)	ν = kinematic viscosity (m ² /s)
	ρ = density (kg/m ³)

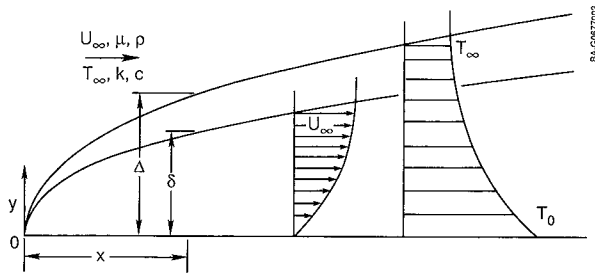


Fig. 2 Velocity and thermal boundary layers for laminar forced convection over a flat-plate without suction

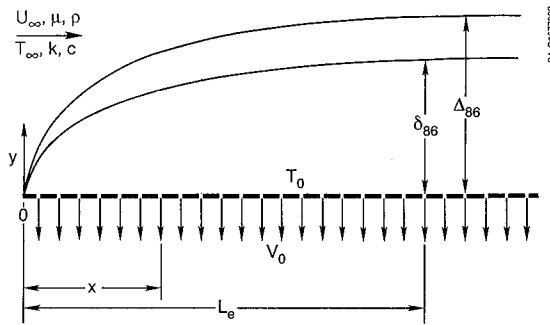


Fig. 3 Velocity and thermal boundary layers for laminar forced convection over a flat-plate with suction

of 130. Because the boundary layer thickness is constant in the asymptotic region, the Reynolds number based on boundary layer thickness will no longer increase. If the flow is laminar and the Reynolds number is below 70,000, transition to turbulence on a smooth plate with uniform suction will not occur no matter how long the plate.

Schlichting also shows that when the starting length is taken into account, the minimum suction velocity for stability is

$$v_0 = 1.2 \times 10^{-4} U_{\infty} \quad (6)$$

For practical wind velocities, this is much less than the suction velocities expected for typical unglazed transpired collector applications.

3.3 Starting Length. The above results for an asymptotic boundary layer will only apply after a certain starting length. Arpaci and Larsen (1984) give an integral solution from which we can obtain this length. By substituting a parabolic velocity profile which meets the boundary conditions into the integral boundary layer momentum equation, they find that the asymptotic thickness consistent with the assumed profile must be $2\nu/v_0$, or δ_{86} . (This is why we chose the 86 percent definition earlier rather than the more typical 99 percent value.) Writing the differential equation in terms of δ_{86} , and integrating across the boundary layer, Arpaci and Larsen obtain

$$-\ln\left(1 - \frac{\delta}{\delta_{86}}\right) - \frac{\delta}{\delta_{86}} = \frac{15}{2} \frac{v_0 x}{U_{\infty} \delta_{86}} \quad (7)$$

If we define a starting length L_s to be that distance along the plate at which the velocity boundary layer thickness is 99 percent of δ_{86} , we can obtain the following starting length from the integral solution:

$$L_s = .96 \frac{U_{\infty} \nu}{v_0^2} \approx \frac{U_{\infty} \nu}{v_0^2} \quad (8)$$

As we expect, the starting length becomes infinite as the suction velocity goes to zero. It is worthwhile to note here that,

based on a series solution of Iglisch (1949), Schlichting (1979) reports a starting length of $4U_{\infty} \nu/v_0^2$. However, Maddaeus and Shanebrook (1983) point out that a number of experimental studies contradict Iglisch and show starting lengths to be in the range of $0.5 U_{\infty} \nu/v_0^2$ to $U_{\infty} \nu/v_0^2$. Thus, our approximate result for a 99 percent starting length seems reasonable.

Assuming a maximum wind speed of 10 m/s, a typical ventilation suction velocity of .05 m/s, and an average temperature of 30°C for which $\nu = 15.7 \times 10^{-6} \text{ m}^2/\text{s}$, the starting length is 6 cm.

3.4 Temperature Profile. We next consider the energy equation. Analogous to the case of the momentum equation, for constant suction velocity there exists at sufficiently large x an asymptotic region where heat convected through the porous plate by suction is exactly balanced by heat conducted from the plate out into the fluid. In this region, the energy equation simplifies and becomes linear:

$$-v_0 \frac{\partial T}{\partial y} = \alpha \frac{\partial^2 T}{\partial y^2} \quad (9)$$

For a constant heat flux plate, the boundary conditions are as follows:

$$\text{At } y = 0: \quad dT/dy = -q''/k$$

$$\text{At } y = \infty: \quad T = T_{\infty}$$

where the heat flux q'' represents the net radiant heat flux, which in the case of a transpired absorber is the difference between the absorbed solar flux and the radiation heat loss. Integrating and applying the boundary conditions we obtain

$$T = T_{\infty} + \frac{q''}{\rho c_p v_0} e^{-\frac{\rho c_p v_0}{k} y} \quad (10)$$

Setting $y = 0$ and rearranging terms, yields

$$\rho c_p v_0 (T_0 - T_{\infty}) = q'' \quad (11)$$

Since T_{coll} equals T_0 for a homogeneous suction plate, Eq. (11) states that in the asymptotic region all of the net heat flux from the wall goes into the suction fluid. Note that because the wall temperature in the asymptotic region of a constant heat flux wall is constant, we would get exactly the same result for an isothermal wall. This can be shown by using the isothermal boundary condition (at $y = 0$, $T = T_0$) in place of the wall heat flux boundary condition and solving as above. Defining Δ_{86} as the value of y at which $(T - T_0) = .86(T_{\infty} - T_0)$ and combining Eq. (10) and (11) we obtain

$$\Delta_{86} = 2.0 \frac{\alpha}{v_0} = \frac{\delta_{86}}{\text{Pr}} \quad (12)$$

So the asymptotic thermal boundary layer thickness is constant and is thicker than the velocity boundary layer thickness by a factor of $1/\text{Pr} = 1.4$ for air.

3.5 Convection Heat Loss. We now apply these results to an unglazed transpired solar collector. In the asymptotic region the thermal boundary layer thickness is constant, because there is no net flux of heat into the boundary layer. All the heat conducted into the boundary layer is removed convectively by the suction air. In the starting length, on the other hand, the velocity and thermal boundary layers grow because there is a net flux of momentum and heat into the free stream. The total amount of heat lost from the plate into the boundary layer along the starting length will be the same as the heat carried off by the air flowing off the far end of the plate, as can be seen by a simple energy balance on the boundary layer. To determine the amount of this heat loss we can either integrate (over x) the net heat flux over the starting length or integrate (over y) the heat flux leaving the far end of the plate.

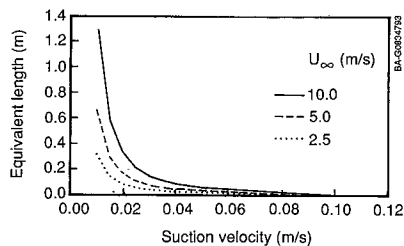


Fig. 4 Calculated equivalent convection heat loss length versus suction velocity at various wind speeds

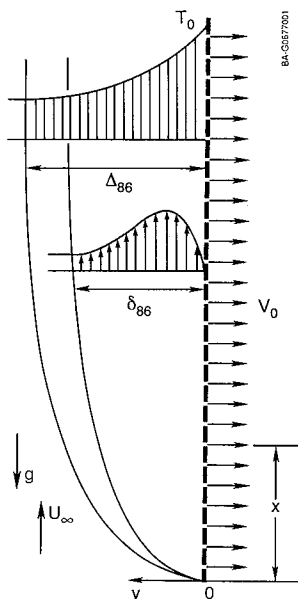


Fig. 5 Velocity and thermal boundary layers for laminar free convection over a flat-plate with suction

Since we know the velocity and temperature profiles in the asymptotic region, performing the second integration is straightforward. The energy per unit plate width leaving the far end of the plate is

$$\frac{Q_{\text{conv}}}{W} = \int_0^{\infty} \rho c_p u(y) [T(y) - T_{\infty}] dy. \quad (13)$$

We define an equivalent convection heat loss length, L_e , to be the length of plate that multiplied by the net radiant heat flux equals the convective heat losses per unit width off the end of the plate. Substituting Eq. (4) and (10) into Eq. (13) and performing the integration we obtain

$$L_e = \frac{Q_{\text{conv}}}{W q''} = \frac{U_{\infty}}{v_0^2} \left[\frac{\nu}{\text{Pr} + \text{Pr}^2} \right], \quad (14)$$

or

$$L_e \approx \frac{L_s}{\text{Pr} + \text{Pr}^2}. \quad (15)$$

Figure 4 shows equivalent convective loss lengths for air flow at various wind speeds and suction velocities.

For a wind speed of 10 m/s, a suction velocity of .05 m/s, and an average temperature of 30°C for which $\text{Pr} = .71$, the equivalent length would be only 5 cm. Thus the convective heat losses are equivalent to the net solar energy on a 5-cm wide strip of collector and would be negligible for a large collector.

4 Laminar Free Convection

4.1 Velocity Profile. Up to this point we have only discussed forced convection. Heat loss will also occur by natural

convection, and one might expect that this could be an especially important term when there is no wind. In this section, we give the theory for natural convection for a vertical collector (see Fig. 5). The governing equations are the same as before, but now the x-momentum equation contains a buoyancy term.

A scale analysis (Bejan 1984) indicates that the thermal and velocity boundary layers for a nonporous wall grow as $x^{1/5}$ for a constant heat flux wall. As was the case for forced convection, transition to turbulence will eventually occur, in this case when the Rayleigh number exceeds a value of about 10^9 .

In the case of suction, we follow exactly the same reasoning as for the forced convection case. At a sufficiently great distance up the wall, we expect an asymptotic solution in which u is a function of y only and $v = -v_0$ everywhere. The x-momentum equation is linear:

$$-v_0 \frac{du}{dy} = \beta g (T - T_{\infty}) + \nu \frac{d^2 u}{dy^2}. \quad (16)$$

The boundary conditions are as follows :

At $y = 0$: $u = 0$, $dT/dy = -q''/k$ (constant heat flux wall)

At $y = \infty$: $u = 0$, $T = T_{\infty}$.

The energy equation and its boundary conditions are exactly the same as for the forced convection case. Thus the temperature profile is the same as for forced convection, and the wall temperature is again constant in the asymptotic region. Because the temperature profile is the same as for the forced convection case, so is the thermal boundary layer thickness.

To obtain the velocity profile we substitute the temperature profile into the x-momentum equation and integrate it to obtain (Arpaci and Larsen, 1984):

$$u = \frac{\beta g \alpha^2 q''}{v_0^3 k (\text{Pr} - 1)} \left(e^{-\frac{1}{\text{Pr}} \frac{v_0 y}{\alpha}} - e^{-\frac{v_0 y}{\alpha}} \right). \quad (17)$$

The velocity boundary layer thickness is constant in the asymptotic region (although it cannot be defined in the same way as in the forced convection case). We can determine the distance from the wall at which the maximum velocity occurs by differentiating Eq. (17) and setting it to zero, thus obtaining

$$y = \frac{\alpha}{v_0} \frac{\text{Pr} \ln(\text{Pr})}{\text{Pr} - 1}. \quad (18)$$

Substituting Eq. (18) back into (17), we obtain for the maximum velocity

$$u_{\text{max}} = \frac{\beta g \alpha^2 q''}{v_0^3 k (\text{Pr} - 1)} \left(e^{-\frac{\ln(\text{Pr})}{\text{Pr} - 1}} - e^{-\frac{\text{Pr} \ln(\text{Pr})}{\text{Pr} - 1}} \right). \quad (19)$$

For air at 30°C, a suction velocity of 0.05 m/s, and a heat flux of 1000 W/m², the maximum free convection velocity is only 0.002 m/s and occurs at a distance of 0.4 mm from the wall.

4.2 Convection Heat Loss. Using the known solutions for the velocity and temperature profiles in the asymptotic region, we can now integrate to determine the convective heat loss as we did for the forced convection case and obtain the equivalent convection heat loss length as

$$L_e = \frac{Q_{\text{conv}}}{W q''} = \frac{\beta g \alpha^3 q''}{v_0^5 k (\text{Pr} - 1)} \left(\frac{\text{Pr}}{\text{Pr} + 1} - \frac{1}{2} \right). \quad (20)$$

For a suction velocity of $v_0 = 0.005$ m/s, a heat flux of 1000 W/m², and properties of air at 30°C, the thermal equivalent plate length for convective heat loss is 1.3×10^{-5} m. Thus for typical suction rates that would be used for ventilation preheat, end heat loss due to natural convection is negligible. This is because even though the asymptotic thermal boundary layer thickness and

temperature profile are the same as for the forced convection case, the velocities caused by natural convection are much lower than for typical forced convection conditions as shown in the previous section. This greatly limits the heat loss over the wall edge.

5 Turbulent Forced Convection

5.1 Wall Turbulence. In the application of a transpired solar collector, the flow may be turbulent due to imperfect wall conditions (edge effects, discrete suction holes, surface roughness, corrugations, etc.)

Considerable research on transpired flat plates with wall turbulence was done at Stanford (Moffat and Kays, 1984). Their experiments looked at the effects of pressure gradient, roughness, and surface curvature on heat transfer from a constant temperature wall with suction or injection. By writing the momentum and energy boundary layer equations in integral form (which is valid for both laminar and turbulent boundary layers), they show that in the asymptotic region the skin friction coefficient and Stanton number become simply

$$\frac{C_f}{2} = St = \frac{v_0}{U_\infty} \quad (21)$$

The latter part of this equation is equivalent to our earlier statement that all of the net wall heat flux goes into the suction fluid. Thus an asymptotic boundary layer has been reached when the above conditions are met.

Distinctions between laminar and turbulent asymptotic layers are described by Dutton (1958). Dutton tested both nylon fabric and a perforated plate and found that the turbulent boundary layer reaches a constant thickness for suction ratios, v_0/U_∞ , greater than .0073, although it appears from his graphs that an asymptotic turbulent state might have eventually been reached at greater distances down the plate for lower suction rates. Dutton found that the inner part of the turbulent boundary layer resembled the laminar asymptotic profile, and, when v_0/u_∞ reached .01, the entire boundary layer achieved the laminar asymptotic shape. Moffat and Kays (1984) indicate that for suction rates of v_0/u_∞ greater than .004, the turbulent boundary layer reverts to an asymptotic laminar boundary layer.

5.2 Convective Heat Loss. As the case with any turbulent flow problem, we will have to rely on empirical data. Verrolet, et al. (1972) report experimental velocity and temperature profiles for a range of suction rates. Unfortunately, the highest ratio of suction to free stream velocity tested was only 0.003. (For a typical ventilation application, this ratio would be 0.020 for a 2.5 m/s wind.) Even for this suction ratio, the thermal boundary layer had not quite become asymptotic at the point where temperatures were measured. Nevertheless, to get a rough estimate of convective heat losses we used these experimental profiles in numerically performing the same type of integration as for the laminar cases. The result showed convective heat loss for the turbulent boundary layer about an order of magnitude larger than for the laminar asymptotic boundary layer. At our higher suction ratios, we would expect this difference to be considerably less. Clearly, data covering the solar collector operational regime is needed to provide a better estimation of convective heat loss.

The preceding discussion applies to forced convection. Although discrete suction sites or surface roughness could theoretically cause turbulence in a transpired free-convection situation, the velocities along the wall and convective heat losses are so small for the laminar case that we expect the larger heat losses for the turbulent case to also be negligible. In fact, for the air temperature rise typical of a ventilation preheat application, it can be shown that even a nontranspired collector would have a small free-convection heat loss. In any case, we have not identified any empirical velocity and temperature profile data in the literature for turbulent asymptotic free-convection boundary layers with either

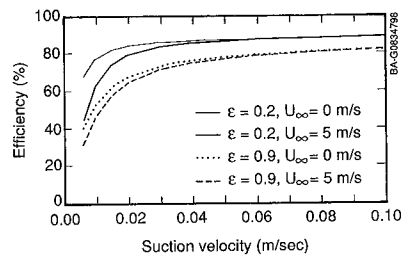


Fig. 6(a)

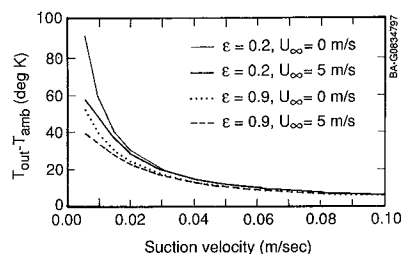


Fig. 6(b)

Fig. 6 Predicted thermal performance of a vertical unglazed transpired solar collector as a function of suction velocity, absorber emissivity, and wind speed. Collector size = 3 m × 3 m. $T_{amb} = 10^\circ\text{C}$, $T_{sky} = T_{amb} - 15^\circ\text{C}$, $T_{gnd} = T_{amb}$, and $I_c = 700 \text{ W/m}^2$.

homogeneous or discrete suction. Such data could be useful for studying transpired collectors working at higher temperatures.

6 Collector Efficiency

6.1 Predictive Model. Based on the previous analysis, we will assume that natural convection heat losses are negligible. For forced convection with high suction ratios we can assume an asymptotic laminar boundary layer. Hence, from Eq. (11) and (14) we can express the convective heat loss term as

$$Q_{conv} = 0.82 \left(\frac{U_\infty v_0}{v_0^2} \right) W [\rho c_p v_0 (T_{coll} - T_{amb})] \quad (22)$$

Having determined approximate expressions for the convective and radiative heat losses, we can now complete our solution for the collector efficiency. Substituting the relationships for Q_{rad} and Q_{conv} into Eq. (1) we are left with one equation in one unknown, T_{coll} . This nonlinear equation is iterated to find T_{coll} . Once T_{coll} is known, the collector efficiency is given as

$$\eta = \frac{\rho c_p v_0 (T_{coll} - T_{amb})}{I_c} \quad (23)$$

Tests of both small and full-scale absorbers at NREL give efficiencies that are in good agreement with our model. A detailed description of these results will be presented in forthcoming papers.

6.2 Model Results. The simple model allows us to investigate various performance sensitivities. Figure 6 shows predicted thermal performance for a vertical unglazed transpired solar collector as a function of suction velocity for wind speeds of 0 and 5 m/s and absorber emissivities of 0.9 and 0.2. The efficiency of the unglazed transpired solar collector is influenced by the fact that radiation losses are directly to ambient but there is no glazing optical penalty. For suction velocities greater than 0.05 m/s, efficiencies are nearly constant and independent of wind-speed. As suction velocity decreases, the effect of wind speed on collector efficiency increases, especially for the low emissivity absorber. The benefits due to the low emissivity absorber generally increase as the suction velocity decreases.

For once-through applications, the collector inlet temperature equals the ambient temperature. Therefore, $T_{out} - T_{amb}$ indicates the collector temperature rise and, indirectly, the delivered air temperature. Except at lower suction velocities, the effect of absorber emissivity on efficiency is more important than the effect on delivered air temperature.

For a typical ventilation preheat suction velocity of 0.05 m/s, the collector temperature rise is approximately 12°C regardless of absorber emissivity, and efficiencies are approximately 78 percent and 84 percent for absorber emissivities of 0.9 and 0.2, respectively.

Because of the importance of radiation heat loss, as the ambient temperature drops so does the surface temperature, and thus efficiency increases with decreasing ambient temperature. This effect is of course true for any solar collector, but is especially so for this one. Chau and Henderson (1977) and others have noted the magnitude of this effect for matrix absorbers.

7 Issues and Discussion

7.1 Pressure Drop Considerations. For a transpired collector we need a sufficiently high pressure drop across the absorber to provide reasonably uniform flow. Typical rules of thumb used for header sizing would dictate a ΔP across a transpired wall on the order of ten times the ΔP in the plenum behind the wall. Also, researchers in the area of boundary layer control for wings noted the importance of maintaining sufficient pressure drop across the surface to prevent any localized outflow. To prevent any local outflow on a transpired wall, the ΔP must also be high enough to overcome local negative pressure coefficients.

For wall applications the effect of flow around the building is important. Low pressures will occur on the leeward face of a building and anywhere where the flow separates or velocities are high such as at edges and corners. Davenport and Lui (1982) report mean local pressure coefficients on a smooth building wall as low as -1.2 while Murakami (1990) reports local values as low as -2.0 . In addition Arens and Williams (1977) point out that local pressures can drop much further due to the effects of surface irregularities such as casements. In the case of a transpired wall, if corrugations are used in the absorber to provide structural rigidity, local outflow could occur if the wall does not have a sufficient ΔP to overcome the low local pressures just downwind of ridges.

For a perforated metal plate, the pressure drop is roughly proportional to the square of the velocity, and the open area ratio is the most important design parameter (Gregory, 1961). Open area ratios of about one percent should provide sufficient pressure drop. Clearly, however, we do not want to use a higher ΔP than absolutely needed or we would waste fan power. *For a one percent open area, in a transpired wall, at a flow rate of 0.05 m/s, a ΔP of 50 Pa would provide positive suction over an entire smooth wall in wind speeds as high as 10 m/s. Assuming a fan efficiency of 20 percent this corresponds to a power requirement of 16 W/m². For a wall operating at an efficiency of 80 percent with an average solar flux of 500 W/m², the fan power represents about 4 percent of the average energy collected.* Thus it appears that pressure drop requirements are reasonable; however, testing is needed to determine the extent to which fan power can be minimized.

7.2 Extension to Discrete Suction. As stated earlier, our theory is valid for perfectly homogeneous suction. According to Wuest (1968) suction can be considered homogeneous if the spacing between pores is less than the boundary layer thickness. Aeronautics researchers (Lachmann, 1961) found that suction through discrete holes can actually cause transition to turbulence by bending vorticity lines which run parallel to the plate and perpendicular to the flow. The resulting vortex stretching and bending results in streamwise vorticity which can precipitate transition to turbulence. The asymptotic boundary layer will still have a constant

average thickness whether it is laminar or turbulent, but its local thickness will vary at each point depending on the point's distance from suction sites. Thus a perforated plate can be expected to have an asymptotic boundary layer with a dimpled shape.

Dutton (1960) found that for a perforated plate, when the suction rate was sufficient to produce a laminar asymptotic layer, the average thickness of this layer was the same as one would expect if the same total mass influx were uniformly distributed. In addition, he obtained approximately the same boundary layer thicknesses for a nylon fabric surface as for a perforated plate. This data was for higher free-stream velocities and closer hole spacings than might be encountered for transpired absorber applications, however. Recent experimental results at NREL indicate that the heat loss theory applies well to perforated plates, and these results will be included in a future paper.

7.3 Absorber Heat Exchange Effectiveness. Because the radiation heat loss is based on surface temperature, T_{coll} , we need to relate this to the outlet temperature. For an absorber with homogeneous suction, these temperatures are the same. Surfaces such as a porous concrete or a finely woven fabric will approach this situation. However, for a surface such as a perforated metal plate, the outlet air temperature will be below the surface temperature. We can relate T_{out} and T_{coll} via a heat exchange effectiveness:

$$\epsilon_{HX} = \frac{T_{out} - T_{amb}}{T_{coll} - T_{amb}} \quad (24)$$

The heat exchange effectiveness will depend on the overall heat-transfer coefficient for the air passing through the absorber. A number of studies of heat transfer to air flowing through perforated plates have been done in conjunction with compact heat exchanger design, but these are generally for a stacked array of very porous plates. Sparrow and Ortiz (1982) experimentally studied air flow normal to a perforated plate and addressed front surface heat transfer. However, the naphthalene sublimation technique they used neglected heat transfer at hole edges, and they looked at considerably higher Reynolds numbers and porosities than would be typical for a transpired absorber. Because the data included only two pitch-to-diameter ratios (2.0 and 2.5), it does not appear reasonable to extrapolate the results to our case in which pitch-to-diameter ratios are closer to 10.

In order to adequately optimize hole size and spacing, we have begun a program to experimentally and numerically determine overall heat transfer correlations for low Reynolds numbers and a wide range of pitch-to-diameter ratios and will report on results shortly.

7.4 Three-Dimensional and Nonparallel Flow. Our derivation for convective heat losses assumes that the boundary layer is parallel to the absorber and convects heat over only one edge. Sparrow, Ramsey, and Mass (1979) show that for wind impinging on the collector at large attack angles, a stagnation area will be created and three-dimensional flow can occur off many edges thereby increasing the loss above what we have assumed. Thus the above value must be multiplied by a correction which will depend on wind direction. More research is needed to determine the magnitude of this correction. However, for large walls, we expect the wind to be directed parallel to the wall over most of its surface.

7.5 Free-Stream Turbulence. An additional issue not addressed by our basic theory is the impact of free-stream turbulence. With the high mean velocities encountered for airplane wings, the turbulent velocity fluctuations of the atmosphere are usually neglected in studies of wing boundary layer control. However, in tests of conventional flat-plate collectors in actual wind conditions, some researchers have found that heat loss coefficients can increase several fold due to the free-stream turbulence in the wind.

Test, Lessman, and Johary (1981) found that in outdoor testing of a flat plate with an attack angle to the wind of 40 deg, the curve of heat loss coefficient versus distance along the plate was similar in shape to what one would expect for a laminar boundary layer, but the heat loss coefficient was about twice as large. The higher values than one would expect based on low-turbulence wind tunnel tests corresponded to the higher turbulence intensities experienced outdoors. Significantly, free-stream turbulence increased the heat loss without causing transition of the laminar boundary layer.

In a later study, McCormick, Test, and Lessmann (1984) created turbulence intensities higher than typical outdoor values by using a set of horizontal slats in a wind tunnel. They found that the turbulence increased the velocity gradient at the wall and thickened the boundary layer. At the highest turbulence intensities (on the order of 30 percent) the boundary layer behaved as if it were turbulent, but the velocity profiles differed from what one expects from a classic turbulent boundary layer resulting from wall friction. Free-stream turbulence could increase the heat loss from a transpired wall as it does for a nontranspired wall. However, initial outdoor test results for our small collector with a fabric absorber have not yet indicated that this is a significant problem.

8 Conclusions

The basic theory developed in this paper indicates that for unglazed transpired solar collectors, heat losses due to natural convection are negligible, and those due to wind should be small for large collectors operated at typical suction velocities. The theory is based on parallel laminar external flow and a homogeneous suction surface; further work is underway to extend it to less ideal circumstances.

A simple performance model accounting for radiation and convection losses has been developed. The model indicates that high efficiencies can be obtained for typical ventilation preheat flow rates and modest collector temperature rises. At lower flow rates, collector temperature rises are higher, but efficiencies are lower and wind effects are more important. Selective surface absorbers would be useful in achieving higher collector temperature rise.

9 Research Needs

More research is needed to determine the effects of both wall and free-stream turbulence as well as nonparallel flow on the nature of the thermal boundary layer on a transpired, constant heat flux surface. Because of the effect of absorber surface temperature on radiation heat loss, more work is also required to determine the heat exchange effectiveness for a heated, perforated plate. For building wall applications, more knowledge is needed of local pressure coefficients to determine the minimum flow rate needed to prevent outflow under various wind conditions. The National Renewable Energy Laboratory is currently engaged in a program to investigate these issues experimentally and numerically. This work is not only aimed at large ventilation walls but at exploring the lower limits of collector size and upper limits of delivered temperatures for which the concept of transpiration without glazing makes sense.

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