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Performance of Surface and Volumetric Solar Thermal Absorbers

Thermal surface absorbers convert all incident radiation to heat at a single local temperature. The fluid flows perpendicular to the radiation's propagation direction. In contrast, in volumetric absorbers the fluid flows parallel to the radiation's propagation direction, and the absorber may exhibit temperature stratification along the radiation's direction. This raises the question whether reabsorption of parts of the thermal emission coming from the hotter absorber sections renders the volumetric absorber superior to the surface absorber. For the case of isotropic radiation, we compare the efficiency of the volumetric and the surface absorber with each other and with an isothermal absorber. We find that the nonselective volumetric absorber is less efficient than the nonselective surface absorber for the assumption of perfect heat transfer between absorber and fluid. Thus we conclude that in practical high-flux applications, the superiority of volumetric absorbers stems from the enhanced heat transfer area. If the fluid flows against the direction of radiation propagation the volumetric absorber is slightly more efficient than if it flows with this direction. We also discuss the effect of the two-flux approximation on simulation of isotropic volumetric absorbers.

1 Introduction

For high temperature, high concentration solar thermal systems it is important to minimize the dominant radiative losses. For an isothermal system, e.g., a well-mixed cavity, all the absorber necessarily radiates according to the fluid's exit temperature, and the magnitude of radiative losses is mainly determined by the absorber material's spectral properties, i.e., the absorber's selectivity. For systems with a large difference between inlet and outlet temperature, e.g., for systems with secondary processes having large temperature spread or for open systems using the inlet fluid at ambient temperature, one can try to bring the effective temperature of the absorber's reradiation as near as possible to the inlet temperature. This could be performed utilizing a surface absorber where different parts of the aperture area are at different temperatures (Bejan, 1988; Ries et al., 1995), or by employing a volumetric absorber where the temperature increases along the depth of the absorber from inlet to outlet temperature. These two absorber concepts are depicted in Fig. 1.

For a *surface absorber,* the heat transfer fluid flows parallel to the aperture, i.e., perpendicular to the direction of the radiative flux. If considered as a heat exchanger, the absorber may be said to work in "cross-current" operation. Generally the heat is not directly absorbed by the fluid, but by an absorber surface; a sufficiently high heat transfer rate through the absorber wall as well as between the absorber and the fluid is required. If the heat conduction in the directions of the absorber surface can be neglected, the fluid sequentially interacts with the different areas on the absorber. This is called a "nonisothermal" absorber (Bejan, 1988; Ries et al., 1995). If this type of heat conduction is dominant, an "isothermal" absorber results.

In a *volumetric absorber* the problem of heat transfer between the absorber and the fluid is solved by increasing the heat transfer area beyond the aperture surface. Rather than being absorbed at the surface, the radiation penetrates into the volume of the absorber and is gradually absorbed. The fluid generally flows normal to the aperture, either parallel to the radiation's direction, i.e., in "co-current" operation, or antiparallel to the radiation's direction, i.e., in "counter-current" operation (see Fig. 1). For low thermal radiation losses, the effect of heat conduction in the flow direction should be small. If it is dominant, the absorber is again isothermal.

Several simplifying assumptions are made throughout this work. It is assumed that the fluid's temperature equals the absorber's temperature. This may be closely approached by a volumetric absorber, but not by a surface absorber; the effect of this difference is discussed below. Losses other than radiative losses, such as convective losses, are neglected. This assumption is valid for receivers operating at high temperature, such as the DIAPR (Karni et al., 1996). Heat conduction in the direction of fluid flow is ignored, except for the isothermal absorber. It is assumed that the radiation entering the absorber has the same radiance temperature for all locations on the aperture, for all directions and both polarizations. The effect of spatial distribution is outside the scope of this paper. It may be treated by methods such as Ries et al. (1995). We discuss in the following the spectrally nonselective absorber, i.e., absorber for which the local absorptivity does not depend on photon energy. For the surface as well as for the volumetric absorber the optimum requires complete absorption, thus both appear black from the outside. With these assumptions in mind we can, without loss of generality, assume that the incident radiation is blackbody radiation. Furthermore, we assume that the absorptivity/emissivity coefficient does not depend on temperature. A nonselective volumetric absorber might be realized by a black mesh, with a volumetric absorption coefficient depending on the packing density of the mesh.

We consider the case where the inlet and outlet temperatures are given by the secondary thermal process which utilizes the heat from the absorber, and that a fluid with constant specific heat c_p transfers the heat.

2 Surface Absorber

In stationary operation the temperature $T(A)$ of a fluid passing sequentially a black surface absorber is, as a function of the area A, passed by the fluid with $0 \le A \le A_f$, given by

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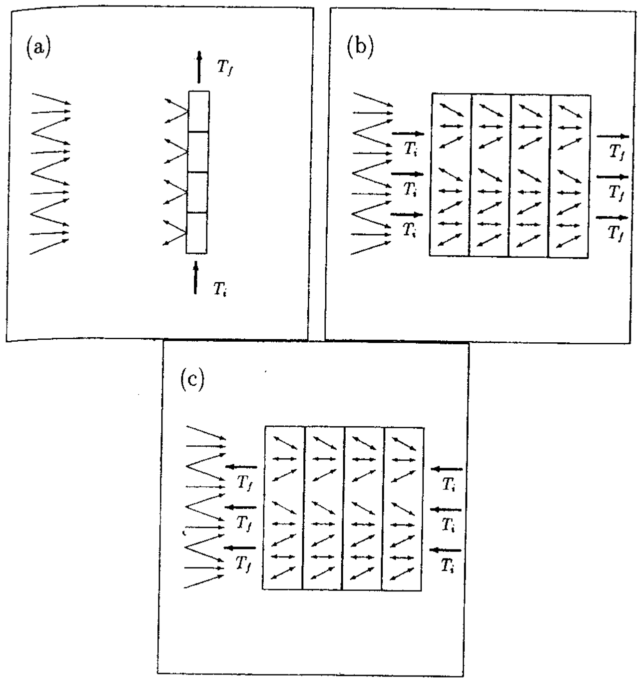

Fig. 1 Scheme of the surface absorber (*a*) and of the volumetric absorber in co-current operation (*b*) and in counter-current operation (*c*). Thin arrows correspond to radiation, thick arrows to fluid flow. For the discussion in the text, the absorbers are divided in sections along the direction of fluid flow.

$$j_S c_p \frac{dT}{dA} = M^{\text{inc}} - \sigma T^4, \tag{1}$$

where σ is Stefan's constant, and M^{inc} is the irradiance. Integration yields an implicit equation for the outlet temperature T_f:

$$j_S c_p \int_{T_i}^{T_f} \frac{dT}{M^{\text{inc}} - \sigma T^4} = A_f, \tag{2}$$

and the thermal efficiency η of the surface absorber is

$$\eta = \frac{j_S c_p (T_f - T_i)}{A_f M^{\text{inc}}} = \frac{T_f - T_i}{\displaystyle\int_{T_i}^{T_f} \frac{M^{\text{inc}}}{M^{\text{inc}} - \sigma T^4}\, dT}. \tag{3}$$

The efficiency can be expressed by the dimensionless temperatures t_i and t_f referred to the stagnation temperature $T_\infty = \sqrt[4]{M^{\text{inc}}/\sigma}$, i.e., $t_i = T_i/T_\infty$ for the inlet and $t_f = T_f/T_\infty$ for the outlet.

The efficiency of the black surface absorber according to Eq. (3) is (Ries et al., 1995):

$$\eta = \frac{t_f - t_i}{\phi(t_f) - \phi(t_i)}, \quad \text{with}$$

$$\phi(t) = \frac{1}{4} \ln \left(\frac{1+t}{1-t} \right) + \frac{1}{2} \arctan(t). \tag{4}$$

The efficiency of the isothermal absorber is obtained as the limit of η in Eq. (3) for $t_i \to t_f$,

$$\eta = 1 - t_f^4. \tag{5}$$

3 Volumetric Absorber

For a volumetric absorber, which is homogeneous in directions parallel to the aperture and has radiative properties with rotational symmetry around the normal of the aperture, the equations of radiative and fluid transport in stationary operation with one-dimensional fluid flow are (Brewster, 1992; Siegel and Howel, 1992):

$$\pm \frac{\partial I^\pm}{\partial x} = -\frac{\alpha(\theta)}{\cos(\theta)} \left[I^\pm - \frac{\sigma T^4}{\pi} \right] \tag{6}$$

$$j_V c_p \frac{dT}{dx} = 2\pi \int_0^{\pi/2} \frac{\partial (I^- - I^+)}{\partial x} \cos(\theta) \sin(\theta)\, d\theta. \tag{7}$$

Here I^+ and I^- are the intensity (radiance) along the absorber in forward (I^+) and backward (I^-) direction, respectively. The intensities I^+ and I^- are functions of the location x and of the angle θ, the angle between the radiation's propagation direction and the normal to the aperture. The efficiency is

$$\eta = \frac{j_V c_p (T_f - T_i)}{M^{\text{inc}}}. \tag{8}$$

The volumetric absorption coefficient α is assumed to be independent of photon energy and polarization. The volumetric absorber is assumed microscopically black; therefore we exclude scattering and reflection. For isotropic irradiance, the boundary condition at the inlet is

$$I^+(0, \theta) = \frac{M^{\text{inc}}}{\pi}. \tag{9}$$

At the outlet it is assumed that a mirror reflects all radiation,

$$I^+(L, \theta) = I^-(L, \theta), \tag{10}$$

where L is the length of the absorber.

Nomenclature

- A = area passed by the fluid (m^2)
- c_p = fluid specific heat per mass at constant pressure ($JK^{-1}\, kg^{-1}$)
- j_S = mass flux (per unit time) (kgs^{-1})
- j_V = mass flux density (per unit area and time) ($kgm^{-2}\, s^{-1}$)
- L = length of the absorber (m)
- I = intensity ($Wm^{-2}\, sr^{-1}$)
- M = radiant flux (Wm^{-2})
- t = temperature, normalized to T_∞
- T = absolute temperature of fluid and absorber (K)
- x = spatial coordinate, zero at the aperture (m)
- α = volumetric absorption coefficient (m^{-1})
- η = thermal efficiency
- σ = Stefan's constant, $5.67 \cdot 10^{-8}$ $Wm^{-2}\, K^{-4}$ ($Wm^{-2}\, K^{-4}$)
- θ = angle with respect to the normal of the aperture

Subscripts

- eff = effective
- i = fluid inlet, $x = 0$ for "co-current," $x = x_f$ for "counter-current" operation
- f = fluid outlet, $x = x_f$ for "co-current," $x = 0$ for "counter-current" operation
- ∞ = stagnation

Superscripts

- inc = coming from the sun into the aperture
- $+$ = forward radiation, coming from the direction of the aperture
- $-$ = backward radiation, into the direction of the aperture

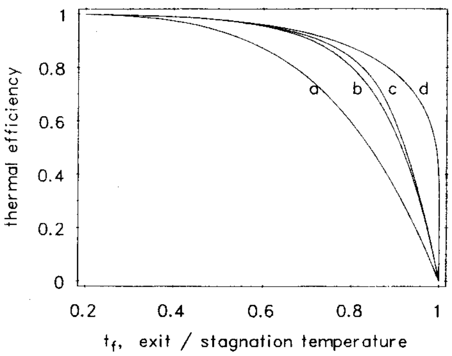

Fig. 2 Thermal efficiency of absorbers with radiative losses as a function of t_f, the ratio of the exit temperature to the stagnation temperature, (*a*) for an isothermal absorber, (*b*) for a volumetric absorber according to the two-flux model, in co-current operation, (*c*) in counter-current operation, (*d*) for a surface absorber. For the cases (*b*), (*c*), and (*d*) the normalized inlet temperature t_i was set to 0.2.

The two-flux model (Brewster, 1992; Siegel and Howel, 1992) is formally obtained as a special case if the volumetric absorptivity depends on the angle of incidence such that $\alpha(x, \theta) = \alpha(x, 0) \cos(\theta)$. Substituting into Eqs. (6) and (7) yields

$$\pm \frac{dI^{\pm}}{dx} = -\alpha \left[I^{\pm} - \frac{\sigma T^4}{\pi} \right] \qquad (11)$$

$$j_V c_p \frac{dT}{dx} = \pi \frac{d(I^- - I^+)}{dx}. \qquad (12)$$

The intensities I^+ and I^- do not depend on θ and are functions of x only. The two-flux model does not correspond to isotropic absorption and emission, where α does not depend on the direction θ. Generally it is used as an approximation to Eq. (6) and Eq. (7), and for isotropic absorber/emitters an effective coefficient α_{eff} in Eq. (11) is taken as a factor of 2 or $\sqrt{3}$ (Brewster and Tien, 1982) times the absorption coefficient α in Eq. (6). However, the two-flux model may also be taken literally. This means that in any direction the fraction of the radiation absorbed along a distance ds is $\alpha(x, 0)dx$, i.e., depends only on the distance dx projected in the direction normal to the aperture. This might be realized by an array of thin plates oriented parallel to the aperture. In contrast, for isotropic absorption/emission, the absorbed fraction is $\alpha ds = (\alpha/\cos(\theta))dx$. Then, I^+ and I^- are generally functions of both, x and θ.

For co-current operation, where the fluid enters the absorber at the aperture, $j_V > 0$ in Eq. (12). For counter-current operation $j_V < 0$. We assume that the absorber has infinite optical thickness, $\int_0^{x_f} \alpha dx = \infty$. The boundary conditions are Eq. (9), Eq. (10) and $T(0) = T_i$, $T_f = T(x_f)$ for co-current operation, or $T(x_f) = T_i$, $T_f = T(0)$ for counter-current operation.

For numerical calculations, the absorber was partitioned into 100 sections. The optical thickness of each section was chosen such that adjacent sections roughly differ equally in radiative flux. A nonlinear relaxation algorithm was applied as follows: For co-current operation, the system is solved from inlet to exit with fixed I^-. A find-zero procedure using alternatively bisectioning and the regula falsi (Press et al., 1992, Chapter 9) is used to evaluate the new temperature for each section. Then I^- is calculated with Eq. (11), and this scheme is repeated until convergence. For counter-current operation, I^+ is calculated from Eq. (11), and then the system is solved from exit to inlet with fixed I^+.

Figure 2 shows the thermal efficiency for the isothermal absorber (Eq. (5)), the surface absorber (Eq. (3)), and the volumetric absorber, in co-current and counter-current operation. All efficiencies are shown as functions of t_f. For the cases (*b*), (*c*), and (*d*) the normalized inlet temperature t_i was set to 0.2. For example if $M^{\text{inc}} = 1$ MW m^{-2}, $T_\infty = 2050$ K, and the inlet temperature is 410 K. For the isothermal case (*a*) losses depend only on the outlet temperature t_f and therefore the inlet temperature is irrelevant. If t_i is near to t_f, the efficiencies of all types of absorbers shown are equal. For $t_f < 0.6$ all nonisothermal absorbers (*b*, *c*, *d*) are nearly equal and exhibit far less losses then the isothermal absorber (*a*). Practical receiver temperatures may be determined by the requirement of reasonable efficiency, say 80 percent or 90 percent, implying t_f in the range of 0.6 to 0.8. For real receivers, such levels of efficiency will be obtained at lower exit temperature, possibly around 0.5.

A possibly surprising result is that the volumetric absorber, although recycling part of its own reradiation, is less efficient than the surface absorber. In both cases, the fluid is heated while flowing from inlet to outlet, without heat conduction in the direction of flow. However, the different sections of a volumetric absorber which are at different temperatures interact with each other through radiative heat exchange, whereas the different sections of a surface absorber absorb and emit radiation independently from each other. The interaction produces entropy additional to the entropy produced by absorption and thus lowers the work obtainable from the outlet fluid. The absorber with maximum interaction, the isothermal absorber, has the lowest efficiency.

For given inlet temperature and fluid flow rate, the two-flux model underestimates the efficiency of the isotropic absorber. For the isotropic absorber the normal radiation penetrates deeper into the absorber while the radiation at grazing angles is absorbed near the inlet. Thus the isotropic absorber is closer to the surface absorber than the two-flux absorber. However, for isotropic radiation the efficiencies calculated with the two-flux model are very close to those obtained with the isotropic model as shown in Fig. 3. The discussion here therefore does not distinguish between the two-flux and the isotropic absorber.

The isotropic absorber demonstrates a certain level of directional selectivity, where radiation from different directions is attenuated at different rates as a function of x. A fully directionally selective absorber may be thought of, where for each location x only radiation from a certain direction θ is absorbed. For isotropic irradiance, the performance reaches that of the surface absorber. For nonisotropic irradiance, an absorber might be tailored, similarly to the spectrally tailored, fully selective absorber (Bejan, 1988, Chapter 9). Such a fully directionally selective absorber can in principle replace concentration by geometrical means.

It may also be surprising that the co-current operation is less efficient than counter-current operation. One might argue that in co-current operation the radiation emitted by the hot fluid exit is partially recycled by reabsorption, whereas in counter-

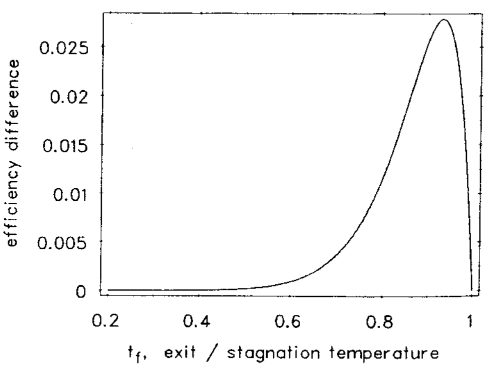

Fig. 3 Difference of the thermal efficiency of an isotropic minus that of a two-flux volumetric absorber with radiative losses as a function of t_f, the ratio of the exit temperature to the stagnation temperature, in co-current operation. The two-flux absorber corresponds to curve (b) in Fig. 2.

current operation the radiation from the hottest section is emitted at the entrance of the solar radiation and therefore not recycled (Buck, 1988). In a volumetric absorber, the radiance temperature plays the same role as the supply-side temperature of a conventional heat exchanger.

We argue that the advantage of the counter-current operation is even larger if the heat transfer coefficient between absorber and fluid is finite. For co-current operation, the heat transfer rate and thus the temperature difference between absorber and fluid is peaked near the aperture. This introduces radiative losses from the absorber at the cold side of the fluid, additional to the losses coming from the hot region at the fluid exit. Significant losses due to high absorber temperatures near the aperture, much higher than that of the working fluid, have been observed in (Buck, 1988; Kaminski and Kar, 1992; Menigault et al., 1991; Posnansky and Pylkkanen, 1991).

It might be argued that a finite optical thickness should be selected for the co-current mode to avoid the large region causing reradiation losses; we show in the following that this is not the case. Assume that, with the flow rate fixed, an additional absorber slab is put between the exit of a given volumetric absorber and the reflecting back wall. This decreases the solar radiation reflected back through the aperture, say by γM^{inc}. Additional emission losses from the slab are reradiated, say at temperature T, hence the net reduction is $\gamma(M^{inc} - \sigma T^4) > 0$. The original absorber receives a lower flux from the slab compared to the flux received without the slab; thus its temperature also decreases giving another reduction of reradiation. Thus by adding a slab reradiation decreases, i.e., the efficiency increases; the optimum optical depth is infinite.

4 Discussion

The objective of this work is to show that the volumetric absorber is less efficient, but closely approaches the performance of the ideal nonisothermal surface absorber. Since the volumetric absorber provides a much larger heat transfer area, nonperfect heat transfer degrades the performance of a surface absorber much more than it degrades that of a volumetric absorber. In real absorbers, the volumetric design may therefore be equal or even superior to the surface absorber. The benefit of volumetric absorbers does not consist then in recycling of radiative losses, but rather in more efficient heat transfer between absorber and fluid.

The isotropic volumetric absorber is less efficient than the surface absorber, since the volumetric absorber necessarily shows radiative interaction between different sections. However, in the region of outlet temperatures of 60 percent to 80 percent of the stagnation temperature, the efficiencies are far better than for the isothermal absorber. The counter-current operation of the volumetric absorber is slightly superior to co-current operation. Practical considerations, but not reduction of radiative losses, might prefer the co-current operation.

The surface absorber, which may be considered as a cross-current heat exchanger, has the highest efficiency. We thus conclude that the volumetric absorber with one-dimensional flow may be viewed as a slightly nonperfect nonisothermal absorber. However, a hypothetical volumetric absorber with directional selectivity such that in each slab only a certain incidence angle is absorbed can exactly match the efficiency of the surface absorber.

At high temperatures radiative losses usually dominate. There is no simple argument concerning the effect of other losses because the systems considered are very different, in particular the temperature of the front surface differs greatly between co-current and counter-current operation. Convective losses should favor co-current versus counter-current operation due to lower front surface temperatures.

This investigation has been restricted to the case where the irradiance is homogeneous. In the case of nonhomogeneous irradiance with regard to the location on the aperture, a surface absorber may even be improved by tailoring the flow pattern to the irradiance profile (Ries et al., 1995). However, in the case of nonhomogeneous irradiance with regard to the incidence angle, the photon energy, or the polarization, a black surface absorber might even be surpassed by a volumetric absorber with radiative properties tailored to the irradiance distribution.

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