From: Solar Collectors, Energy Storage, and Materials,
F. de Winter, Ed., MIT Press, Cambridge MA, 1990

4 Thermal Theory and Modeling of Solar Collectors

Noam Lior

This chapter reviews the advances made in thermal theory and modeling of nonconcentrating and concentrating solar collectors. It attempts not to emphasize thermal *research*, which is covered in chapter 8 of this book, nor optical aspects (in coatings that affect radiative transfer or in concentrating collectors), which are covered in chapter 7. An effort is made to describe the current, least speculative understanding of the thermal theory of solar collectors and of the state-of-the-art modeling. Each major section ends with a "progress summary" that highlights the progress made since the early 1970s. According to the editor's dictate, this chapter is not to serve as a source of all equations needed for thermal modeling, but it should be an up-to-date resource for the relevant references and progress accomplished. The missing equations and further detail should be sought in the references quoted, where they can be found in their most original form, unspoiled by any errant middleman.

Based on the work of solar collector theory pioneers such as Hottel (see Hottel and Woertz 1942; Hottel and Erway 1963), Tabor (1955, 1958), Whillier (1953, 1964a, b), Bliss (1959), and others, the state-of-the-art in the 1960s was summarized by Hottel and Erway in the book edited by Zarem and Erway (1963), by Whillier (1967) in an ASHRAE book, and in the first edition of the book *Solar Energy Thermal Processes* by Duffie and Beckman (1974). These publications serve as the datum level to which later progress is compared in this chapter.

The rapid progress made in the mid- and late 1970s was summarized in an updated (1977) version of the ASHRAE book (Liu and Jordan 1977), the booklet by Edwards (1977), the book by Kreith and Kreider (1978), the 1980 edition of the book by Duffie and Beckman, a number of review chapters by Löf (1980), Rabl (1980, 1981), and Kreith and Kreider (1981) in the volumes edited by Dickinson and Cheremisinoff and by Kreider and Kreith, the book by Garg (1982), and several other books, such as by Howell et al. (1982) and Rabl (1985). The number of books and comprehensive reviews on this topic diminished precipitously (alongside with the solar energy research budget) in the mid-1980s, and newer information is primarily contained in journals and conference publications.

This chapter begins by describing, in section 4.1, the overall thermal balance in the collector. The first section then deals in more detail with its various components, such as the collector exterior, the window system, enclosed insulating spaces, and the absorber and the working fluid. It addresses the effect of transient conditions, performance sensitivity to design parameters, and the differences between the modeling of a single collector and of collector arrays. At the end of the chapter a state-of-the-art modeling method is outlined.

4.1 The Thermal Energy Balance for Solar Collectors

4.1.1 General Description of Solar Collectors

Types of Nonconcentrating Solar Collectors *Flat plate solar collectors* utilize a flat absorber plate to convert the electromagnetic energy of solar radiation to heat. To reduce heat losses to the ambient, the insolated front of the absorber is usually separated from the ambient by a "window" that allows transmittance of solar radiation to the absorber but impedes heat losses from the absorber to the ambient. This is usually accomplished by one or more panes of glass or plastic transparent to radiation in the solar (shortwave) spectrum. These panes are mounted parallel to the absorber, with small air gaps between them. The uninsolated back and sides of the collector are insulated by conventional, opaque insulation. Useful heat is taken away from the absorber by putting a working fluid in contact with it: either directly by flow over or below the absorber (as typically done in air-heating collectors) or by flow-through conduits (often simply tubes) in good thermal contact with the absorber. Manifolding devices are used to distribute the fluid properly over the absorber area at the inlet to the collector and to collect the heated fluid into the outlet conduit. Figures 4.1 and 4.2 show typical configurations of liquid-heating and air-heating flat plate solar collectors, respectively.

To eliminate the resistance to heat transfer associated with the conduit for the heated fluid, a number of flat plate collector designs in which the solar radiation is absorbed directly into a flowing layer of the heated fluid have been designed and tested. The fluid is made more absorbent to radiation by adding solid suspensions or dyes that absorb solar energy. The "black fluid" in the designs considered so far either flows as a thin film on the absorber plate or passes through transparent tubing. The film or tubing is typically contained in a conventional solar collector box separated by one or more transparent panes from the environment.

Cylindrical solar collector are usually constructed in the shape shown in figure 4.3. This design allows the use of a relatively thin glass window for containing an insulating vacuum between the absorber and this exterior glass window. Evacuation of the gap between the absorber and the window reduce both convective and conductive heat losses significantly, and reduce them altogether at a perfect vacuum. Many designs have been proposed, built, and tested (see Graham 1979). One design by Corning Glass (figure 4.3a) has

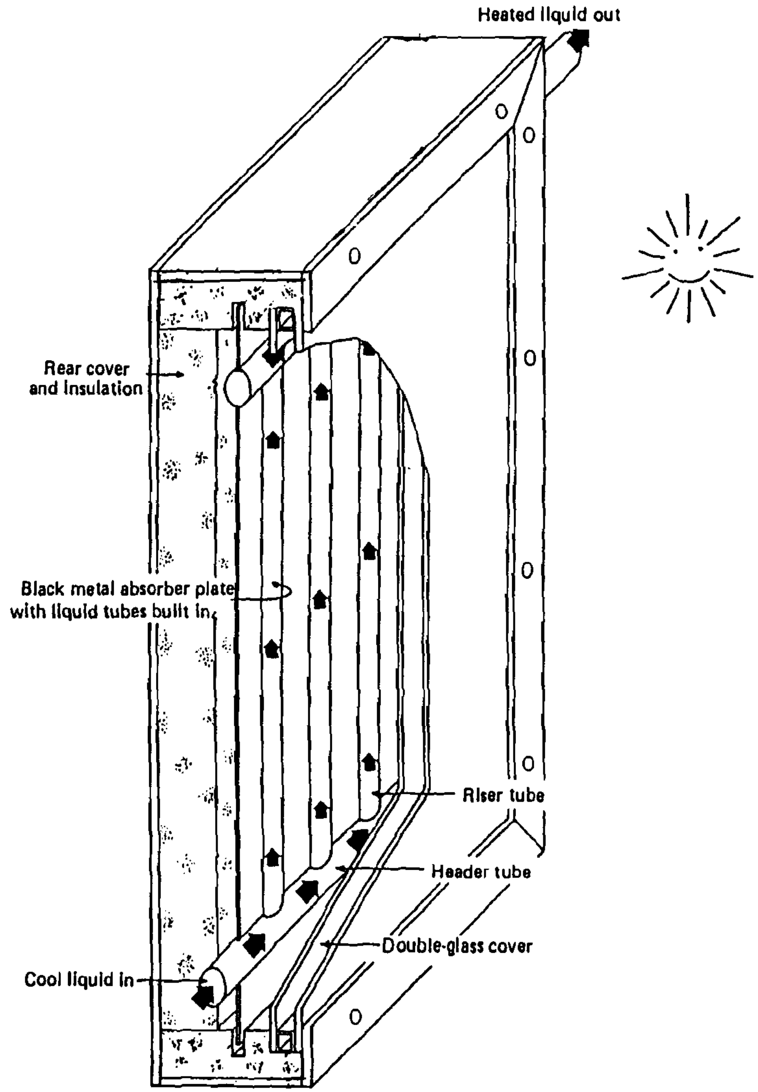


Figure 4.1
Typical liquid-heating flat plate solar collector.

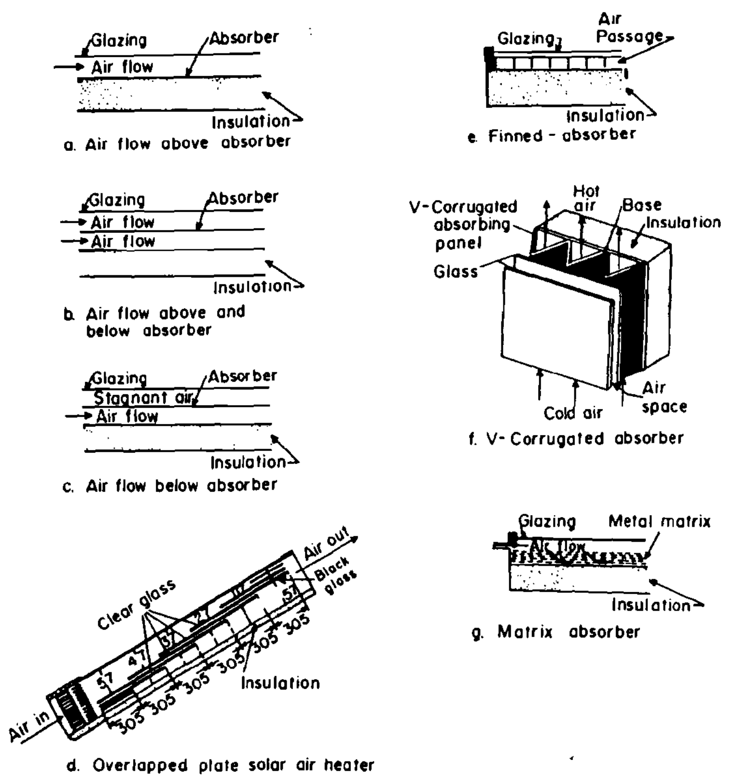


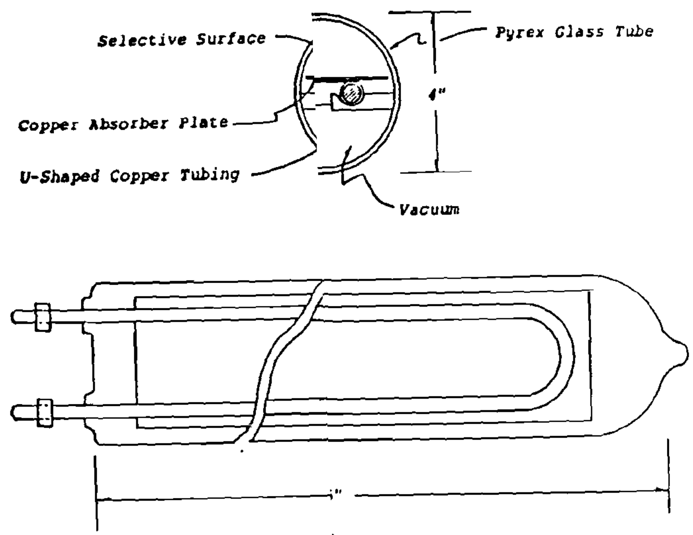
Figure 4.2
Air-heating solar collector configurations: (a) airflow above absorber; (b) airflow above and below absorber; (c) airflow below absorber; (d) overlapped plate solar air heater; (e) finned absorber; (f) V-corrugated absorber; (g) matrix absorber.

simply a flat plate absorber in the middle of the cylinder attached to a U-tube conducting the fluid. In another by Phillips (figure 4.3b) the U-tube is not attached to an absorber. In the Owens-Illinois design (figure 4.3c) the fluid is fed in through a central small tube and is returned through a concentric annulus formed between that tube and a larger cylindrical absorber. Figure 4.3d shows a design in which a Thermos bottle design is employed for the glass vacuum envelope and a copper U-tube is attached to a copper sleeve simply inserted into the center.

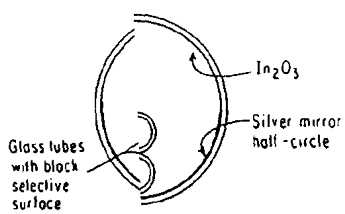
Types of Concentrating Solar Collector Receivers In a flat plate collector the heat loss area is equal to the insolation collection area. The primary advantage of the concentrating solar collector is that the heat loss area (that of the receiver) is smaller (up to several thousand times) than the insolation collection area (that of the reflector). This allows higher efficiency for a given useful output temperature, or a higher temperature for the same efficiency as attained by a flat plate solar collector. The various methods of concentrating solar radiation will not be discussed here because they are not within the scope of this chapter (some discussion of concentrators appears in chapters 2, 3, and 7). In general, a concentrator would be considered in the thermal design only if it has some thermal effect on the receiver such as in producing a radiative exchange surface or in confining or channeling convection. For thermal design it is usually sufficient to know the radiative flux incident on the receiver.

At the lower energy collection temperature levels (e.g., $< 200°C$) the receiver configurations are similar to the flat plate collectors shown in figures 4.1 and 4.2 and to the evacuated cylindrical collectors shown in figure 4.3. In some designs using trough (line-focusing) concentrators, the receiver is simply a blackened tube conducting the heated fluid. The tube is often insulated from the ambient by a larger concentric glass tube, with a thermally insulating annular air gap between the two tubes. The glass tube can also be coated with an antireflective coating. Cavity receivers and external receivers are most often used for the high temperatures typically obtained from point-focusing concentrators (dish or distributed mirrors focusing on a central tower; see Hildebrandt and Dasgupta 1980; Kreith and Wang 1986; Lior 1986).

Cavity receivers consist of an insulated enclosure with an optical aperture just large enough to admit the concentrated solar radiation beam, which contains on its interior the heated fluid conduits (see McKinnon et al. 1965; Grilikhes and Obtemperanskii 1969; Kugath et al. 1979; Strumpf et al. 1982; Davis 1982; Wu et al. 1983; Borgese et al 1984; Harris and Lenz 1985). Its geometry has the objective of maximizing the absorption of the incident radiation, minimizing heat losses (radiant and convective) from the cavity to

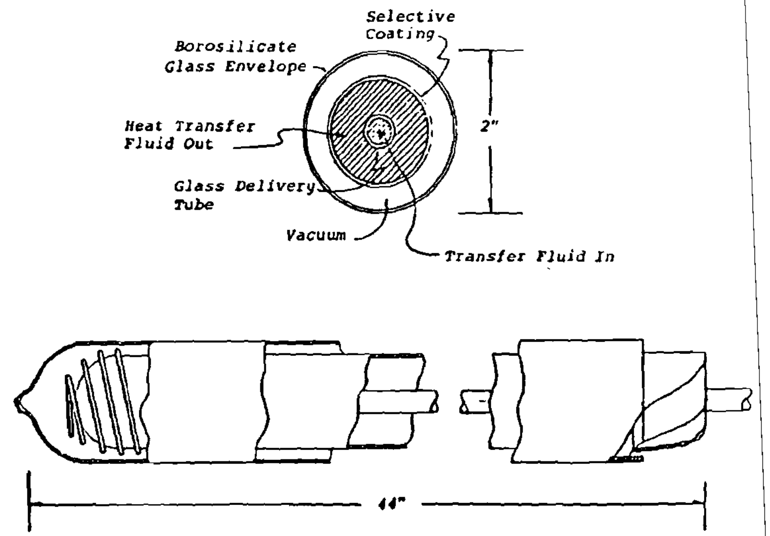


)

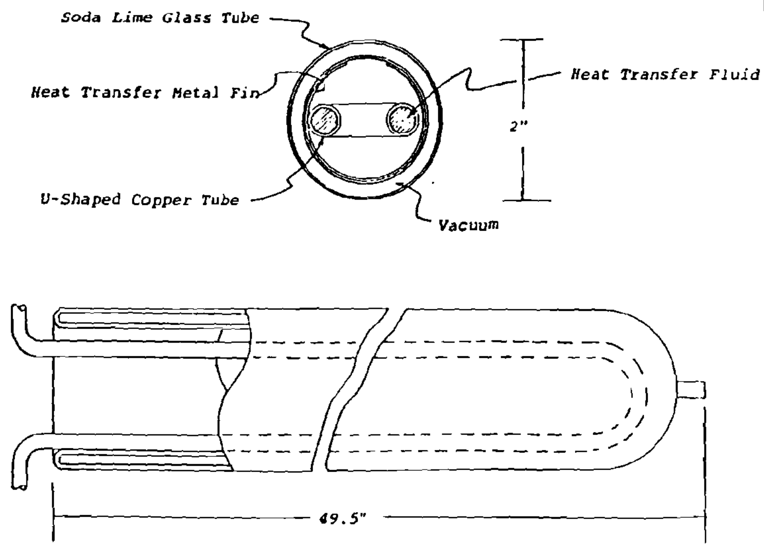


)

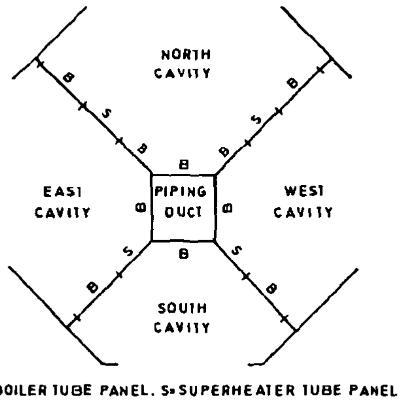
Figure 4.3
Evacuated cylindrical solar collectors: (aing Cortec™ evacuated tube collector; (b) Philips (Aachen) GmbH with In_2O_3 infrared re; (c) Owens-Illinios SUNPAK™ evacuated tube collector; (d) General Electric TC-100 S(TRON™ evacuated tube collector.



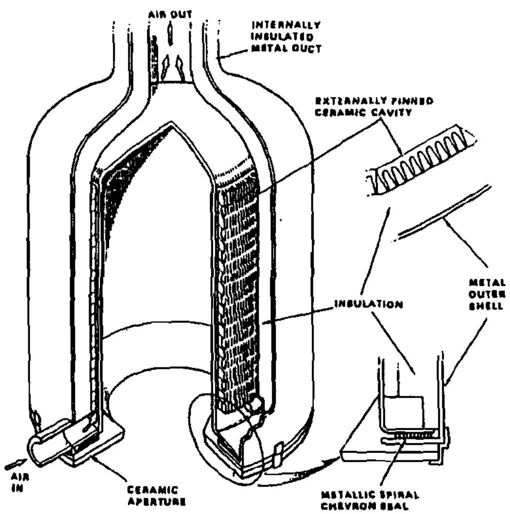
(c)



(d)

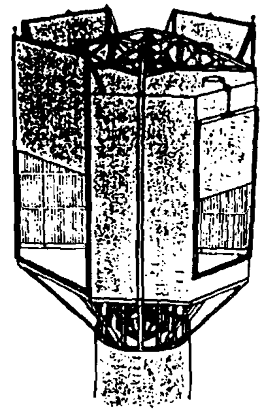


a. "Quad" central tower cavity receiver:
steam boiler and superheater (Wu et al., 1983)



b. Ceramic finned-shell air heating receiver
for parabolic-dish concentrator (Strumpf et al., 1982)

Figure 4.4
Cavity receivers: (a) "quad" central tower cavity receiver—steam boiler and superheater (Wu et al. 1983); (b) ceramic finned-shell air heating receiver for parabolic dish concentrator (Strumpf et al. 1982); (c) central tower molten salt receiver (Martin-Marietta)



c. Central tower molten salt receiver
(Martin-Marietta)

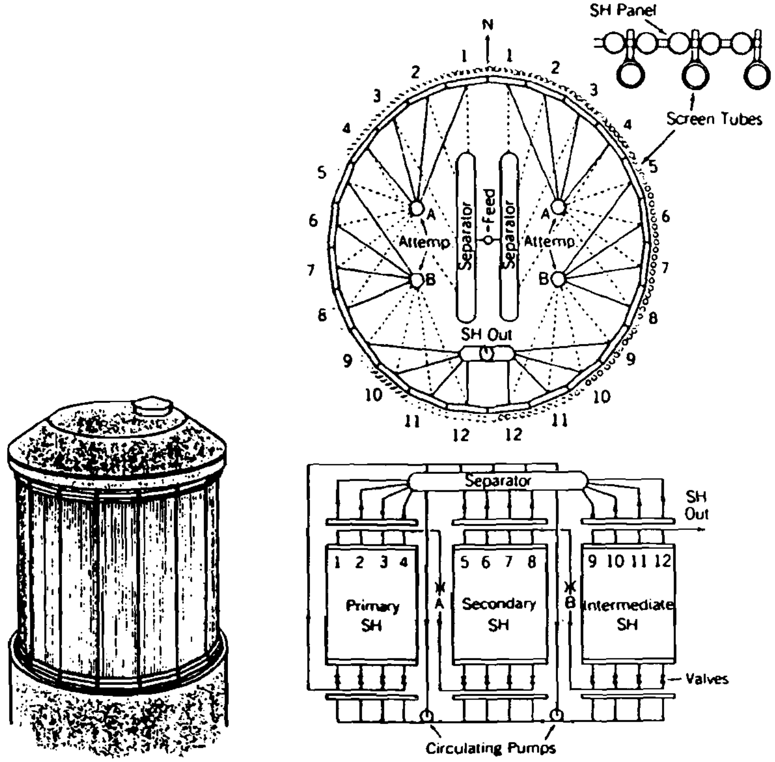
Figure 4.4 (continued)

the ambient, accommodating the heat exchangers that transfer the radiant energy to the heated fluid, and distributing the radiant flux over the surface of the heat exchangers properly. Typical cavity receiver configurations are shown in figure 4.4.

External receivers are similar to nonconcentrating collectors in that they expose the heated fluid conduits (surface treated to increase absorptance of the radiation) directly to the radiative flux without the radiation-trapping benefits of the cavity (see Dunn and Vafaie 1981; Chiang 1982; Durant et al. 1982; Wu et al. 1983; Yeh and Wiener 1984). They consequently have lower efficiencies than cavity receivers but are smaller and more economical to manufacture. In a design study such a receiver could be considered for use with the central tower solar concentrator power scheme, as shown in figure 4.5.

To reduce resistance to heat flow and to eliminate the need for exposing the fluid conduits to high radiative fluxes and temperatures, a number of receiver designs have been proposed, and partially tested, in which the solar radiation is absorbed directly into either a falling liquid (molten salt) film (figure 4.6; see Bohn 1985; Lewandowski 1985) or into free falling solid particles (figure 4.7; see Hruby 1985; Hunt et al. 1985).

More detail on various collector designs and configurations is given in chapter 2 of this volume.



a. b.

Figure 4.5
External receivers: (a) sodium (General Electric); (b) steam (Durrant et al. 1982).

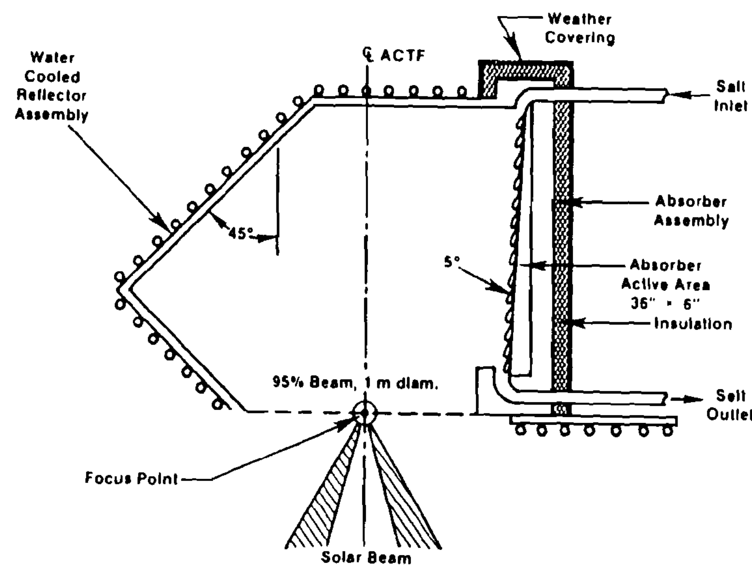


Figure 4.6
Direct absorption molten-salt receiver. Source: Kreith and Anderson (1985).

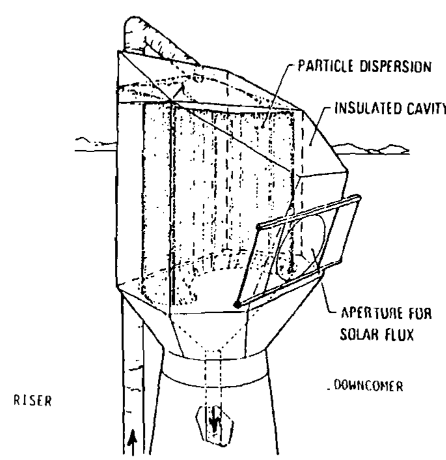


Figure 4.7
Solid particle cavity receiver. Source: Hruby (1985).

4.1.2 The Thermal Energy Balance

The description of the thermal theory and modeling of solar collectors will start here with the energy conservation equation. Each component will be identified and then decomposed into its heat transfer rate subcomponents. The progress made in the ability to evaluate these components will be described in the subsequent sections 4.2 through 4.11.

In its general form the conservation law states that the rate of energy input (E_i) is equal to the sum of the rates of useful heat obtained from the collector (q_u), energy losses from the collector (E_l), and heat storage in the collector (q_s):[1]

$$E_i = q_u + E_l + q_s. \tag{1}$$

In many cases the thermal capacity of the collector is small, or the transients are moderate. In these cases the last term drops out. In steady-state analyses the last term is zero, and the rest of the terms are assumed to be quasi-invariant with time.

The *radiative* energy input E_R is composed of the components of the beam insolation I_b and diffuse insolation I_d (I_b and I_d are measured on a horizontal surface at the same location and time) that are normal to the collector surface (see figure 4.8), multiplied by the collector window area (A_w):

$$E_R = (R_b I_b + R_d I_d)A_w = RIA_w, \tag{2}$$

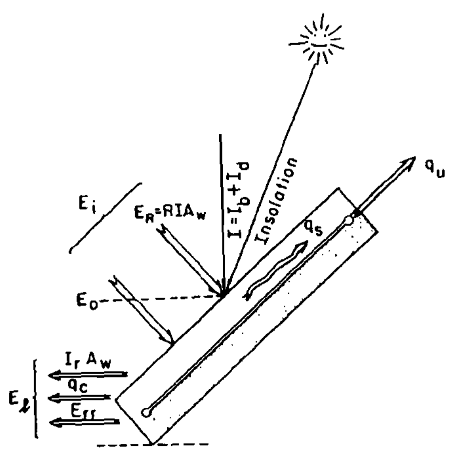


Figure 4.8
Overall energy balance (external envelope control volume) of a solar collector. E_0 is any nonsolar heat input.

where R_b, R_d, and R are the ratios of the beam, diffuse, and total radiation on the collector surface to that on the horizontal surface,[2] respectively;

$$R_b = \frac{I_{bc}}{I_b}, \tag{3}$$

$$R_d = \frac{I_{dc}}{I_d}, \tag{4}$$

$$R = \frac{I_c}{I}, \tag{5}$$

where the subscript c indicates the collector surface, and

$$I = I_b + I_d. \tag{6}$$

The early work by Liu and Jordan (1963) and many subsequent analyses have assumed that the diffuse radiation component is directionally isotropic. Progress made since then indicates that circumsolar diffuse radiation can be as much as an order of magnitude higher than the diffuse radiation from directions farthest from the sun and that the distribution of diffuse radiation also depends significantly on the composition of the atmosphere and the cloud cover. This consideration is very important in areas where the diffuse component is a significant fraction of the total solar radiation. Consequently R_d at a given geographic location will not only be a function of the tilt of the collector but will also depend on the solar angle. With a clear sky, it is often assumed that $R_b = R_d = R$.

From the available data it appears that the diffuse component has a spectral distribution similar to that of the total solar radiation, with possibly a very slight shift to the shorter wavelengths. This allows the assumption in thermal modeling that both of the radiation components have the same spectral distribution.

Any further discussion of the directional and spectral dependence of insolation is beyond the scope of this chapter and is treated in more detail in volume 2 and in chapter 3 of this book. The preceding comments have been made primarily to indicate that the energy input term is a function of direction and wavelength; this fact influences the thermal phenomena in the collector.

As the radiation q_R incident on the collector strikes its exterior surface, it proceeds to interact with the collector components through one or more reflective, absorptive, and transmissive (the latter if a transmissive component is in its way) processes. Absorption of radiation into any of the collector components is a process that converts the electromagnetic energy of radiation into heat. Reflected and transmitted radiation are still in the original form of electromagnetic energy. But since all practical reflection and transmission processes are accompanied by some amount of absorption, the amount of the radiant energy of a beam after reflection or transmission is always smaller than the beam's radiant energy before such interactions have occurred. The control volume drawn around the external envelope of the collector (Figure 4.8) indicates the energy fluxes crossing that envelope, which must satisfy conservation regardless of the processes occurring inside the control volume. It shows the fraction lost due to reflection from exterior and interior surfaces, I_r, the fraction reradiated from the exterior and interior surfaces to the ambient, E_{rr}, the fraction convected to the cooler ambient, q_c, and the useful heat picked up by the heated fluid, q_u. The energy difference remaining, q_s, is the rate of energy storage in the collector structure (with nonsolar energy inputs ignored):

$$I_c A_w - I_r A_w - E_{rr} - q_c - q_u = q_s, \tag{7}$$

but

$$I_c - I_r = (1 - \rho) I_c = \alpha I_c \tag{8}$$

(since none of the incident radiation is usually transmitted through the entire collector), where ρ is the reflectance of the collector assembly and α the absorptance of the collector assembly.

Using the notation of equation (1) in equation (7) the total energy loss from the collector can be obtained:

$$E_l = I_r A_w + E_{rr} + q_c. \tag{9}$$

Using equation (8), the reflection loss is separated from the other terms on the right-hand side of equation (9) to yield the following equation for the useful heat:

$$q_u = \alpha I_c A_w - (E_{rr} + q_c) - q_s. \tag{10}$$

The second term on the right-hand side of equation (10) represents the heat losses due to reradiation to the ambient (from the exterior surface of the collector), as well as to the fraction of radiation emitted from the interior surfaces and subsequently transmitted into the ambient, and due to convection from the collector exterior surface to the ambient. To solve equation (10) for q_u, all the terms on the right-hand side must be expressed as known functions of given quantities. Thus for a given collector component configuration,

$\alpha = \alpha$ (direction, spectrum, temperature),

$I_c = I_c$ (direction, spectrum),

$E_{rr} = E_{rr}$ (temperature, and emittance, transmittance, and reflectance at this spectrum and temperature of all participating components of the collector; temperature of the ambient),

$q_c = q_c$ (temperature of the exterior surface of the collector, temperature of the ambient, thermophysical transport properties),

$q_s = \partial(\rho V c_p T)/\partial t = q_s$(temperature, density, and volume of each of the collector components, including the working fluid; time).

It is important to note that a solution of the real problem described by equation (10) requires the determination of complete radiative transfer—namely, transmission, reflection, and absorption—and of all of the temperatures in the entire collector structure.

Electrical circuit analogies have been used to aid in clarifying the energy balance as well as the internal energy transfer. An early analogy of this type that includes the consideration of thermal capacity of the collector elements was suggested for flat plate collectors by Parmelee (1955) and used in the transient analysis of such collectors by Kamminga (1985). These analogies were also used for the description of steady-state energy balances and heat transfer by Duffie and Beckman (1974, 1980), Kreith and Kreider (1981), and many others.

Superimposed on a sketch of a two-plate solar collector, a fairly comprehensive circuit analogy is shown in figure 4.9. The incident insolation I is diminished in passage through each of the transmissive covers by reflection from each of the cover/air interfaces and by absorption in the cover. The amount I_2 arrives at the absorber surface, and a fraction of it is absorbed and converted into heat. The remainder is reflected. Not noted on the schematic for reasons of clarity is the fact that all reflected radiation, as well as radiation emitted by the collector components, undergoes further transmissive, reflective, and absorptive processes as it interacts with collector components its path. Further detail on this is given in subsection 4.2.1. Thermal resistances wired in parallel indicate that both radiative and convective, or conductive, transport are present. The exterior surfaces of the collector exchange radiation with the objects "visible" to these surfaces. Typically the sky-exposed side of the collector will exchange radiation with the sky, at the "sky temperature" T_s, while other parts of the collector surface may exchange radiation with the surrounding ground, obstructions such as trees, buildings, or other collectors, and possibly the sky too. An equivalent temperature of this part of the environment can be formulated, shown in figure 4.9 as

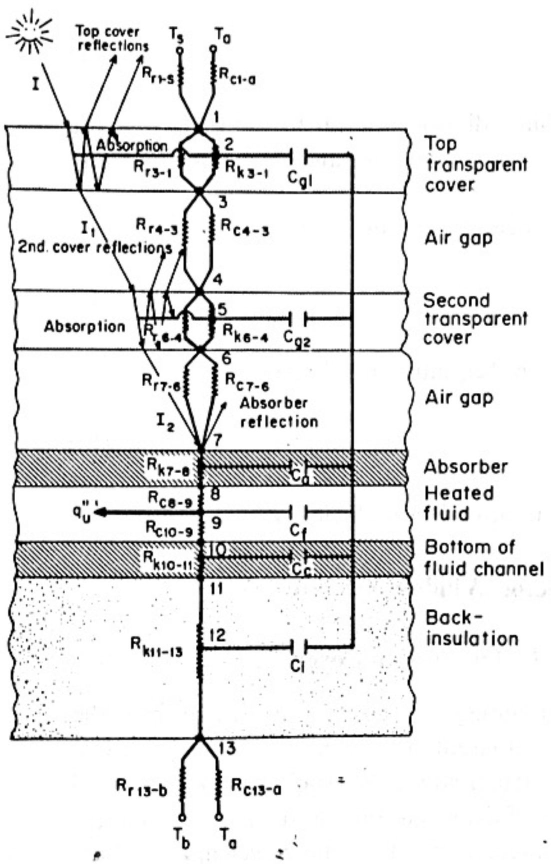


Figure 4.9
Heat transfer electric circuit analog for double-glazed solar collectors.

C_{g2}, C_a, C_c, C_f, and C_i represent the thermal capacitance of the two cover plates, the absorber and the fluid channels bottom, the working fluid, and the back insulation, respectively. q_u'' is the useful heat flux (electric current in the analog circuit) taken away by the working fluid. In operation I, T_a, T_s, T_b, R_{c1-a} (through the wind velocity), and q_u (through working fluid flow rate and inlet temperature) are (or have) transient inputs.

Although figure 4.9 may be applied to the entire collector, with area-averaged parameters, one must recognize that practically all of the resistances and fluxes are location dependent. Consequently such a schematic may be used also to describe transport in a differential element of the collector and integrated over the area. Two other aspects not shown in figure 4.9 are (1) the effect of the collector sides in partially shading the absorber from the incident radiation, on convection and radiation in their vicinity, and in additional heat losses through the sides, and (2) the existence and influence of the fluid distribution manifolds (headers).

It is often assumed that the collectors operate in quasi-steady state, that is, that the conditions remain steady for given time periods (e.g., for an hour) with step-updates in the forcing parameters after each such period. Other often-used assumptions are that all internal emission may be ignored (since most of it gets absorbed in the internal components anyway), that there is one-dimensional heat flow through the covers and the back insulation, that the conditions over the collector are uniform, that there is negligible temperature drop through a cover plate, and that heat losses from the collector surfaces are all to the same heat sink at temperature T_a. An analog circuit diagram of this simplified system is shown in figure 4.10a. Substitution of equivalent resistances, and computation of the insolation striking the absorber, with a further assumption that solar radiation does not generate heat as it is attenuated in passage through the cover plates, results in the most simplified (and most-often-used) analog circuit of the solar collector, depicted in figure 4.10b. This reduces equation (10) to the well-known energy balance expression

$$q_u = IR(\alpha\tau)_{ef}A_w - U_p(T_p - T_a) = I_c(\alpha\tau)_{ef}A_w - U_p\Delta T_{pa}, \tag{11}$$

where $(\alpha\tau)_{ef}$ is the overall effective product of absorptance and transmittance of the collector and T_p the absorber temperature. Comparison of equation (11) with (10) shows that in (11) the energy storage term was eliminated (steady state), that the window cover plates act only to attenuate the incoming solar radiation without their thermal participation, and that the convective and radiative heat losses were combined into a Newtonian overall heat loss coefficient U_p multiplied by the first-order temperature difference between

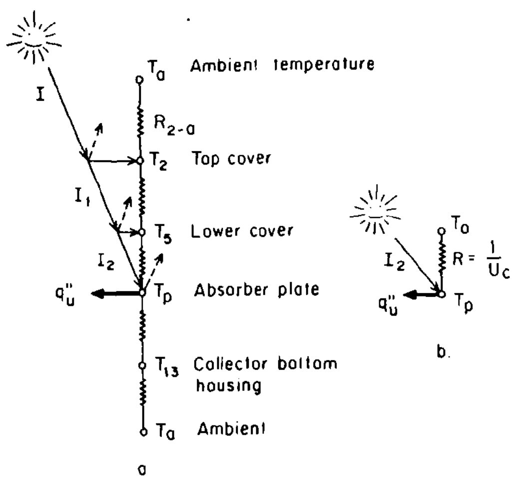


Figure 4.10
Simplified heat transfer circuit analogs for solar collectors: (a) first simplification; (b) most simplified.

the *absorber plate* and the ambient T_{pa}. Despite the seemingly simple form of equation (11), it should be noted that the theoretical determination of $(\alpha\tau)_{ef}$ and U_p requires the analysis (albeit somewhat simplified) of radiative-convective heat transfer within the collector.

The useful heat from the collector, q_u, amounts to the increase in energy of the heated fluid as it passes through the collector. This heat is usually acquired either by transfer from the hotter walls of the fluid conduits or by direct absorption of solar energy into the fluid (or solid particles, in the case of one of the receivers considered).

The temperature of the absorber plate, T_p, is not convenient for practical analysis and design of collectors since it is usually unknown. The known or desired temperature is usually the temperature of the heated fluid. In collectors where the heated fluid does not absorb radiation directly, T_p is expressed in terms of the heated fluid temperature by analyzing the heat transfer between the absorber/fluid-conduit system and the fluid. If the fluid is absorbing radiation directly, this absorption is added to the heat transfer analysis. In both cases, as will be explained further in section 4.3 a relationship is found between the absorber and fluid temperatures, completing the thermal design process.

At this point it should be noted that the efficiency of a collector, η, is defined as

$$\eta = \frac{q_u}{I_c A_w}. \tag{12}$$

As mentioned above, I_c is the solar radiation normal to the surface of the collector before any possible transmission, reflection, and absorption with any of the collector components occur.

Using equation (10), the efficiency can be expressed as

$$\eta = \alpha - \left[\frac{(E_{rr} + q_c) + q_s}{I_c A_w}\right]. \tag{13}$$

Using the simplified energy balance [equation (11)] gives

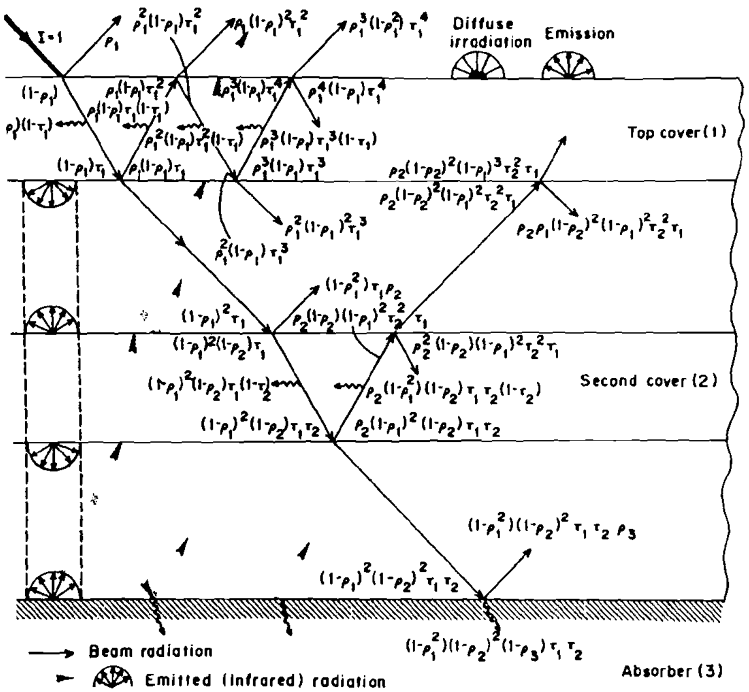
$$\eta = (\alpha\tau)_{ef} - \left(\frac{U_p \Delta T_{pa}}{I_c A_w}\right), \tag{14}$$

the most frequently used expression for collector efficiency.

4.2 Heat Transfer in the Collector Window System

4.2.1 Radiative Transfer through the Transparent Covers

Radiative transfer in a system containing an absorber separated from the ambient by one or more partially transparent plates consists of multiple transmissions, reflections, absorptions, and emissions. Two such plates are depicted in figure 4.11. Because of the nature of the radiation and the properties of the related materials that these phenomena depend on the wavelength and the direction, temperature, and various surface properties. Although most texts indicate only the multiple reflections, absorptions, and transmissions that a directional beam undergoes in the solar collector window system, it is worth noting that emitted (infrared) radiation undergoes similar processes too. The relationship with temperature couples the radiative problem with other heat transfer processes in the system, such as convection and the ubiquitious conduction. Strictly speaking, the formulation of this problem is difficult, and it would involve a set of integro-differential equations that are hard to solve. It is no surprise therefore that earlier formulations and solutions (starting perhaps with Stokes 1862) assumed that radiative transfer could be decoupled from other heat transfer, that the fully spectral models could be represented by two-band models (shortwave in the solar spectrum, and infrared), that glassplates that are opaque to the infrared band were used, and often that the absorption of the shortwave band in the plates would be negligible (i.e., that imperfect transmittance was due only to reflections). The state of the art



H.11
Ive transfer in the window system of a double-glazed solar collector.

ad 1970 is described by the earlier review by Dietz (1963) and by the work
alas and Stephenson (1962), Stephenson (1965), and Whillier (1953,
l Whillier's review (1967), and the book by Duffie and Beckman (1974).
view of the radiative characteristics of a single semitransparent plate
wrformed by Viskanta and Anderson (1975). Siegel (1973) showed that
tht radiation method" was much easier to employ for finding the overall
tnittance of a stack of parallel semitransparent plates than the older
rcing technique. This method was employed by Shurcliff (1973) for the
c nonabsorbing covers. Sharafi and Mukminova (1975) and Viskanta
aylor (1976) obtained solutions for multilayer systems with varying
o properties, such as with short-wavelength radiation antireflective and
ind reflective coatings. Wijeysundra (1975) included the determination of
thiation absorbed in each cover by solving a system of N equations, where

N is the number of the collector plates (plus an additional cover plate for the absorber). Edwards (1977) proposed and successfully used the "embedding technique," which is computationally simpler, for the solution of the same problem. Viskanta et al. (1978) presented a general analysis (using the net radiation method) to predict the spectral directional radiation characteristics of single and multiple plates of semitransparent material, with parallel optically smooth surfaces that can also be coated with one or more thin-film materials to achieve desired spectral selectivity. Scattering inside the glass was assumed to be negligible, and absorption in the plates was taken into consideration. Elsayed (1984) extended the last two studies by considering both the perpendicular and parallel components of the unpolarized incident radiation throughout the plate stack, rather than use the average plate properties for both parallel and perpendicular components as done in the previous studies. He found that the differences are small for one or two covers, but they increase with the number of covers, to about 25% for six covers. Morris et al. (1976) analyzed radiative transfer through thin-walled glass honeycombs used for convection suppression in collectors and found that high shortwave transmittance and low emittance can be obtained with this design. A good review of radiative behavior of windows, of window stacks relative to solar collector applications, and of window coatings was given in the book by Siegel and Howell (1981, pp. 718–747).

The earlier analyses assumed that the cover plates were opaque to infrared radiation, and they ignored infrared interactions (emission, absorption, reflection, and transmission) between the semitransparent plates of the stack. This could lead to serious errors, especially for plates that are significantly transmissive in the infrared. Lior et al. (1977) developed a computer program for the analysis of solar collectors and systems (SOLSYS) that included the infrared emissive, absorptive, and reflective interactions in the stack, but they still assumed the plates to be opaque in the infrared spectrum. More recent work by Hassan (1979), Hollands and Wright (1983), and Edwards and Rhee (1981) includes all of the infrared interactions in the analysis of the combined radiant/convective transfer in solar collectors. The analysis by Hollands and Wright (1983) allows the use of different radiative properties on each side of the plate and makes the assumption that each cover is radiantly grey in the wavelength range of interest (3–30 μm). Edwards and Rhee (1981) have made one step-further in allowing nongrey behavior. The additional rigor included in the last two studies results in more cumbersome expressions for the radiant transfer, but efficient computer algorithms have been developed for their solution.

Collectors with cylindrical glass envelopes interact with radiation in a dif-

ferent way from those with flat covers because of the difference in geometry. For example, the transmittance of a cylindrical envelope is smaller than that of a flat plate for normally incident insolation because the solar rays strike the cylindrical envelope at incidence angles of 0–90 deg while the flat plate incident angle is uniformly 90 deg. Perhaps the earliest analysis of this configuration (for evacuated single-cover cylindrical collectors) was performed by Felske (1979) who considered a configuration with an absorber plate spanning the diameter of the glass tube and running along its length (as in figure 4.3a). He determined the transmittance of the cylindrical cover in terms of the fraction of solar radiation striking the absorber but neglected absorption in the cover, change of ray angle due to refraction, and internal reflections, and did not analyze the absorber–cover radiant interactions. This work was extended by Garg et al. (1983), with similar simplifications to collectors with two concentric covers.

Saltiel and Sokolov (1982) used three-dimensional ray tracing (the algorithm, however, was not presented in the paper) to perform an optical and thermal analysis of a cylindrical evacuated collector, with a cylindrical semitransparent absorber placed eccentrically inside. Their analysis is significantly more comprehensive than the two described above in that they refrain from the above-listed simplifications. Still, they assume that the inner and outer cylinders are opaque in the infrared range and that the thermal radiation is at a single wavelength. A ray-tracing technique using the Monte Carlo method was developed as a computer program for analyzing cylindrical double-wall evacuated solar collectors by Window and coworkers (Window and Zybert 1981; Window and Bassett 1981; Chow et al. 1984).

Cellular structures, such as honeycomb panels, have been considered for placement between the absorber and the cover plate to suppress convection and reduce reradiative losses. Sparrow et al. (1972), Tien and Yuen (1975), Felland and Edwards (1978), and Symons (1982) studied the radiative transfer in such structures, and presented results that could be used in thermal design.

"*Thermal traps*," in which the solar radiation is converted into heat by absorption in a semitransparent solid or stagnant fluid, were proposed and analyzed in detail by Cobble and coworkers (Cobble 1964a, b; Safdari 1966; Pellette et al. 1968; Lumsdaine 1970), and the results were validated experimentally (Cobble et al. 1966; Lumsdaine 1969). More recently both experimental and theoretical studies were performed by Abdelrahman et al. (1979) and Arai, Hasatani, and coworkers (see Arai et al. 1980, 1984; Bando et al. 1986) on volume heat trap solar collectors by using absorption in semitransparent fluids, in most cases containing in suspension absorbing fine particles. Absorption coefficients and transient temperature profiles were

obtained by a simultaneous solution of the radiative-conductive problem, using a multiband model for the radiative properties.

Progress Summary *The primary progress made was in (1) relaxing most of the simplifying assumptions—for example, by including the transmission of infrared radiation through the covers and infrared radiation exchange among the covers—by allowing spectral dependence of the properties, and by being able to analyze the effects of optical coatings on the semitransparent covers and thus extending the ability to consider a variety of materials and techniques for improving the transmittance of insolation and reducing the heat and radiative losses; (2) developing efficient algorithms for the solution of these more complicated problems; (3) formulating and solving the radiative problem of cylindrical evacuated collectors; (4) proposing, and advancing the state of knowledge on, "volume heat trap" collectors consisting of fine-particle suspensions in fluids.*

4.2.2 Natural Convection

The Basic Phenomena In many types of collectors and receivers air is contained between the hotter absorber and the cooler ambient. The temperature difference causes natural convection of the air, creating greater heat losses from the absorber to the ambient than would have taken place if the air were stagnant and the heat loss were by conduction only. In solar collectors the insulating air is typically confined in an enclosure formed by the sides of the collector, one or two of the window panes, or the absorber plate. If convection suppression partitions (or honeycomb structures) are used in the collector, the convection occurs in the volume enclosed by these partitions and by the window panes or absorber. Some cavity receivers in concentrating collectors have an open solar radiation entance aperture, and because of this opening on one or more walls, the natural convection enclosure is incomplete.

Relatively little was known about natural convection in solar collectors at the early 1970s. Many still used the dimensional empirical equation proposed by Hottel and Woertz (1942) and Whillier (1953) for the heat transfer coefficient h_{12}, between parallel plates 1 and 2, at temperatures T_1 and T_2 respectively:

$$h_{12} = c(T_1 - T_2)^{1/4} \equiv c\Delta T_{12}^{1/4}. \qquad (15)$$

Tabor (1958) summarized and consolidated a number of empirical results on natural convection between flat parallel plates that existed at that time (Mull and Reiher 1930; Robinson and Powlitch 1954); some of these correlations already included the effects of tilt and Grashof (or Rayleigh) number (Fishenden and Saunders 1950; de Graaf and Van der Held 1953, 1954), although they were in some conflict with each other. In addition these correlations have been obtained for configurations and boundary conditions that did not represent solar collectors well and were based on a relatively small number of data points. These equations were used through the early 1970s (see Duffie and Beckman 1974).

The relevance to the improvement of solar collector efficiency, the interest in the development and evaluation of methods for the suppression of convection, the serious paucity of necessary knowledge of the phenomena, and perhaps also the intrinsic interest in the scientific fundamentals of natural convection have combined to motivate a very large amount of research in that field over the last 15 years or so. This has resulted in significant improvements in understanding and in the ability to make quantitative predictions over a very large range of parameters. Some of that progress relative to solar energy applications has been summarized by Buchberg et al. (1976), Lior et al. (1983), Kreith and Anderson (1985), Hoogendorn (1985), Wang and Kreith (1986), and Lior (1986).

The state-of-the-art of natural convection in enclosures at the beginning and mid-1970s was reviewed by Ostrach (1972) and Catton (1978). Much work has been done till that time on the development of empirical heat transfer correlations and on the prediction of onset of instability in horizontal fluid layers heated from below, and analytical/numerical methods have been established for the solution of laminar two-dimensional natural convection problems in horizontal and inclined layers of infinite span. Aided by the advent of more powerful computers and motivated in part by solar energy applications, the development of effective computer programs for the analysis of three-dimensional laminar natural convection in rectangular, cylindrical, and spherical enclosures with arbitrary boundary conditions, and their experimental confirmation, was begun in the mid-1970s. This is exhibited in the publications of the research team of Ozoe, Churchill, Lior, and coworkers (see Ozoe et al. 1976, 1977a, b, 1978, 1979, 1981a, 1983a, 1985a; Chao et al. 1981, 1983a, b), Chan and Banerjee (1979a, b), and in the reviews by Ostrach (1982) and Lior et al. (1983). Some of the advances resulting from this work are highlighted below.

It is now understood more clearly that real natural convection flows in enclosures are not two-dimensional and that a velocity component parallel to the familiar roll-cell axis is also present. The flow thus resembles a double helix, with fluid particles moving along both the circumference of the roll cell and the direction of its principal axis—from the walls into the enclosure, up to a certain distance, and then back toward the walls. This third flow velocity component is due both to the drag at the end walls and to thermal gradients

generated at these walls because of the diminished rate of circulation. The significance of the three-dimensionality of the flow becomes even more pronounced when the enclosure is tilted or when partial partitions are inserted. It was determined that steady laminar natural convection in horizontal enclosures with a bottom temperature higher than the top is characterized by a train or roll cells with their axis parallel to the short side of the enclosure, an observation that also served to justify the many two-dimensional analyses of the phenomenon. This is no longer correct when the box is tilted: *Inclination*, about its longer side causes the axis of the roll cells to become oblique to the side, and at some critical angle all the roll cells form one large cell that has an axis perpendicular to the short side of the box. Tilting the box along its shorter side gradually merges the parallel roll cells into one large circulating cell, with its axis still parallel to the short side of the box. As the box is tilted from the horizontal position, the Nusselt number is first seen to decrease gradually to a minimum that coincides with the transition from one convective pattern to another, reaches a maximum at a higher angle of inclination (~ 60 to 90 deg), and then diminishes monotonically as the angle is increased to 180 deg. Both the minimum and maximum of the Nusselt number occur at slightly higher values for boxes inclined about the long side than for those inclined about the shorter side, but the values of Nu are about the same in both cases and similar to those of figure 4.12.

Three-dimensional calculations produce lower Nu values than two-dimensional ones, for the same case, principally because the three-dimensional calculations account for the slowdown and redirection of the circulation by the solid ends.

Computer time and memory are major stumbling blocks in the computation of natural convection in enclosures of practical size. The research team of Ozoe, Churchill, Lior, and coworkers (Ozoe et al. 1982, 1983b) used the observation that the convection occurs in roll cells confined to almost-fixed volumes in the enclosure to develop a new computational method that would reduce computer time and memory. In this method a number of typical (according to boundary conditions) cells are computed individually, and the solutions are then patched together to produce overall heat transfer coefficients for the enclosure. The results obtained for both horizontal and inclined enclosures were quite encouraging.

The fact that the minimal Nu number occurred at angles of inclination at which the roll cells were lined up with the longest axis, or the one along which the motion is most tortuous, indicated that the manipulation of roll-cell orientation by such means as *internal partial baffles* may result in the reduction of Nu. Numerical and experimental studies by Chao et al. (1983a, b) for a

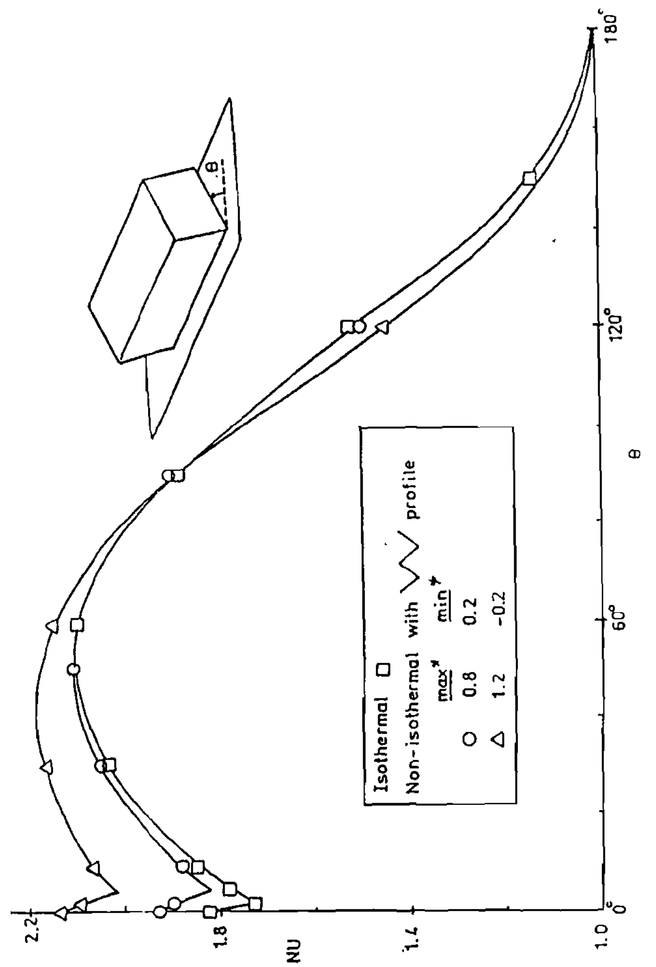


Figure 4.12
Mean natural convection Nusselt number across and enclosure (Ra = 6,000, Pr = 10) as a function of the inclination angle θ. The hotter surface has a double-sawtooth temperature distribution as shown. Source: Chao et al. (1981).

$2 \times 1 \times 1$ box with a partial fin perpendicular to the long side of the enclosure and attached to it, and for Ra up to 2×10^4, have shown that the fin produces a 30% reduction in Nu (at $Ra = 6 \times 10^3$) for all angles of inclination as compared to the finless box and that the dependence on angle of inclination is similar to that in figure 4.12. Similar effects of partial partitions were shown by Lin and Bejan (1983) for $10^9 \leq Ra \leq 10^{10}$ and by Nansteel and Greif (1984) for $10^{10} \leq Ra \leq 10^{11}$.

It is well-known that natural convection phenomena are quite *sensitive to the boundary conditions*. Three-dimensional computations, which in some cases were validated in by experiments (see Chao et al. 1981; Ozoe et al. 1983c), have shown that the Nusselt number decreases as the temperature of the hot plate becomes more uniform: For example, figure 4.12 shows the results of computations for a double-sawtooth temperature distribution on the lower surface (to simulate the effect of cooling pipes along the absorber), where Nu is up to about 20% higher for the nonuniform boundary condition investigated.

Although it is obvious that *radiative exchange among the enclosure surfaces* in a solar collector may have an important role in the determination of the boundary conditions and consequently on the natural convection, accelerated progress on the understanding of the interactions between radiation and natural convection was made only in the last few years (see reviews by Viskanta 1982 and Yang and Lloyd 1985). Except for a few special cases. such as receivers containing solid suspensions, the gas in solar collectors is non-participating. This fact simplifies the analysis significantly. Analytical studies by Edwards and Sun (1971) and Hatfield and Edwards (1982) have determined that radiation increases the critical Rayleigh number for the onset of convection because it tends to smooth out temperature differences on the surfaces. In considering both wall conductance and radiation, Kim and Viskanta (1983, 1984) pointed out that Nu is not representative of the overall heat transfer across the enclosure and that, for their conditions, radiant exchange tended to reduce the intensity of natural convection.

Natural convection in solar collector or receiver enclosures is often *turbulent*. Due both to uncertainties in the modeling of turbulence in this case and to numerical obstacles, solutions of turbulent natural convection in enclosures are still in their infancy and are mostly for two-dimensional models (see Fraikin et al. 1980; Farouk and Guceri 1982; Markatos and Pericleoeus 1984; Ozoe et al. 1985b). Hjertager and Magnussen (1977), and more recently Ozoe et al. (1986), developed and solved a three-dimensional model for the turbulent case. The models use the $k - \varepsilon$ formulation that was originally developed for forced convection, and there is thus still uncertainty about the values of the coefficients to be used for turbulent natural convection (for an analysis of sensitivity to model coefficients, see Ozoe et al. 1985b).

More detail on natural convection in enclosures is available in the recent reviews by Hoogendorn (1986) and de Vahl Davis (1986). A summary of results specific to typical solar collector configurations is given below.

Natural Convection between Parallel Plates (Large Aspect-Ratio Enclosures)
The interest in solar collectors prompted the rexamination of older correlations on natural convection between parallel plates and the development of improved and more appropriate empirical correlations and theoretical anayl-ses, which also include effects of inclination (see Hollands and Konicek 1965; Arnold et al. 1976; Hollands et al. 1976).

Elsherbiny et al. (1982) conducted a comprehensive experimental investigation of the heat transfer in air-filled, high aspect ratio enclosures with isothermal walls and produced results that at this time are recommended for use in thermal design of collectors. Their experiments covered the ranges $10^2 \le Ra_L \le 2 \times 10^7$, $5 \le H/L \le 110$ (H is the length along the inclined side of the collector and L the distance between the two plates) and $0 \le \phi \le 90$ deg, where ϕ is the angle of the enclosure axis with respect to the horizontal. They found the transition from the conduction to convection regimes in vertical enclosures to be a strong function of aspect ratio when $H/L < 40$. The recommended heat transfer correlations for vertical layers and enclosures are as follows:

For vertical layers ($\phi = 90$ deg),

$$Nu_1 = 0.0605\, Ra_L^{1/3}, \tag{16}$$

$$Nu_2 = \left[1 + \left\{\frac{0.104 Ra_L^{0.293}}{1 + (6310/Ra_L)^{1.36}}\right\}^3\right]^{1/3}, \tag{17}$$

$$Nu_3 = 0.242(Ra_L/AR)^{0.272}, \tag{18}$$

$$Nu_L = [Nu_1, Nu_2, Nu_3]\max = Nu_{90}. \tag{19}$$

For inclined layers ($\phi = 60$ deg),

$$Nu_1 = \left[1 + \left\{\frac{0.0936 Ra_L^{0.313}}{1 + G}\right\}^7\right]^{1/7}, \tag{20}$$

$$G = \frac{0.5}{[1 + (Ra_L/3160)^{20.6}]^{0.1}}, \tag{21}$$

$$Nu_2 = \left(0.104 + \frac{0.175}{AR}\right) Ra_L^{0.283}, \tag{22}$$

$$Nu_L = [Nu_1, Nu_2]\max = Nu_{60}. \tag{23}$$

The notation used in equation (19) and (23) indicates that the correlation with the maximum value should be used.

For tilt angles between 60 deg and 90 deg, Elsherbiny et al. (1982) suggest a linear interpolation between the limiting correlations given above:

$$Nu_\phi = \frac{(90\,\mathrm{deg} - \phi)Nu_{60} + (\phi - 60\,\mathrm{deg})Nu_{90}}{30\,\mathrm{deg}}. \tag{24}$$

For tilt angles between 0 and 75 deg, Hollands et al. (1976) recommend the correlation

$$Nu_L = 1 + 1.44\left(1 - \frac{1708}{Ra\cos\phi}\right)^{\cdot}\left[1 - \frac{1708(\sin 1.8\phi)^{1/6}}{Ra\cos\phi}\right] + \left[\left(\frac{Ra\cos\phi}{5830}\right)^{1/3} - 1\right]^{\cdot}. \tag{25}$$

where L is the distance between the plates at temperature T_1 and T_2, respectively, and the Rayleigh number Ra is given by

$$Ra = \frac{2g(T_1 - T_2)L^3}{\nu^2(T_1 + T_2)} Pr \tag{26}$$

and

$$[X]^{\cdot} \equiv \frac{|X| + X}{2}.$$

It should be noted that when $Ra < 1708/\cos\phi$, the Nusselt number in equation (25) is exactly equal to unity. Since by definition

$$q_c = Ah(T_1 - T_2) = A\,Nu\frac{k}{L}(T_1 - T_2), \tag{27}$$

the condition $Nu = 1$ implies that the heat transfer is by pure conduction.

Natural convection in cylindrical enclosures pertians primarily to partially evacuated collectors, line-focus concentrator receivers, and CPC collectors. Kuehn and Goldstein (1976a, b, 1978) developed a comprehensive correlation for the Nusselt number for natural convection between horizontal concentric (1976a, b, 1978) and eccentric (1978) cylinders at constant (but different) temperatures, as a function of the Prandtl and Rayleiyh numbers, for $10^2 <$ Ra $< 10^{10}$ and $10^{-2} <$ Pr $< 10^3$. Kuehn and Goldstein (1980) also presented a simplified correlation for Pr = 0.71 and laminar flow. Addressing natural convection between eccentric cylinders both numerically and experimentally, Lee et al. (1984) considered $10^2 <$ Ra $< 10^6$, diameter ratios of 1.25 to 5, and eccentricity ratios of up to ± 0.9 for air (Pr = 0.71). They illustrated the internal convective flow patterns and isotherms and have provided the overall heat transfer coefficient as a function of Ra.

Compound parabolic concentrators (CPC) are similar to line-focusing collectors. One of the differences is in a design proposing to enclose the entire concentrator trough, containing the receiver, with a front cover. This forms an irregular, noncircular cylinder. Natural convection in this device, oriented vertically, was computed for $2 \times 10^3 <$ Ra $< 1.3 \times 10^6$ by Abdel-Khalik et al. (1978). Meyer et al. (1980) determined the heat losses from a trough collector as a function of Ra and tilt angle. Some experimental results, summarized by Kreith and Anderson (1985), have also been obtained by a number of researchers.

Iyican et al. (1981) developed a correlation for natural convection and convective/conductive heat transfer in trapezoidal groove collectors.

Suppression of Natural Convection and Its Consequences As early as 1929 (see Veinberg 1959) honeycomb structures were placed in the air space between the absorber and the window cover in solar collectors to suppress natural convection and to reduce reradiated energy transfer to the ambient, and thus to reduce heat losses. It is obvious that such structures must not impede significantly the absorption of solar radiation by the absorber and that they should add minimal heat losses by conduction through their walls. Early work on this concept was done by Francia (1962) and by a number of researchers in France (see Perot et al. 1967). Perhaps the first analysis that considers both convection and radiation in such structures was performed by Hollands (1965), who used a rather simplified model. Tabor (1969) pointed out that the structures should be transparent to solar radiation and preferably opaque to infrared. To minimize cost, weight, and heat conduction, he recommended very thin materials that should also withstand the temperatures and the degradation due to insolation. Excluding glass from most applications because of cost and weight, he recommended the development of appropriate plastic materials.

Hollands (1973) performed experiments on natural convection in horizontal honeycomb panels (vertical cell axis), and he determined that the critical Rayleigh number for the onset of convection lies between those determined

for perfectly conducting and perfectly insulating side walls. Charters and Peterson (1972) brought up the important question of the critical conditions for inducing convection in such a honeycomb cell and commented that the fluid inside inclined honeycomb structures is always unstable, and thus no significant suppression effect can be expected in that configuration. It is worthwhile noting that (1) realistic boundary conditions, which exhibit nonuniform temperatures on the hot and cold walls and/or on the side walls, make the enclosed fluid always unstable, and there is essentially no critical Rayleigh number (i.e., $\mathrm{Ra}_{cr} = 0$ always) even for horizontal plates (vertical cell axis), and (2) the larger surface-to-volume ratio associated with "convection-suppression devices" nevertheless produces less vigorous convection and reduces heat losses due to convection; at low Ra numbers experiments indicated that the overall heat transfer coefficient is very close to conduction only.

Among the many types of convection-suppression devices developed and tested for solar collectors, successful results were obtained with 0.2-mm wall glass tubes (see Buchberg and Edwards 1976; McMurrin et al. 1977), and 0.076-mm-thick Lexan and Mylar (Marshall et al. 1976). "Bubble-sheet" packaging material was indicated to have potential if made of proper plastic materials, and aluminized plastic honeycombs, with their axis tilted toward the sun, that use reflection rather than transmission for channeling the solar energy to the absorber and for reducing reradiation were tested with reasonable success (Lior and Saunders 1973). Experiments by Symons and Gani (1980) have shown that flat collectors with a single antireflection etched low-iron glass cover, a convection-suppression device, and selective black absorber, performed up to 110°C better than commercially available cylindrical evacuated collectors configured with a specular reflector in the back. The advantage is realized primarily because the $(\alpha\tau)_{ef}$ of the cylindrical collector was significantly lower because of both the shape of the outer cover and its higher surface reflectivity.

For an inclined square honeycomb the Nusselt number depends on the Rayleigh number, the inclination (angle ϕ between the plates and the horizontal), and the aspect ratio of the honeycomb $AR = L/D$ (=depth/width). For the range $0 < \mathrm{Ra} < 600 \cdot \mathrm{AR}^4$, $30 < \phi < 90$ deg, and AR = 3, 4, and 5, the Nusselt number for air is given by Cane et al. (1977) in the form

$$\mathrm{Nu} = \frac{\bar{h}L}{k} = 1 + 0.89\cos(\phi - 60\ \mathrm{deg})\left(\frac{\mathrm{Ra}}{2420\mathrm{AR}^4}\right)(2.88 - 1.64\sin\phi). \tag{28}$$

This relation may also be used for hexagonal honeycombs if D is replaced by the hydraulic diameter. For engineering design the honeycomb should be chosen to give a Nusselt number of 1.2 according to Hollands et al. (1976)

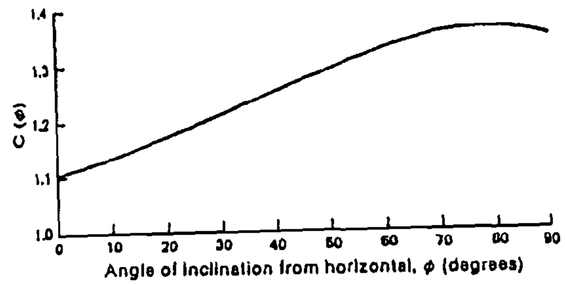


Figure 4.13
Function $C(\phi)$ for use in equation (29). Source: Hollands et al. (1976).

The optimum geometry is found from

$$AR = C(\phi)\left(1 + \frac{200}{T_m}\right)^{1/2}\left(\frac{100}{T_m}\right)(T_1 - T_2)^{1/4}L^{3/4} \tag{29}$$

if L is in centimeters and T in Kelvin. For air at atmospheric pressure and moderate temperatures, $280\ K < T_m < 370\ K$. The function $C(\phi)$ is plotted in figure 4.13. Empirical correlations for Ra_{cr} and Nu for inclined *rectangular-celled* diathermaneous honeycombs were further developed by Smart et al. (1980). They also concluded that square honeycombs are superior in most cases to rectangular ones.

Slats (rectangular enclosure with very large planar aspect ratios) placed along the east–west axis of the collector were also considered for suppressing natural convection. Meyer et al. (1979) found in small-scale laboratory experiments that convection was reduced for aspect ratios (distance between slats/depth of slats) below 0.5 and that convection heat transfer actually increased above the values observed for enclosures without stats for large aspect ratios, with a maximum occurring for aspect ratios of 1 to 2. Experiments with solar collectors using thin glass slats by Guthrie and Charters (1982) have confirmed that the slats improve the collector efficiency for normally incident insolation (by about 40% at 100°C) but that solar transmittance is reduced significantly for other solar incidence angles.

It was determined that small gaps between the honeycomb panel and the absorber and top glass cover do not affect the convection suppression capacity of the panel (Edwards et al. 1976). Demonstrating the strong coupling that exists between radiative transfer and the heat conduction in honeycombs, Hollands et al. (1984) have shown that an analysis that decouples the modes may severely underpredict the real heat transfer rates across the honeycomb panel. The radiation tends to increase the temperature gradients in the gas at the hot plate. An acceptably simplified analytical method that considers the coupled problem has also been presented. In addition Hollands and Iynkaran (1985) observed that conduction through the air layer next to the hot plate raises the temperature of the honeycomb panel and thereby also raises the reradiated energy loss from the panel. The use of such a panel thus tends to diminish the improvement one may expect from having a selectively coated (low emissivity) absorber in the collector. To take best advantage both of the reduction of radiative losses by the use of low emittance coatings and of the reduction of convective losses by using cellular convection suppression structures, they recommended and tested a configuration in which the honeycomb was separated by a 10-mm air gap from the hot plate, thus reducing the coupling between conduction to and radiation from the honeycomb panel. Such a collector was built and tected, and its improved performance was demonstrated (Symons and Peck 1984).

Progress Summary *Motivated in large part by solar energy applications, researchers in the past decade have made enormous progress in understanding natural convection in enclosures of various tilt and aspect ratios and in the ability to predict it. Their methods were both experimental and numerical. The new numerical techniques they developed allow the solution of three-dimensional problems for moderate aspect ratios and laminar flows. This solution has begun to be extended to turbulent flows, using the k — ε model. To day the effect of convection-suppression devices is reasonably well understood and predictable, and much progress has also been made in understanding the coupling between conduction and radiation in such devices and enclosures.*

4.3 Heat Transfer in the Absorber and the Heated Fluid

4.3.1 The Plate Tube Absorber

The absorber of many liquid-heating flat plate collectors consists of a number of parallel tubes (risers) connected at the inlet to a distribution manifold and at the outlet to a collection manifold, and attached with good thermal contact to the absorber plate. This is shown in figure 4.1, and in better detail in figure 4.14. Many cross-sectional shapes of the tubes, and methods for their attachment to the absorber plate, have been used.

The solar energy converted to heat at the absorber surface S is conducted through the absorber and the tube walls into the fluid that is flowing through

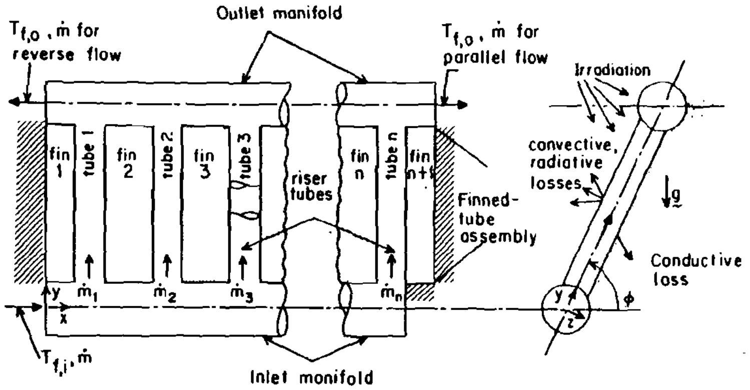


Figure 4.14
Plate-tube absorber manifold assembly geometry.

the tubes and that is thus being heated. The conventional approach for determining the heat gain of the fluid was established by the Hottel-Whillier (HW) model, in which the plate-tube assembly is modeled as the well-known fin-tube problem (figure 4.15). In that model it is assumed that for each differential element of length along the tube, the temperature distribution in the plate is one-dimensional, in the x-direction perpendicular to the tubes, and any gradients in the y-direction along the tubes and z-direction perpendicular plate are assumed to be zero. It also assume that the temperature profile is symmetric about the tube, with the tubes's centerline at the minimum temperature point and the fin's centerline (at $w/2$) at the maximum temperature. Performing an energy balance with this model for a unit length of the fin-tube (see Duffie and Beckman 1980), it is easy to show that the amount of heat collected by the fin q'_{fin} is

$$q'_{fin} = (w - d)\eta_f[S - U_p(T_b - T_a)], \tag{30}$$

where η_f is the fin's efficiency, S is the radiative flux absorbed per unit of absorber, U_p is the overall heat transfer coefficient from the absorber to the ambient, and T_b is the fin's base temperature (at the junction with the tube). Although typically the absorber plate (fin) is of uniform thickness, fins thicker at the root are known to have higher efficiency. Kovarik (1978) developed a method to determine the optimal profile of the fin in solar collector, based on an objective function of minimum cost per unit of heat output.

Since the tube also collects energy directly over its projected area d, the total useful energy gain per unit length of tube is

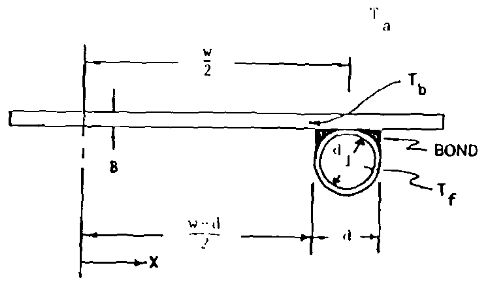


Figure 4.15
The basic fin-tube schematic.

$$q'_u = [(w - d)\eta_f + d][S - U_p(T_b - T_a)]. \tag{31}$$

That useful heat gain is transferred to the fluid, and the model expresses this in terms of the sum of the convective resistance between the fluid and the tube wall, and the contact resistance due to the bond between the tube and the plate (fin)

$$q'_u = \frac{T_b - T_a}{(1/h_{f,i}\pi d_i) + (1/C_b)}, \tag{32}$$

where $h_{f,i}$ is the heat transfer coefficient between the tube wall and the fluid, d_i is the internal diameter of the tube, and C_b is the bond conductance (detailed discussion of the effect of C_b can be found in Whillier 1964b and Whillier and Saluja 1965).

Another energy balance is made along the tube, equating the q'_u with the increase of enthalpy of the heated fluid for that length, which is expressed by the equation

$$\dot{m}c_p\left(\frac{dT_f}{dy}\right) - nwF'[S - U_p(T_f - T_a)] = 0, \tag{33}$$

where $\dot{m}$ is the overall mass flow rate through the collector, n is the number of parallel tubes, c_p is the specific heat of the fluid, T_f is the local temperature of the fluid, and F' is the "collector efficiency factor." The collector efficiency factor is indentified from equations (30) through (33) as

$$F' = \frac{1/U_p}{w\{[1/U_p(d + (w - d)\eta_f)] + (1/C_b) + (1/\pi d_i h_{f,i})\}}. \tag{34}$$

It is usually assumed at this point that U_p and F' are independent of position, and equation (33) is integrated to yield an expression for the temperature rise of the fluid as a function of distance along the tube and the other parameters (which are assumed to be constant):

$$\frac{T_f(y) - T_a - (S/U_p)}{T_{f,i} - T_a - (S/U_p)} = e^{-(U_p n w F' y/\dot{m} c_p)}, \tag{35}$$

where $T_{f,i}$ is the fluid temperature at the inlet to the tube.

Equations (33) and (35), together with the assumption of constancy of the heat transfer coefficients and F' along the tube (y-direction), indicate that the plate temperature must also vary exponentially with y.

Collector efficiency can now be expressed in the conventional way, using the "collector heat removal factor" F_R, which is defined as

$$F_R = \frac{\dot{m} c_p}{A_w U_p}[1 - e^{-(A_w U_p F'/\dot{m} c_p)}], \tag{36}$$

and the collector efficiency [see also equations (10) through (12)]

$$\eta = \frac{q_u}{A_w I_c} = F_R\left[(\alpha\tau)_{ef} - \frac{U_p(T_{f,i} - T_a)}{I_c}\right]. \tag{37}$$

Abdel-Khalik (1976) developed an analytical model of an absorber that has a serpentine tube bonded to the plate and solved it for two segments of the serpentine. He concluded that a general equation can be projected for the calculation of the heat removal factor F_R for any number of segments, with small error.

Zhang and Lavan (1985) extended the solution to four segments of the serpentine and that extrapolation from the two-segment solution can lead to much larger errors than predicted by Abdel-Khalik (1976). Either of these solutions calculates the heat transfer only for the straight parts of the serpentine, ignoring its U-bend portions, and assumes essentially one-dimensional heat transfer in the plate.

Equations (30) through (35) serve not only to show the conventional method of calculating heat transfer through the absorber but also to highlight the critical assumptions made in the model. In addition to the simplifications mentioned already, several others stand out: (1) U_p is actually a complicated combination of convective and radiative heat transfer between the absorber and the ambient, as discussed in sections 4.1 and 4.2; to say the least, it is not constant, and it depends on the temperature, in a nonlinear manner at that. (2) $h_{f,i}$ is not constant either; it depends strongly on the location along the tube if the internal flow is developing, on the temperature because the convec-

tion is often mixed (forced and natural), and because of the properties. (3) The temperature might not be uniform through the thickness of the fin or the tube, especially if low conductivity materials are used. (4) The plate's temperature field perpendicular to the tube may not be symmetrical because of the effects of unequal flow though the parallel tubes and edge effects. (5) The temperature T_b at the base of the fin might not be equal to the temperature at the interior diameter of the tube nor might the latter be constant around the internal circumference of the tube (obviously the top is heated and the bottom is not). To be able to understand collector operation better and to design more efficient and economical units, research during the last decade has examined many of these simplifications, and the results are summarized below.

Rao et al. (1977) solved anlytically the two-dimensional fin-tube problem, still assuming constant heat transfer coefficients, and concluded that the HW model results are accurate enough for the design of conventional flat plate solar collectors but not for collector design optimization. Chiou (1979, 1980) solved the two-dimensional problem numerically and also found excellent agreement with the HW model. Taking an analytical approach to the heat conduction problem in the absorber, Phillips (1979) noted that heat is actually conducted along the absorber in a direction counter to the flow of the heated fluid, reducing the amount of heat transferred to the fluid and thus collector efficiency. He found that the HW model therefore predicts efficiencies that are too high, by up to about 30% in the range of the parameters he considered.

Typically it is assumed that the flow of the heated fluid through each of the parallel risers is the same, and this is indeed desirable: The studies by Chao et al. (1981) and Jones and Lior (Jones 1981; Jones and Lior 1987) indicate that a uniform temperature absorber produces a higher efficiency collector than one with a nonuniform temperature. At the same time it can not be taken for granted that the flow is distributed equally through the risers: The dual manifold system must be designed to meet that objective. Few studies of flow distribution in such collector manifold systems were made. The earliest proposed for solar collectors was by Dunkle and Davey (1970) who have established, and solved analytically, a highly simplified flow model with a continuous slit (instead of discrete risers) distribution and collection inertia-dominated manifolds. Bajura and Jones (1976) have treated the inertia-dominated dual-manifold system with discrete risers. Jones and Lior (1978), Jones (1981; see also Jones and Lior 1987), Menuchin et al. (1981), and later Hoffman and Flannery (1985) included both inertial and frictional effects in the analysis of dual-manifold systems. Apart from having established a method to compute flow distribution in such a manifold system, it was suggested that essentially uniform water flow distributions are obtained if the riser-to-manifold tube diameter ratio d_r/d_m is $\frac{1}{4}$ or less in the range of flows pertinent to collectors.

Jones and Lior (Jones 1981; also in Jones and Lior 1987) examined the effects of flow maldistribution on collector efficiency using a three-dimensional conjugate model of an unglazed collector (see subsection 4.3.3), and they found negligible influence ($<2\%$) in the range of $\frac{1}{4} < d_r/d_m < \frac{3}{4}$ and pertinent flat plate liquid-heating collector parameters. Solving a two-dimensional collector model numerically and examining arbitrarily imposed flow maldistributions (in most cases of much larger magnitude than those found by Jones and Lior), Chiou (1982) found reductions of 2%–20% in collector efficiency due to flow maldistribution.

Progress Summary *The primary progress in this well-trodden area was in advancing from the one-dimensional absorber plate–tube models to two-dimensional ones, in the ability to determine flow distribution among the risers in a more correct way, and in examining the effect of maldistribution on absorber heat transfer. It was confirmed that collector efficiency improves somewhat as the absorber temperature becomes more uniform.*

4.3.2 Convection to the Heated Fluid

Much is known about convective heat transfer in conduits (see Shah and London 1978; Kreith and Kreider 1978), and when judiciously applied, this available information can be directly used in the thermal analysis and design of solar collectors.

Three aspects related to the proper choice of the convection correlation or to the formulation of the analytical/numerical problem that may be important in solar collectors are flow and thermal development, nonuniformity of the boundary conditions on the interior surface of the conduit tube), and buoyancy effects on the convection (existence of mixed convection).

Usually the flow rate in conduits of liquid-heating collectors is very low, and the flow is consequently laminar and possibly developing for a fair fraction of the tube length. This should be examined by applying one of the conventional criteria for flow development before a decision is made on the correlation or analytical method to be used. It should, however, be noted that most of these criteria were established for constant temperature or heat flux boundary conditions, and without consideration of buoyancy effects—that is, for conditions that do not represent the situation in collectors exactly. For example, Lior et al. (1983) computed that in mixed convection in a vertical tube with linearly increasing wall temperature the Nusselt number is 28%–40% higher than that for the constant wall temperature case, and that flow development

is different too. Morcos and Abou-Ellail (1983) examined numerically buoyancy effects in the entrance region of an inclined collector composed of parallel rectangular channels with realistic boundary conditions, and found Nusselt numbers up to 300% higher (at $Ra = 10^5$) than those predicted without the inclusion of buoyancy effects.

Cheng and Hong (1972) analyzed numerically the case of mixed laminar convection in uniformly heated tubes of various inclination and determined that both friction factor and Nusselt number increase significantly with the inclination angle. Baker (1967) observed that augmented mixed convection would occur in solar collector tubes due to variations in the circumferential temperature and that the heat transfer coefficients (for $370 < Re < 2{,}700$ laminar flow in horizontal tubes) were about 10% higher than those for tubes with circumferentially uniform temperature. Such augmentation becomes particularly important when the heating is from below. In turbulent horizontal flow, on the other hand, circumferentially nonuniform heating appears to increase the thermal development length but to have either essentially no effect on the average Nusselt number (see Black and Sparrow 1967; Schmidt and Sparrow 1978; Knowles and Sparrow 1979) or a small opposite one, seen to reduce it by up to 20% for heating from below (Tan and Charters 1970). The reduction in heat transfer due to buoyancy effects may result from the tendency to relaminarize the flow. This is also consistent with the general conclusion of several of these researchers that the highest local convective coefficients are encountered at the least heated circumferential positions, and vice versa.

For mixed convection in a tube attached to an absorber plate, with conditions typical to flat plate solar collectors, Sparrow and Krowech (1977) and Jones (1981) concluded from analysis that circumferential variations in the thermal conditions of the tube can be neglected.

As seen from equations (36) and (37), increasing the overall flow rate through the collector improves its efficiency and the rate of heat collection. At the same time the efficiency improvement is an asymptotic function of the flow rate; increases of flow rate require more energy and capital equipment investment in pumping, and they increase the operating pressure in the collector and balance of system. Hewitt et al. (1978) and Hewitt and Griggs (1979) proposed a method, based on economic optimization, for determining the optimal flow rate through liquid- and air-heating collectors (but they have not considered possible implications of the pressure increase in the system, which is required for increasing the flow). Optimal control strategy of mass flow rates in flat plate solar collectors was determined for several combinations of objective functions and system models by Kovarik and Lesse (1976)

Progress Summary *Much work was done and important progress was made in providing information on convection with flow inside inclined tubes, with circumferentially varying thermal boundary conditions, flow development, and buoyancy effects. Ways to determine collector flow rate based on economic considerations were established.*

4.3.3 Formulation and Solution of the Overall (Conjugate) Thermal/Flow Behavior of the Collector

Many years of experience have shown that the Hottel-Whillier thermal model for flat plate solar collectors is adequate for designing conventional collectors and for estimating their performance; experience has also shown that overall collector efficiency is not too sensitive to most of the design parameters when perturbed around a base-case conventional design system definition (see also section 4.10). Realizing, however, that numerous extreme simplifications are inherent in this model (many of which are described in this chapter), it is clear that collector optimization and the development of new collector designs require more rigorous thermal/fluid models. To avoid the need for specifying approximate or arbitrary boundary conditions for each thermal subproblem of the overall collector problem—namely, the radiative and thermal transfer in the window/absorber system, natural convection in the window system, conduction in the absorber, flow distribution in the collector, and convection to the heated fluid—and thus to avoid adding to the solution error, the conjugate heat problem describing the entire collector should be solved. In the conjugate solution all of the subproblems are solved simultaneously, with one serving automatically as the boundary condition for the other.

A comprehensive three-dimensional computer program and effective solution technique were developed for this purpose by Jones (1981; see also Jones and Lior 1987). The specific example solved was a 1.5 × 2 to 6 ft unglazed solar collector with a dual-manifold system containing four risers. The problem was divided into three subproblems: (1) dual-manifold system hydrodynamics, (2) radiant-conductive finned-tube heat transfer, and (3) riser tube fluid dynamics and heat transfer. Each subproblem was solved numerically, and the resulting system of equations was solved simultaneously using an iterative scheme. The solution is refined with each cycle of the iteration since any one subproblem is solved subject to boundary conditions that result from the most recent solutions of the remaining two.

The subproblem models and solutions are fairly general but oriented to solar collector conditions. For example, the solution to the mixed convection heat transfer and fluid mechanics problem in inclined collector risers was

bounded by the one providing the lowest Nu, occurring in the case of horizontal tubes without buoyancy effects, and by the one providing an upper limit for Nu, corresponding to vertical tubes with buoyancy effects included. Flow development was included in the analysis, and the effects of buoyancy on heat transfer and flow distribution were established. Notably, in comparison to the case where buoyancy is neglected, the effects of buoyancy on flow in the this system indicate a maximal (1) 5% increase in riser Nusselt number, (2) 24% decrease in flow maldistribution fraction, and (3) 38% reduction in overall dual-manifold pressure drop.

Solution of the conjugate problem was found to provide remarkable insight into the behavior of the collector and gave quantitative relationships among the different components and the operating conditions. It also confirmed the fact that the efficiency of reasonably well-designed conventional flat plate collectors can be predicted by the HW model to within 3% of the value obtained from the significantly more elaborate conjugate model.

Morcos and Abou-Ellail (1983) developed a partially conjugate numerical model of an inclined solar collector with parallel rectangular flow channels in which they considered mixed developing laminar convection in the channels, with circumferentially nonuniform thermal conditions on the channel walls; the latter conditions were determined through simultaneous solution of the channel wall conduction problem. They found that entry length is reduced as the Rayleigh number increases and that buoyancy serves to increase the Nusselt numbers over those predicted with buoyancy neglected.

As an alternative to the solution of the complicated conjugate problem once it was realized that the thermal behavior of the collector is a nonlinear function of the temperature, attempts were made to improve on the conventional HW linear relationship between efficiency and temperature difference by developing and examining nonlinear relationships. Cooper and Dunkle (1981) developed a nonlinear model in which three nondimensional groups were added, but they determined that little improvement was obtained over the conventional linear model in characterizing the daily performance. Phillips (1982) also developed a nonlinear model, with coefficients determined empirically, and found that it correlated simulated collector data better than the linear model. Comparison with collector test data was, however, not made. It should be noted, in summary, that it is already common in collector testing to express the efficiency in terms of a quadratic polynomial in $(\Delta T/I)$.

Progress Summary *For thorough analysis of the thermal-fluid behavior of collectors and collector components, a conjugate flow distribution and heat transfer model was developed, and a solution technique was developed and used* *successfully. The model does not yet include natural convection in the insulating spaces as one of the subproblems. Nonlinear lumped-system collector models were developed and shown to represent collector efficiency a little better than the linear HW model.*

4.3.4 Plate Absorbers with Flow over Their Entire Surface

To reduce sealing complexity and thus the cost of the collector, liquid-heating collectors typically channel the liquid flow through discrete tubular conduits attached to the absorber plate, as described in subsection 4.3.1. This reduced flow area also increases the velocity of the liquid, and consequently the heat transfer coefficients between the tubes and the liquid. Because of the relatively low heat transfer coefficients between solids and gases and because leakage of air is somewhat more tolerable than leakage of liquids, the heated air in air-heating solar collectors is exposed typically to the entire area of the absorber and is not channeled through a number of discrete small conduits (figure 4.2). The thermal analysis (see Whillier 1964a and a more recent detailed review by Gupta 1982) is very similar in principle to that for plate-tube absorbers (subsections 4.3.1 and 4.3.2), with the main difference in the fact that absorber-to-fluid heat transfer coefficients are calculated for plates instead of tubes.

Going beyond the conventionally used lumped-system energy balance solutions, Liu and Sparrow (1980) investigated numerically the effects of radiative transfer in a shallow and wide rectangular airflow channel with black walls and a configuration consisting of the top plate heated and the bottom insulated, as is used in many solar collectors in which the air to be heated flows below the absorber (see figure 4.2b). They found that radiative transfer from the hotter to the cooler plate is significant in the developed laminar flow region in that it allows the lower plate to transfer by convection up to 40% of the heat to the air while reducing the temperature of the upper plate and consequently its losses to the ambient. A theoretical model of a similar configuration for turbulent flow was developed and validated experimentally by Diaz and Suryanarayana (1981), but they did not consider radiative exchange. The importance of the large heat transfer coefficients at the channel entry was confirmed. Diab et al. (1980) performed a two-dimensional analysis of a number of air-heating solar collector configurations by applying a nodal formulation. Apart from the value of the technique for this purpose, an important contribution of this study was to indicate, once again (see subsection 4.3.3), that the conventional single-module lumped system analysis predicts efficiencies that are within 5% of a more elaborate (here six-module) analysis. An integral solution for the single rectangular channel solar col-

lector, with fully developed flow in the channel, was obtained by Grossman et al. (1977), using a second-degree polynomial for the temperature profile, and by Naidu and Agarwal (1981) who used a more correct, fourth-order, polynomial.

To improve the thermal performance of air-heating collectors, a number of designs that increase the contact area between the air and the absorber, or the heat transfer coefficients, or both, have been proposed and analyzed. The effects of such improvements on collector cost and needed pumping power were examined and constrained to acceptable levels in the designs that were recommended for development and production. Many of these designs involve a rather complex and different flow geometry. It would not be practical to present general results, but some of the main studies will be mentioned for reference.

An analysis of the *overlapped glass plate* air heater (figure 4.2d) proposed by Miller (1943) and developed by Löf et al. (1961) was formulated and performed by Selçuk (1971). Christopher and Pearson (1982) produced an analysis of convective heat transfer (by solving the two-dimensional continuity, momentum, and energy equations) in an air heater with *overlapped opaque louvers*.

Analyses of air heaters with *finned absorbers* (figure 4.2e) were performed by Bevill and Brandt (1968), Cole-Appel and Haberstroh (1976), Cole-Appel et al. (1977), and Youngblood and Mattox (1978), among others. Fin dimensions, spacing, and surface radiative properties were some of the parameters considered for improving the collector performance. *V-corrugated* absorbers (figure 4.2f), with air flowing in the V-channels below the absorber, were analyzed by Hollands (1963) and Shewen and Hollands (1981), and comparisons were made with rectangular-channel flow passages under flat plate absorbers. Short flow passages were recommended to take advantage of the high heat transfer coefficients in the developing flow, and the V-corrugated design was found to produce a more efficient collector as compared to the flat plate absorber design for the same pressure drop and airflow rate.

Passage of the air through an absorber formed of a *porous matrix* (figure 4.2g), as proposed by Bliss (1955), increases the contact area significantly, and possibly also the heat transfer coefficients. Chiou et al. (1965) laid the foundation for the design and analysis of such collectors and determined friction factors for various slit-and-expanded aluminum foil matrices. Swartman and Ogunlade (1966), Beckman (1968), Lalude and Buchberg (1971), Buchberg et al. (1971) (the latter two studies proposed a honeycomb convection suppression device on top of the porous bed), and Lansing et al. (1979) proposed and laid the ground for the analysis of *packed and porous bed absorbers*. ...

impingement solar air heaters in which the heated air impinges via many small jets upon the back of the absorber have been studied by Honeywell (1977).

Progress summary *Some progress was made in the understanding of heat transfer in ducts with asymmetric boundary conditions, in advancing to two-dimensional models, and in better understanding of ways to enhance heat transfer between the fluid and the absorber.*

4.3.5 Heat Transfer to the Liquid in Solar Collectors for Boiling Liquids

Many applications require the generation of steam or vapor by solar energy, for example, for driving prime movers. Even if there is no need for steam or vapor, the high heat transfer coefficients associated with boiling in tubes have appeal for the improvement of collector efficiency. At the same time flow boiling in tubes is likely to introduce higher pressure drop than single phase flow and heat transfer, and the flow is apt to be less stable.

A few theoretical and experimental studies have been made on this subject. Experiments with boiling acetone and petroleum ether (Soin et al. 1979), fluorocarbon refrigerants such as R-11 and R-114 (Downing and Waldin 1980; Al-Tamimi and Clark 1983), butane (Bol and Lang 1978), and boiling water in a the tubular receiver of a line-focus parabolic trough solar concentrator (Hurtado and Kast 1984) have demonstrated both feasibility and improved heat transfer coefficients. All three of the reported analyses, for the plate-tube collector by Al-Tamimi and Clark (1983) and Abramzon et al. (1983) and for the cylindrical receiver of a line-focusing collector by May and Murphy (1983), essentially use the Hottel-Whillier equation and consider two regimes along the tube: first the subcooled regime along which the liquid is rising in temperature but not boiling yet, and the performance is evaluated with the conventional single-phase coefficients, followed by the boiling regime, downstream of the section at which saturation temperature was attained. The calculation incorporates the determination of the distance along the tube at which transition from the subcooled single-phase regime to the boiling two-phase regime occurs. The analytical approaches are fairly similar, differing only in the fact that Abramzon et al. (1983), and May and Murphy (1983) perform energy balances on small elements along the tube and integrate the equations along the same path, whereas Al-Tamimi and Clark (1983) use the available HW results, which already have been integrated for each of the two regimes based on the HW approach. Consequently Al-Tamimi and Clark (1983) proposed the same equation for the boiling collector as (37),

$$\eta_B = F_{BR}\left\{(\alpha\tau)_{ef} - \left[\frac{U_p(T_i - T_a)}{I}\right]\right\}, \tag{38}$$

in this case with the heat removal factor F_{BR} expressed as

$$F_{BR} = F_R F_B, \tag{39}$$

where F_R is the HW heat removal factor evaluated for single-phase conditions and the same flow rate used in the boiling collector [see equation (37)], and F_B is the correction to the heat removal factor to account for boiling:

$$F_B = \frac{1 - \exp(-az^*)}{1 - \exp(-a)} + \frac{(1 - z^*)\exp(-az^*)}{F_R/F'_B}, \tag{40}$$

where

$$a \equiv \frac{U_p F'}{(\dot{m}/A_w)c_{pl}}, \tag{41}$$

$$z^* \equiv \frac{L_{nb}}{L}, \tag{42}$$

L_{nb} is the distance along tube needed for the liquid to rise to the saturation temperature (determined from an energy balance on the liquid), L is the overall length of the tube, and F'_B is F' [equation (34)] for the boiling part of the tube. The boiling (internal) heat transfer coefficient was determined usually from the correlations by Chen (1966) and Bennet and Chen (1980).

Abramzon et al. (1983) further concluded that collectors with internal boiling come within 2%–3% of attaining "ideal" efficiency, that is, the efficiency that could be attained for internal heat transfer coefficients and Reynolds number approaching infinity.

Deanda and Faust (1981, AiResearch Mfg. Co.) have designed and developed an insulated, cylindrical coiled tube boiler that is mounted at the focal plane of a parabolic solar reflector. It was designed to perform as once-through boiler, with or without reheat, for generating steam for a Rankine cycle.

Progress Summary *Almost all of the work in this area was done during the last decade. As expected, practically ideal internal heat transfer coefficients (from tke collector efficiency standpoint) can be obtained if boiling is allowed inside the collector tubes. Experiments with a few fluids demonstrated this fact By adapting the HW model to this problem, a reasonable first step was made in the ability to predict boiling collector efficiency.* More work is needed, primarily *in determining possible adverse effects of increased pressure drop and flow instability and in determining boiling heat transfer and pressure drop for the boundary conditions specific to solar collector systems.*

4.3.6 Collectors with Free Flow and/or Direct Absorption of Radiation in the Heated Fluid

Improvements in heat transfer and in collector capital costs might be realized if the conduits for the heated fluid could be eliminated or simplified: Instead of irradiating the solid conduit (absorber) and then transferring the heat via conduction through the wall, and convection, to the heated fluid, the flowing fluid could be exposed to the solar radiation directly. If the fluid layer is highly absorptive to solar radiation (this is affected both by the opacity of the fluid and by the thickness of the layer), all of the radiation would be converted to heat as it passes through the fluid. Otherwise, the design would ensure that most of the radiation transmitted through the fluid layer be absorbed in the solid substrate on which the fluid is flowing. As the temperature of the substrate thus rises above that of the fluid, heat would be transferred from the substrate to the fluid by convection and absorption of the substrate radiosity. Several such configurations were evaluated, and the primary interest at present is in evaluating the applicability of these concepts to high temperature central-solar-tower-type receivers.

Free flowing collectors with at least some absorption in the substrate, such as the Thomason "Solaris Trickle Collector," were evaluated and analyzed by Beard et al. (1977, 1978), and the analysis was advanced a little by Vaxman and Sokolov (1985). The Solaris collector exhibited a problem that needs to be addressed in all free flow collectors that heat a liquid with a free surface: evaporation from the free surface with subsequent condensation on the inside of the transparent cover plate. This poses several difficulties, the primary being an effective transfer of heat from the liquid to the ambient, and reduction of cover transmittance due to condensation, both resulting in low collector efficiency. The use of liquids that have a very low vapor pressure at the operating temperature can alleviate this problem (see Beard et al. 1977).

A much more rigorous analytical approach that takes into consideration detailed (spectral, directional) radiative interactions in the liquid/substrate system was developed and validated by Arai et al. (1980), Hasatani et al. (1982), Bando et al. (1986), Wang and Copeland (1984), and Webb and Viskanta (1985). The latter two studies have also included the film fluid mechanics in the analysis. Webb and Viskanta (1985) also discovered that there exists a critical fluid layer opacity that yields the optimal radiation collection efficiency: Fluid layers that are less optically thick than this critical value are too thin to adequately absorb and transport the incident radiation; layers whose opacities are greater than the critical thickness absorb too strongly near the surface, and subsequent emission reduces their performance.

Fully absorbing 'black liquid' collector studies include those by Minardi and Chuang (1975), Trentelman and Wojciechowski (1977), Landstrom et al. (1978, 1980), Samanó and Fernandez (1983), and Janke (1983). Efficiencies 10%–15% higher than those of conventional plate-tube absorber collectors were observed experimentally. Landstrom et al. (1978, 1980) were also engaged in the development of a low-cost commercial collector made primarily from acrylic material. Black dyes dissolved or suspended in the heated liquid were used to produce the "black liquid." Stability problems were encountered, and an acceptable and proven "black liquid" needs yet to be found. A complete set of radiative properties of india ink suspensions of different concentration were measured as a function of wavelength by Wagner et al. (1980).

A radiation-absorbing fluid can also be produced by suspending very fine (often in the submicron size) absorbing solid particles in a flowing gas. This concept has been explored, both analytically and experimentally, for a "volume heat trap" collector by Arai et al. (1984) and recommended by Hunt (1978) for use with air as the working medium in a solar central receiver. Analysis and experiments by Hunt et al. (1983, 1985) have indicated that the gas assumes almost immediately the particle (magnetite was used) temperature if the particles are small enough (about 0.1 μm or less), and that very small concentrations of solids ($\approx 10^{-3}$ kg/m^3) are needed to achieve effective heating of the gas. It is noteworthy that the thermal and radiative exchange between the particles and the gas cannot be calculated by continuum theories because the particle size is of the order of the mean free molecular path of the gas (Yuen et al. 1986).

Another approach was taken by Sandia Laboratory (Hruby 1985) in which larger particles (sand-size, hundreds μm) fall in front of the receiver aperture and are thereby irradiated and heated. Rather than just serve to heat the air, these particles serve as the heat transfer medium and are also used as the medium for heat storage. The particle materials proposed were sintered bauxite for temperatures up to 1,000°C and doped fused zircon for higher temperatures. Analysis and experiments were being conducted.

Progress Summary *"Black liquid" collectors have been proposed and briefly tested, but little or no activity is evident in this area now. Promising work is proceeding on direct absorption of radiation in fine particles suspended (or falling) in air.*

4.3.7 Shading Effects

Whenever the solar rays are not normal to the collector, the collector side walls shade a part of the absorber. Considering typical flat plate collector configurations, the use of a "shading factor" s of about 0.97 was recommended by which to multiply the conventional collector efficiency prediction equation. In this way the thermal analysis of the collector and the prediction of its efficiency can be made, assuming that no shading occurs, and the final efficiency is simply multiplied by this shading factor.

The use of such a constant factor is imprecise, in that the factor depends on collector configuration, internal detail, orientation, location, and the diffuse fraction of insolation. It is obvious that deeper collectors will have a larger part of their area shaded by the side walls and will thus have a lower shading factor. This is of particular interest with the use of convection-suppression devices in front of the absorber. Furthermore, as Lior et al. (1977) remarked, the shaded areas still accept diffuse insolation, and both the shaded and unshaded parts of the absorber will also intercept some radiation through reflections and reradiations from interior surfaces of the side walls and from window panels. The amount of radiation absorbed in this way of course depends on the configuration and on radiative properties. In their analysis of partially shaded collector arrays, the shaded areas were computed as a function of time (and the solar incidence angle), and it was assumed that these areas accept only diffuse radiation, not the radiosity coming from other internal surfaces. Nahar and Garg (1980) developed the equation for the unshaded area fraction $(1 - s)$ of an equator-facing collector:

$$(1 - s) = 1 - \left(\frac{x_0 x + D_y - x_y}{x_0 D}\right), \tag{43}$$

where x_0 is the collector length, D is the collector width, and

$$x = d \tan \theta_t \sin \gamma_t, \tag{44}$$

$$y = d \tan \theta_t \cos \gamma_t, \tag{45}$$

where d is the depth of collector, θ_t the angle of incidence on the tilted equator-facing collector, and γ_t the azimuth angle of the tilted collector.

They also proposed a simplistic correction for diffuse radiation reaching the absorber due to reflections from the interior surfaces of the side walls. This correction may have the same order of magnitude of error as the conventional assumption of a constant shading factor. It should also be noted (Lior et al. 1977) that a shading factor like this, which simply uses the fractional unshaded area to multiply the overall equation for collector efficiency, is inherently in error (up to 500%, as shown in the computations performed by these authors) since it indiscriminately multiplies both the energy input and energy loss terms in the equation: Shaded areas indeed may collect less energy, but they continue

to lose it. This fact was included in the computer program SOLSYS developed by Lior et al. (1977).

Shading may also occur due to various objects between the sun and the collector, including other collectors (e.g., in an array configuration with more than one row of collectors). Computation of the position, shape, and size of shaded areas based on the geometry of the obstructions and of the target area (e.g., the collector's absorber surface) and on the position of the sun is well understood (see U.S. Post Office 1969; DOE-2 1981), although new and more effective techniques are being developed (see Budin and Budin 1982; Sassi et al. 1983). Jones and Burkhart (1981) developed analytically an extension of the Liu and Jordan (1961) model for insolation incident on a collector, for mutual shading by parallel rows of solar collectors, and pointed to an error in a previous analysis by Appelbaum and Bany (1979).

Progress Summary *The extent and effect of collector shading can now be predicted correctly. This also allows optimal spacing of collector arrays where the collector mounting area is constrained, by allowing partial shading during some periods while producing more heat overall than could be obtained from fewer rows (which are placed farther apart and are thus always unshaded).*

4.4 Heat Transfer through the Back and Side

4.4.1 Collector Insulation

As shown figure 4.9, the heat loss through the back and sides of the collector is governed by the resistance due to heat conduction through the insulation $R_{k11\text{-}13}$, which is also usually predominant, in series with the parallel combination of the convective resistance $R_{c13\text{-}a}$ and the radiative resistance $R_{r13\text{-}b}$ between the exterior surface of the collector back and sides and the ambient. It is common practice to neglect the resistances to the ambient relative to the one through the thermal insulation and to assume that the temperature difference driving both back and edge losses is the same: T_{11} (or T_7) $- T_a$. The calculation of heat loss is thus straightforward (see Gilleland 1980).

Although the above-described approach is often adequate for design purposes, the oversimplified model used does not allow precise optimization of insulation and does not address adequately collector designs and installations for which the simplifications do not apply. One obvious error, as Tabor (1958) also pointed out, is in the description of the edge losses due to heat conduction through the side insulation. The absorber plate is located perpendicular to the edge insulation and is not parallel to it, and the heat flow from the absorber

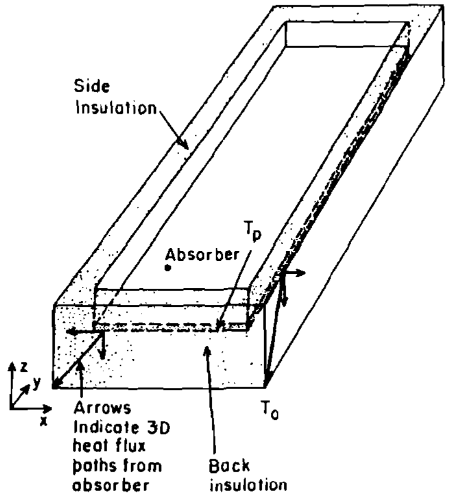


Figure 4.16
Three-dimensional conductive heat transfer from absorber through collector side- and back-insulation.

to the ambient is two-dimensional (three-dimensional in the corner regions), as shown in figure 4.16. Tabor (1958) computed correction factors to account for this and recommended that a good starting point for design is to specify the same thickness for the side insulation as selected for the back insulation.

Another error may arise due to the neglection of the convective and radiative exchange with the ambient. This may become important in areas where no wind is present at the collector and back edge (for the convective resistance) or where radiative exchange may become significant. The latter can occur if (1) the outer surface of the back and edges is at high temperature, either due to high absorber temperatures or smaller amount of insulation, (2) the temperature of the ambient surfaces (or sky) is relatively low, and (3) the temperature of the ambient surfaces and/or the albedo is high, thus actually acting to add energy to the collector through its back and sides.

Sateunanathan and Gandhidasan (1981) pointed out that for small angles of inclination, such as those used in low latitudes, natural convection in enclosures that are hot at the top and cold at the bottom is very small, and they have therefore recommended that the solid back insulation could be replaced with an air gap, preferably including a plate placed parallel to the absorber that would serve as radiation shield between the absorber and the back cover. Their experiments indicated that a collector with such an air gap

insulation performed perhaps even a little better than a collector with fibrous back insulation.

Jones and Lior (1979) developed an insulation design procedure for solar heating systems and presented optimal insulation thickness selection graphs based on a present-value life-cycle cost analysis.

4.4.2 Double-Exposure Collectors

Going an important step beyond the idea of eliminating the solid insulation from the back of the collectors (see the discussion in subsection 4.4.1), Souka (1965) recommended construction of a collector that is glazed on both sides, with the exposure of the back of the absorber to insolation reflected from a mirror placed behind the collector. This almost doubles the amount of solar energy incident on the same collector. His experiments, as well as those by others (see Savery et al. 1976; Savery and Larson 1978) indicated significant improvement in total energy collected. Souka and Safwat (1966, 1969) have also developed a simplified theoretical thermal model of such collectors and have made recommendations for the optimal orientation of the collector and of the mirrors.

Boosting of the solar radiation incident on the front of conventional single-exposure flat plate collectors with flat mirror reflectors has been the subject of a number of studies, beginning with Shuman's work on a solar water-pumping system in Philadelphia in 1911 (see Tabor 1966; McDaniels et al. 1975; Grassie and Sheridan 1977; Baker et al. 1978; Kaehn et al. 1978; Larson 1980a, b). The results were encouraging, and the techniques used for the optical analysis can be also adapted to the optimization of double-exposure collector-mirror systems.

Progress Summary *Almost half of the collector surface area is located at its back. The back can be just insulated to reduce heat losses, or it can be designed to even add to the heat input to the collector, such as is done in double-exposure collectors. Although three-dimensional conduction analysis in the back and side insulation would provide more precise information for insulation optimization than the currently used one-dimensional calculations, enough was already known from the practical standpoint, so not much progress was needed in thermal theory and modeling in this area, and indeed little was done. The replacement of solid insulation in the back with a suppressed-convection air space warrants an economical feasibility study.*

Exposure of the back of the collector to solar radiation reflected from mirrors has conclusively shown a marked improvement in the thermal performance of such a double-exposure collector, but it was not made clear yet whether the *additional costs associated with this installation, and the need to periodically adjust the mirror position and to maintain the mirror surface, can be justified.*

Heat Transfer in Partially Evacuated Enclosures

It was obvious at least from the beginning of the century, when Emmet (1911) of the General Electric Company patented a tubular evacuated solar collector module, that vacuum between the absorber and the cover would reduce heat losses due to both convection and conduction and thus would improve efficiency significantly. Renewed interest was expressed following the work by Speyer (1965), who built and successfully tested several variations of a tubular evacuated solar collector, and Blum (Blum et al. 1973; Eaton and Blum 1975), who proposed, built, and tested flat plate evacuated collectors. The 1970s saw vigorous development of many types of evacuated solar collectors (see Graham 1979), as shown in figure 4.3, and they also attained some market acceptance.

Aspects related to radiative transfer through the cylindrical cover to the absorber were discussed in subsection 4.2.1. Heat transfer in the absorber and the heated fluid were included in section 4.3. Overall performance analysis is described later in section 4.11.

Practically all of the analyses of evacuated solar collectors have assumed the existence of a perfect vacuum, that is, the absence of any conduction or convection in the evacuated space. This simplifies the analysis relative to that needed for non vacuated collectors and is correct for collectors that have been evacuated sufficiently to make these modes of heat transfer small enough to be negligible. For example, a vacuum of about 10^{-2} mmHg (absolute, or 10^{-5} Torr) is needed to reduce the conductivity of air to 1% of its value at atmospheric pressure, and this is indeed the vacuum that has been used in most of the collector designs. The costs of manufacturing evacuated collectors are, however, somehow proportional to the vacuum level that needs to be attained and maintained during the life of the collector. It is therefore of interest to determine overall heat transfer as a function of the absolute pressure (i.e., degree of vacuum) in the enclosure and also to understand the factors that may increase the pressure during the life of the collector, such as joint leakage, volatilization of internal components, and penetration of gases from the adjoining spaces (e.g., the ambient and the working fluid) through the vacuum enclosure (usually glass) into the evacuated space.

Lou and Shih (1972a, b), Thomas (1973, 1979), and Wideman and Thomas (1980) studied conduction heat transfer in rarefied (partial vacuum) between

parallel plates and concentric cylinders and spheres and developed recommended equations.

Glasses are permeable to helium, which is present in the atmosphere at a partial pressure of about 4×10^{-3} Torr. This pressure is usually higher than that used in evacuated collectors (10^{-5} Torr), and if it penetrates into the collector and comes to its equilibrium pressure, it will cause a reduction in collector efficiency. Thomas (1981) presented a method for calculating the helium penetration rates and the consequent conduction heat flux. He concluded that penetration time constants of about 50 years can be expected if the glass envelope is kept close to ambient temperatures but that this would be reduced to a matter of a few months if the glass was operated at temperatures over 200°C. If helium came to its atmospheric equilibrium pressure in the collector, the conductive heat flux could rise from about 1% of the radiative flux, at 10^{-4} Torr, to about 25%.

If the correlations expressing the Nusselt number as a function of the Rayleigh (or Grashof) number just after convection onset are also correct for high vacuums, it is easy to determine the absolute pressure at which natural convection can be suppressed, as Eaton and Blum (1975) have done. They suggested that convection would be suppressed at pressures less than about 7 Torr for conditions typical to flat plate solar collectors operating at absorber temperatures up to 175°C. They also confirmed this experimentally in a qualitative way. It is important to note that convection can thus be suppressed even if the higher vacuums needed to suppress conduction have not been attained (or have slightly deteriorated in time due to leakage).

Progress Summary *Major progress was made in evacuated solar collectors. Such collectors existed only in concept at the turn of the 1970 decade, and yet they have begun competing effectively for a share of the market at the end of that decade. Understanding of the thermal theory of present-generation collectors of this type, and the ability to predict their behavior, became in that short time at least as good as that for conventional flat plate collectors. The primary aspects in which progress needs to be made, apart from the ever-present need for cost reduction, are in improving the energy collection rate* $[(\alpha\tau)_{cf}]$ *and perhaps in the understanding of heat transfer the in partial vacuum in such collector configurations.*

4.6 Heat Transfer from the Collector Exterior

Unless the collector is well shielded from wind, the convective thermal resistance R between its exterior and the ambient (figure 4.9) is dominated by forced convection due to wind flow over the collector (otherwise, the resistance is associated with natural convection). It is a quaint fact that the only way to calculate the heat transfer coefficient h_w due to forced convection over flat collector plates till fairly recently was by using the dimensional empirical correlation developed by Jurges in 1924 (see McAdams 1954) for flow parallel to a plate:

$$h_w = 5.7 + 3.8V, \qquad (46)$$

where V is the wind velocity, with all units in SI. The lack of characteristic length and the independence from properties and inclinations limit the validity of this correlation severely. The Russian literature shows the use of dimensionless correlations in the laminar flow regime, which also take angles of attack and yaw into account (Avezov et al. 1973a, b; Avezov and Vakhidov 1973).

Before development of improved predictive equations for convective wind effects is attempted, it must be realized that (1) wind varies with time in speed, direction, and turbulence, and its mean velocity changes with height, (2) wind arriving at the collector is affected by the topography upstream of the collector and surrounding it (see Kind et al. 1983; Kind and Kitaljevich 1985; Lee 1987), (3) sharp differences in wind speed and direction were found even on the face of each collector (Oliphant 1979), (4) free stream turbulence of the wind, in part generated by upstream and surrounding obstacles, has an important effect on heat transfer and can explain the difference between wind tunnel results with low free stream turbulence and results obtained in the natural environment, where the free stream turbulence may be 20% (based on the local velocity) and the convective heat transfer coefficient twofold higher (Test et al. 1981; Francey and Papaioannou 1985; Lee 1987), and (5) the shape of the leading edge of the collector affects convection at its surface downstream through such phenomena as separation, reattachment, and redevelopment (see Ota and Kon 1979; Test and Lessman 1980).

Sparrow and coworkers (Sparrow and Tien 1977; Sparrow et al. 1979, 1982) conducted a series of experiments in a wind tunnel, to develop correlations for convective heat transfer, using the naphtalene sublimation technique as analog to heat transfer. Square and rectangular plates of about 2 to 5 in. size were placed at different angles of attack and yaw, and both windward and leeward plate configurations were investigated for $2 \times 10^4 < Re < 10^5$ (laminar flow). They have developed a correlation for windward orientations:

$$j = 0.86 Re_L^{-1/2}, \qquad (47)$$

where j is the Colburn j-factor $= Nu/(RePr)$, and

$$Re_L = \frac{V(4A/C)}{\nu}, \qquad (48)$$

where A is the area of the plate and C the length of its perimeter.

They found practically no effect of angles of attack or yaw. With the collector leeward (wind blowing at its back) they found that for Reynolds numbers below 6×10^4 windward-face plates exhibit heat transfer coefficients about 10% higher than leeward-face ones, but this was reversed as Re exceeded 6×10^4: at $Re = 10^5$ the windward-face coefficient became 15% lower than the leeward-face one. They also determined that adding coplanar plates at the edges of the collector moves the highly convecting edge zones to these passive edge plates, and the heat loss can be reduced by up to about 10%.

The above correlation, as well as experimental results obtained by Kind et al. (1983) and Kind and Kitaljevich (1985) obtained in highly turbulent non-uniform flows generated in a wind tunnel, give heat transfer coefficients that may be as much as four-times lower than the Jurges correlation. Onur and Hewitt (1980) made convective heat transfer experiments with 6 in. models under a free jet and obtained results about 10% lower than those of Sparrow and coworkers. Kind and Kitaljevich (1985) also found that heat transfer coefficients for solar collectors mounted at an angle on a flat horizontal roof are 50% higher than those for collectors mounted flush with an inclined roof.

Truncellito et al. (1987) obtained numerical solutions for turbulent forced convection over a plate with an angle of attack for Reynolds numbers up to 3×10^5. They found that the Nusselt number increases slightly with the angle of attack and that the j-factor is the same for $Re \approx 3 \times 10^4$ as that predicted by equation (47), but it is increasingly larger as Re increases. Correlations of Nu as a function of Re, Pr, and the angle of attack were provided.

Lior and Segall (1986) have done experimental studies in a wind tunnel to determine convective heat transfer coefficients on a solar collector array composed of three parallel rows, all facing the wind, with variable spacing and inclination, and $4.8 \times 10^4 < Re < 8.5 \times 10^5$. For the upstream plate the Colburn j-factor was found to be slightly higher than that predicted by equation (47). It was up to about 40% higher for the second plate, due to effects of the wake generated by the plate upstream, but only up to about 30% higher for the third plate, due to the flow pattern. It was a weak function of inclination and spacing for the two downstream plates.

Forced convection heat transfer information for external flow around cylindrical collectors can be found in the book by Zukauskas (1985) which deals exclusively with heat transfer from cylinders exposed to external flow.

Progress Summary *Although important progress was made during the last decade in the understanding of forced convection over inclined plates for Reynolds numbers below 10^5, and in advancing beyond the limitations of the Jurges correlation, we have only begun to understand and to try to predict heat transfer due to wind in the natural environment and for realistic collector/surround geometries. The extension of the past work to larger Reynolds numbers, attainment of better agreement between results of different investigators, and the accounting for real geometries and the natural environment are needed.*

4.7 Transient Effects

The transient nature of solar radiation, ambient temperature, wind, and the heat load indicates that there may be merit in investigating the transient behavior of collectors, instead of using steady-state models such as the HW one. This is particularly important if the collectors have a large time constant, if frequent and strong variations in insolation occur in a region, or if the collector performance has a rapid influence on the ultimate heat load, on other parts of the solar system, or on system controllers. Klein et al. (1974) compared the zero-thermal-capacitance HW model with one-node (all thermal capacity lumped into one term) and multinode (transient energy balances for each component solved simultaneously) models that include the thermal storage term in the energy equation [equation (1)] for conventional flat plate collector parameters. They determined that the collector responded to step changes of the meteorological variables within a fraction of an hour and that therefore the zero-capacitance (steady-state) model is adequate when hourly (or longer-period) meteorological data are used. In other words, they recommended that transient effects need not be considered in performance modeling of conventional collectors. Wijeysundera (1976, 1978) developed a detailed transient model for an air-heating collector, and his results essentially concurred with those by Klein et al. (1974): A two-node model gave accurate results for collectors with up to three cover plates, a single-node model was satisfactory for collectors with one cover plate, but even a steady-state model was adequate if only hourly meteorological data were used.

Siebers and Viskanta (1978), de Ron (1980), Saito et al. (1984), and Kamminga (1985) modeled flat plate collectors by a set of at least three coupled equations, one each describing the transient energy balance in the fluid, each of the cover plates, and absorber. Whereas de Ron (1980) and Kamminga (1985) ignored heat conduction in the absorber and glass and thus ended up with these two equations being first-order ordinary differential, Siebers and

Viskanta (1978) and Saito et al. (1984) in addition consider conduction in the flow direction and thus have two (or three for two cover plates) second-order partial differential equations and one first-order partial differential equation. The latter approach produced excellent agreement with experimental data. Both de Ron (1980) and Kamminga (1985), however, introduced simplified linear models that can represent collector behavior well without the need to use the rigorous, complex models. Other researchers have pointed out that apart from the already-recognized importance of the heat capacity of the fluid, the heat transfer coefficient between the tubes and the heated fluid is an important parameter that should not be ignored in transient analysis and that the single-node model (see Klein et al. 1974) does not describe the transient behavior well. It was found that the transient temperature of the heated fluid in collectors in which the absorber is well-insulated from the cover plate (as it usually is) can be predicted well from models that use only the transient energy balance equations for the fluid and the absorber and ignore the transient terms in the equation for the cover plate.

Interested in investigating transient performance of evacuated tubular collectors, which have a large time constant, Mather (1982) applied a similar analysis to that of de Ron (1980) and obtained excellent agreement with experimental data. A similar model for the same purpose was developed later by Bansal and Sharma (1984). Morrison and Ranatunga (1980) developed a transient model for thermosyphon collectors and verified it experimentally.

Edwards and Rhee (1981) proposed a useful correction in the experimental determination of instantaneous efficiency of solar collectors that uses the time constant of the collectors determined by separate experiment (with no insolation).

In closing, it should be noted that good understanding of the theory of the transient behavior of collectors can also lead to the development of techniques for the rapid experimental determination of the parameters that characterize the collector and its performance. This could be used in collector performance testing and diagnostics and possibly in collector and component R&D.

Progress Summary *As found in the early 1970s, steady-state models are adequate for describing the energy collection performance of conventional flat plate collectors, when hourly (or longer-period) meteorological data are used. Transient modeling is, however, necessary for collectors with large time constants (e.g., many of the evacuated-tube collectors) or when the transient combination of weather, insolation, load, and system operation are such as to require it. Very good transient models with that capability have been developed and verified during the last decade. Good understanding of the theory of the transient* *behavior of collectors can lead to the development of experimental techniques for rapid diagnostics and evaluation of collector performance.*

4.8 Thermal Design of Solar Concentrator Receivers

Receivers are essentially solar collectors too, and their thermal theory would therefore be reviewed here. On the other hand, the few that exist have been custom designed for the specific system in which they operate, and there isn't nearly as much information on their thermal modeling and optimization as available for solar collectors. The review of this subject would be rather brief.

From the viewpoint of thermal theory and modeling, solar concentrator receivers have many obvious similarities to conventional nonconcentrating solar collectors, but they also differ from them in several aspects:

1. For high concentration ratios it is often not practical to transfer the maximally focused radiant flux [about 250 Btu/ft^2 s (2.8 MW/m^2) as it enters or strikes the receiver, an order of magnitude greater than used in conventional fuel-fired boilers] into the working fluid even with the highest practical conductive/convective heat transfer coefficients between the exterior wall of the fluid conduit and the fluid itself, especially if the highest fluid outlet temperature is desired. An attempt to apply that flux to fluid conduits may result in poor efficiency and in hot spots that can damage the receiver. A typical remedy is to redistribute the radiant flux over a larger heat exchanger area inside the receiver once the beam entered it, while keeping the inlet aperture small to reduce radiative and convective losses.

2. Radiant energy exchange becomes dominant and requires much more precise calculation.

3. Due to the larger temperature differences between the receiver and the ambient, and in some cases due to the larger characteristic dimensions, the Grashof number reaches up to 10^{14}, and natural convection becomes highly turbulent and much more vigorous. For open cavities in wind, the forced convection Reynolds number at the same time may reach 10^7. This requires both theoretical and empirical heat transfer information, which is still quite scarce (see Abrams 1983; Siebers and Kraabel 1984).

4. The large temperature differences incurred require the consideration of the temperature dependence of the radiative and convective properties of the materials.

5. In contrast with nononcentrating collectors, the diffuse component of solar radiation usually needs not be considered in the thermal analysis.

A plate-tube design (with tubes spaced very closely, often touching each other) is commonly used for receivers. Tube flow patterns are determined by considerations of heat transfer, thermal stress, heated fluid quality (fraction of vapor), and cost (see Sobin et al. 1974). The plate-tube system gains energy from the solar flux and in cavity receivers also from irradiation and reflections from other surfaces that it views, and it loses heat by reradiation, natural convection, and possibly forced convection if exposed to wind. The useful heat is gained by the working fluid, which may exit in the same phase in which it entered, or phase change may occur during passage. The latter may be in a boiler, in which a subcooled liquid may first be brought to saturation temperature and then change phase into steam. Finally, the generated steam may be superheated before it exits the receiver.

Cavity Receivers (figure 4.4) Cavity receivers are designed to minimize radiative losses by absorbing as much as possible of the incoming radiation into internal walls that do not view the opening. Due to the large temperature differences between the interior walls of the cavity and the ambient, and often the large size of the receiver, natural convection in the cavity can be vigorous, and it would carry some of the heat from the walls to the cavity aperture. That aperture is often open to the ambient because of the high temperatures that such windows may be exposed to, and it also reduces transmittance losses in the window.

Rozkov (1977) studied natural convection in cylindrical receivers. Humphrey and Jacobs (1981) developed a numerical model for predicting laminar flow and temperature fields in a small open cubical cavity subiect to external wind, and they calculated the heat loss from the cavity. Since many receivers would usually incur turbulent convection, the model needs to be developed further and compared with experiments. Clausing (1983) developed a simplified engineering model to determine heat losses from such a cavity receiver, and it was found that his predictions were in close agreement with the experimental data of McMordie (1984) and Mirenayat (1981). He also developed a semiempirical correlation for the heat transfer coefficients between the inner surface of the cavity and the air for different surface angles. Later (Clausing et al. 1986) he determined from experiments that the area of the aperture and its location have a major influence on the overall Nusselt number for heat loss. Natural convection from a 2.2-m electrically heated cubical cavity was measured by Kraabel (1981). The aperture was vertical, comprising one side of the cube. The Grashof number in the experiments was varied between 9.4×10^{10} and 1.2×10^{12}. The experimental results were used to develop a correlation for the Nusselt number, and the correlation was reported interesting result from Kraabel's correlation was that for surface temperatures above 420 K, the Nusselt number for the cavity was larger than that for a single vertical plate, and thus the total heat loss from a cavity can exceed that calculated from the sum of losses from the individual walls.

Wind effects on cavity receiver heat transfer have not yet been studied experimentally, and the theory is also in its infancy.

Radiative exchange inside cavity receivers, and conclusions for the improvement of their performance, have been calculated by McKinnon et al. (1965) and Griliklies and Obtemperanskii (1969). In one cavity receiver, the receiver consisted of a conically wound tube, with the base (opening) of the cone acting as the aperture for the incoming concentrated solar flux (Rice et al. 1981). Rice et al. performed a thermal analysis on a rather simplified model of this receiver and determined its performance. As expected, smaller cone angles for the same size of the cone opening (base) provided higher receiver efficiency because of the reduction of the radiative loss. Convection inside the cone was treated in an oversimplified way, and the conclusions must be regarded as being primarily qualitative.

A detailed design of a 550-MW$_t$ output quad-cavity receiver (figure 4.4) for generating and superheating steam in a solar central power plant has been presented by Wu et al. (1983). In comparing this design to an external receiver for similar duty, the authors found that the losses for this cavity receiver were about half of those for the external receiver. It is noteworthy, though, that not enough is known yet about losses due to combined natural convection and wind effects from cavity receivers; they may be larger than estimated by these authors (see Hildebrandt and Dasgupta 1980). Harris and Lenz (1985) analyzed thermal performance of concentrator/cavity receiver systems and made recommendations for optimization.

A cylindrical cavity receiver with an aperture parallel to the axis of the cylinder (figure 4.17), which is suitable for line-focusing concentrators, was recommended and analyzed by Boyd et al. (1976), and a version of it was analyzed, built, and tested successfully in a parabolic-trough concentrator (Barra and Franceschi 1982), with further recommendations for optimization. CPC collectors may also be regarded in some sense as cavity receivers. Thermal analyses of heat transfer and natural convection in a CPC receiver were performed numerically by Abdel-Khalik et al. (1978), and thermal analysis for an evacuated glass-jacketed CPC receiver was performed by Thodos (1976).

It is important to note that there is a trade-off in cavity receiver designs between having a large receiver aperture, associated with a low concentration ratio, to capture all of the incoming energy, and having a small aperture (high concentration ratio) to minimize radiation and convection losses (see

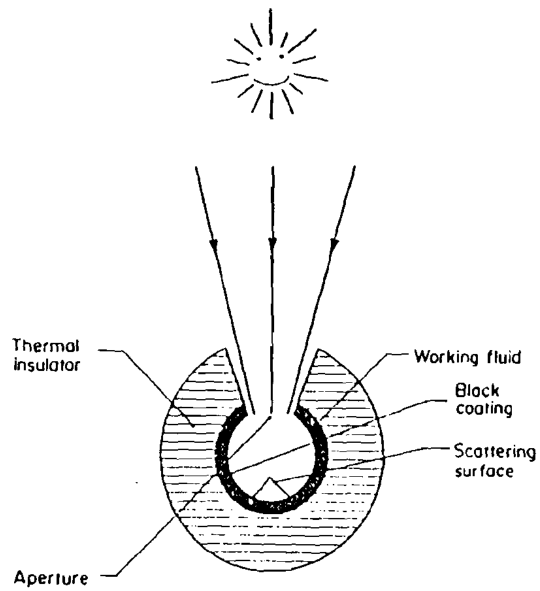


Figure 4.17
Cross section of the cylindrical cavity receiver. Source: Boyd et al. (1976).

Pons 1980). This fact couples the thermal analysis with the design of the concentrator.

Internal radiative exchange and heat transfer through the working fluid conduit walls into the fluid become quite complicated when optimized designs are sought, and the solution of such problems must be done numerically. A finite difference code for simulating heat transfer in cylindrical solar receivers, named "HEAP," was developed at the Jet Propulsion Laboratory (Lansing 1979). A program to analyze cavity radiation exchange ("CREAM") was developed at the University of Houston (Lipps 1983). Other programs have been developed by the Aerospace Corporation and modified by JPL (El Gabalawi et al. 1978) and by the Pacific Northwest Laboratory (Bird 1978), by the Solar Energy Research Institute (Finegold and Herlevich 1980), and by Sandia National Laboratories (MIRVAL: Leary and Hankins 1979; HELIOS: Vittitoe and Biggs 1981). Some of these programs also apply to external receivers.

External Receivers (figure 4.5) The external receiver of Solar 1, the 10-MW solar thermal power plant at Barstow, California, is a once-through boiler of

7-m diameter and 12.5-m height. The average temperature of the exterior of the receiver is 600°C, and the wind velocities perpendicular to the receiver are 0 to 25 m/s. These conditions result in Reynolds numbers (based on diameter) of 0 to 10^8 and Grashof and Rayleigh numbers of 10^{12} to 10^{14}.

Siebers et al. (1982, 1983) conducted experiments on a 3 × 3 m electrically heated plate in a wind tunnel and developed correlations for natural, mixed, and forced convection in the laminar and turbulent regimes. Mixed convection was found to exist in the range $0.7 < Gr/Re^2 < 10.0$. A correlation was also developed to indicate the transition from laminar to turbulent flow. It should be noted that natural convection on this receiver is always turbulent, and it can be neglected relative to the wind-induced forced convection only when the wind velocity exceeds about 3 m/s. Since wind velocities vary between 0 and 25 m/s, there are many periods in which natural convection is the dominant convection mode. Mixed convection for these configurations was studied by Afshari and Ferziger (1983).

Details of the thermal analysis and overall design of an external cylindrical receiver to generate and superheat steam for a 100-MW_e solar central power plant (figure 4.5) has been presented by Yeh and Wiener (1984) and Durrant et al. (1982). The importance of heat losses due to wind and their effect on overall efficiency were determined as a result of the analysis: For example, when wind velocity increased from 0 to 13 m/s, the receiver efficiency decreased from about 91% to 86% (at an ambient temperature of 25°C). A general purpose computer program for the design of solar thermal central external receiver plants, DELSOL2, was developed by Sandia National Laboratories (Dellin et al. 1981).

External receivers, usually in the form of a tube coiled helically around a cylindrical mandrel, are also used with the fixed-mirror distributed-focus concentrator in which the concentrator is a fixed hemispherical dish and the receiver tracks the focal point. The tube carrying the heated fluid is exposed to a highly asymmetric radiative flux. Dunn and Vafaie (1981) have developed empirical correlations for the Nusselt number for both single and boiling water/steam flow inside the coil.

As stated earlier, external receivers in the form of a simple tube conducting the heated fluid are often used with line-focusing concentrators. The tube's centerline is placed at the focal line, and in many designs it is placed concentrically inside another (transparent) tube to insulate it by the annular air (or vacuum) gap formed thereby. Analyses of such receivers were done by Barra et al. (1978a, b), Deanda and Faust (1981), Harrison (1982), and May and Murphy (1983), and reviewed by Chiang (1982), and a numerical analysis of the effects of natural convection in the annular air gap on heat losses was performed by Ratzel et al. (1979).

Direct absorption receivers, where the solar flux is absorbed directly in the working fluid, such as in powders, particles, and molten salt films, are in the research stage, and their thermal principles have been reviewed in section 4.3.6.

Progress Summary *Much progress has been realized in the thermal analysis, design, and construction of both small and large solar receivers. Information about wind effects and natural convection, separately and combined, on heat losses in large receivers is still inadequate. Interesting concepts of receivers that employ direct absorption of the solar flux into powder/air suspensions, solid particles, and falling liquid films are being explored.*

4.9 Solar Collector Arrays

It has been recognized in both theory and practice that arrays composed of a large number of collectors do not perform in a way that could be predicted by the simple addition of the performances of the individual collectors that compose them. Large arrays typically performed poorer, in some installations at half the efficiency predicted from single collector data. The reasons for the difference include increased heat and pressure losses from array piping, flow maldistribution among the collectors, increased heat losses due to wind because of the influence of the array on the wind at its location, much larger thermal capacitance, and sometimes mutual shading of collectors.

Supported by the USDOE, Lior and coworkers at the University of Pennsylvania developed a computer simulation program to address these array design and performance problems (Menuchin et al. 1981), and they have started the experimental study of wind effects on the thermal performance of collector arrays because no data in this area were available (Lior and Segall 1986, described in further detail in section 4.6). The problems associated with large array performance have recently brought about a workshop of the International Energy Agency (Bankston 1984) in which comprehensive information on operating, design, and research aspects of large arrays was presented and discussed. General agreement was obtained among the workshop attendees that many systems operated at half of the predicted efficiency, or worse.

The objective of the work at the University of Pennsylvania was to optimize design of collector arrays by determining the best configuration of collector rows as a function of available collector mounting area, interrow spacing, row orientation, collector inclination and height, wind effects, and system cost.

Two comprehensive computer programs were developed for the simulation and optimization: SOLRAY, which computes the hydrodynamics of flow in arbitrarily piped (any parallel series combination) solar collector arrays, with the inclusion of both inertia and friction effects (based on the work by Jones and Lior 1978), and the University of Pennsylvania program SOLSYS (Edelman et al. 1977) which computes the thermal efficiency of the arrays by using as input the individual flow rates computed by SOLRAY. SOLSYS can also compute the thermal performance of partially shaded collectors. From the information provided by SOLSYS, SOLRAY then determines overall energy efficiencies and costs of different solar collector array configurations, to find the optimum. Heat losses from interconnecting pipes could be calculated from another program developed by Jones and Lior (1979).

For the size and piping of typical flat plate solar collectors, it was found (Menuchin et al. 1981) that the flow in the parallel flow dual-manifold system is friction dominated; that is, the flow rate is lowest in the inner collectors. Since parallel flow dual-manifold systems provide more uniform flow distribution than reverse flow ones, only parallel flow was considered in the optimization. The maldistribution increases with the spacing between the parallel piped collectors and with their number. It also increases as the collector pressure drop is decreased and as the ratio between the diameter of the tube connecting the collector to the manifold and the diameter of the manifold is increased. A life-cycle present-value cost analysis (for 1978 conditions, and includes also both energy and capital costs of pumping) showed that the cost has a minimum for a given number of collectors piped in parallel, which for the total number of collectors considered (48, 96, 192, and 288), consisted of 16 to 24 collectors. Other information useful to array design was also provided.

Using a simple HW model for collector performance and considering a single parallel piped row, Culham and Sauer (1984) suggested that flow imbalance in arrays would have little effect as long as the flow in the collector does not fall below 35% of the recommended design value. Mansfield and Eden (1978) made recommendations on the ways to use thermography for determining malfunctions or flow maldistribution in collector arrays.

Array piping and configuration is of great importance in distributed concentrator solar thermal heat and power systems, particularly because of the high temperature and the greater spacing between the units. A number of computer programs have been developed for economic optimization of the piping of such systems (see Barnhart 1979; Fujita et al. 1982).

It was recognized (Eck et al. 1984) that differences between array performance and single-collector performance predictions may well arise also from the fact that the standard test conditions for collectors (e.g., that by ASHRAE

Standard 93-77) are not representative of operating conditions. For example, the standard for testing collectors specifies solar radiation that is often higher than seen by the operating collectors, and wind speeds and incidence angles that are lower. McCumber and Weston (1979) have indeed demonstrated that if the highly scattered field data for arrays is filtered from points measured at conditions beyond those required by the ASHRAE test standard, it clusters well around the straight-line HW equation. The standard also specifies only "instantaneous" rather than all-day efficiency, and requires quasi-steady operating conditions. It is therefore worthwhile to explore the development and use of standard performance characterization methods that give more realistic long-term operating results.

Progress Summary *The fact that large collector arrays perform worse than predicted by single-collector test data was foreseen and has indeed materialized in most such installations. The basis for computer programs that could predict array behavior correctly or optimize the design has been developed, and the programs should be enhanced as needed and made available to designers. A basis for knowledge of wind effects on heat losses from arrays has also been established (for* Re $< 8.5 \times 10^5$, *parallel flow), but the work should be expanded to address the entire spectrum of wind conditions, and validated with full-scale systems in the natural environment. Flat plate collector performance standards, such as ASHRAE 93-77, were found to produce performance expectations that cannot be met in field operation of large arrays and should be reviewed and modified to produce more realistic results.*

4.10 Collector Performance Sensitivity to Design Parameters

Parametric studies to determine collector performance sensitivity to its components (number of covers, their thickness, tube dimensions, tube-to-fluid heat transfer coefficient, plate spacings, emittance of cover plates, emittance of absorber plate, absorptance of absorber plate, thermal conductivity of cover plates, thermal conductivity and thickness of the absorber plate, and thermal resistance of the insulation), to operating factors (type of fluid and its inlet temperature and flow rate), and to meteorological variables (insolation, ambient temperature, wind velocity, and sky temperature) have been performed by a number of researchers (Test 1976; Wolf et al. 1981; Arafa et al. 1978), using more detailed models than HW. The highlights of the sensitivity to the most influential parameters are presented in this section.

For a single-glazed water-heating collector, increase of the tube/fluid heat transfer coefficient up to about 300 kJ/m² hr K cantly and has little effect thereafter. Similarly increasing flow rate (at constant tube/fluid heat transfer coefficient) up to 20 kg/m² hr improves efficiency significantly and has little effect thereafter. The (conductivity × thickness) product of the absorber has an important effect: Its reduction from 0.5 W/K to 0.001 W/K typically reduces the efficiency by at least one-third. The efficiency increases strongly and linearly with absorber plate absorptance and decreases with emittance. Lowering the emittance of the cover plate from 0.95 to 0.6 improves the efficiency by at least 10%. The effect of insulation (fiberglass) thickness depends on the ambient temperature, but little is gained beyond 5 cm.

Insolation, ambient temperature and fluid inlet temperature have an important influence on collector performance, as expected, and wind has a moderate effect.

For a double-glazed collector the main difference is that there is weaker dependence on absorber emittance. The number of glazings has a major effect, but it depends strongly on the temperature difference between the absorber and the ambient.

Studies of the effect of the type of fluid on collector efficiency were conducted by Youngblood et al. (1979) who used four different fluids: water, ethylene-glycol/water solution (50% by weight), a silicone based heat transfer fluid (Syltherm 444), and a synthetic hydrocarbon (Therminol 44). Each of the fluids was tested in four types of commercial flat plate collectors. It was found that relative to water the other liquids reduced the collector efficiency by (1) 2%–4% for the ethylene-glycol/water solution, (2) 3%–10% for Syltherm 444, and (3) 3.5% for Therminol 44.

Progress Summary *The advanced models developed for collector analysis allow good quantitative sensitivity analysis for the purposes of collector research, development, and design.*

4.11 Summary: A Thermal Modeling Guide and Theory Outlook

Conventional flat plate collector design and performance determination can still be performed quite successfully by using the Hottel-Whillier lumped model, but with up-to-date values for the required coefficients as described in the preceding sections. The most recent and probably most accurate correlation for the overall top heat loss coefficient is the one by Garg and Datta (1984). More detailed information, and somewhat better modeling, can be obtained by performing individual energy balances for each cover and solving the equations simultaneously, but with convective coefficients that are now

much more accurate and possibly with a more precise description of the radiant exchange. For collector R&D, new configurations, and materials, including coatings for the modification of radiative properties, two- (or even three-) dimensional models of the Navier-Stokes and energy equations need to be solved as a conjugate problem if the most accurate representation is needed.

From the thermal modeling standpoint, evacuated collectors are in one respect easier to model since no conduction or convection between the absorber and cover exists (if adequately high vacuum can be assumed). In another respect they are somewhat more difficult to model due to the radiative transfer in the cylindrical geometry, which is often internally asymmetric (see Saltiel and Sokolov 1982; Garg et al. 1983). Bhowmik and Mullick (1985) developed an expression for the overall top loss coefficient for such collectors, for use in an HW lumped model. Several more detailed thermal models have been developed (Mather 1982; Behrendorff and Tanner 1982; Rahman et al. 1984; Banzal and Sharma 1984; the first and last ones transient) to serve as a good start for more precise modeling.

Although thermal modeling of collectors was in the past restricted to steady state, good transient models have been developed in the last decade to be used where appropriate.

Receiver analysis is typical done by using three-dimensional finite difference or element programs, and probably the only uncertainty is in the effects of natural, forced, and mixed convection, especially in large cavity receivers.

Notes

1. One may note that the energy input is primarily in the form of thermal radiation (electromagnetic wave energy), and the losses are partially radiative and partially thermal. The useful energy output is assumed to be only heat, and so is the energy storage term.

2. Radiation on the horizontal surface serves as the basis because solar radiation is usually measured and reported in the horizontal plane.

References

Abdel-Khalik, S. I. 1976. Heat removal factor for a flat-plate solar collector with a serpentine tube. *Solar Energy* 18: 59–64.

Abdel-Khalik, S. I., H.-W. Li, and K. R. Randall. 1978. Natural convection in compound parabolic concentrators A finite-element solution. *J. Heat Transfer* 100, 2: 199–204.

Abdelrahman, M. A., P. Fumeaux, and P. Suter. 1979. Study of solid-gas suspensions used for direct absorption of concentrated solar radiation. *Solar Energy* 22: 45–61.

Abrams, M. 1983. The status of research on convective losses from solar central receivers. Sandia Report 83-8224. Sandia National Laboratories. Livermore CA

Abramzon, B., I. Yaron, and I. Borde. 1983. An analysis of a flat-plate solar collector with internal boiling. *ASME J. Solar Energy Eng.* 105: 454–460.

Afshari, B., and J. H. Ferziger. 1983. Computation of orthogonal mixed convection heat transfer. *Proc. ASME-JSME Thermal Engineering Joint Conference*, vol. 3, pp. 169–173.

Al-Arabi, M., and M. M. El-Rafaei. 1978. Heat transfer by natural convection from corrugated plates to air. *Int. J. Heat Mass Transfer* 21: 357–359.

Al-Tamimi, A. I., and J. A. Clark. 1983. Thermal analysis of a solar collector containing a boiling fluid. *Progress in Solar Energy*, vol. 6, ASES Annual Meeting, pp. 319–324.

Ambrosone, G., A. Andretta, F. Bloisi, S. Catalanotti, V. Cuomo, V. Silvestrini, and L. Vicari. 1980. Long-term performance of flat-plate solar collectors. *Appl. Energy* 7, 1-3: 119–128.

Appelbaum, J., and J. Bany. 1979. Shadow effects of adjacent solar collectors in large scale systems. *Solar Energy* 23: 497–507.

Arafa, A., N. Fisch, and E. Hahne. 1978. A parametric investigation on flat-plate solar collectors. *Sun, Mankind's Future Source of Energy*. Proc. 1978 ISES Congress, New Delhi, India. Elmsford, NY: Pergamon, pp. 917–923.

Arai, N., Y. Itaya, and M. Hasatani. 1984. Development of a "volume heat-type" type solar collector using a fine-particle semitransparent liquid suspsension (FPSS) as a heat vehicle and heat storage medium. *Solar Energy* 32, 2: 49–56.

Arai, N., Y. Kato, M. Hasatani, and S. Sugiyama. 1980. Unsteady heat transfer in an optically thick semitransparent liquid layer heated by radiant heat source. *Heat Transfer—Japanese Research* 9: 22–31.

Arnold, J. N., I. Catton, and D. K. Edwards. 1975. Experimental investigation of natural convection in inclined rectangular regions of differing aspect ratios. ASME paper no. 75-HT-62. AIChE-ASME, Heat Transfer Conference, San Francisco, CA, August 11–13.

Arnold, J. N., D. K. Edwards, and I. Catton. 1977. Effect of tilt and horizontal aspect ratio on natural convection in rectangular honeycomb solar collectors. *J. Heat Transfer* 99: 120–122.

Avezov, R., A. Akhmadaliev, and N. A. Kakhdarov. 1973a. Effect of the angle of attack on the efficiency and heat Transfer of the glass cover of a solar installation under laminar-flow conditions. *J. Appl. Solar Energy* 9, 5: 45–48.

Avezov, R., A. Azizov, S. Khatamov, and G. Sharipov. 1973b. Experimental study of the aerodynamic resistance of a shingle heat-strong unit. *J. Appl. Solar Energy* 9, 5: 49–52.

Avezov, R. R., and A. T. Vakhidov. 1973. Choice of controlling dimension in the oblique flow past the glass surface of a solar installation. *J. Appl. Solar Energy* 9, 6: 90–91.

Bajura, R. A., and E. H. Jones. 1976. Flow distribution manifolds. *J. Fluids Eng.* 98: 654–666.

Baker, L. H. 1967. Film heat-transfer coefficients in solar collector tubes at low Reynolds numbers. *Solar Energy* 11: 78–85.

Baker, S. H., D. K. McDaniels, H. D. Kaehn, and D. H. Lowndes. 1978. Time integrated calculation of the insolation collected by a reflector-collector system. *Solar Energy* 20: 415–417.

Bando, T., M. Nishimura, M. Kuraishi, T. Kasuga, and M. Hasatani. 1986. Effect of optical depth on outdoor performance of "volume heat trap" type solar collector. *Heat Transfer—Japanese Researach* 15: 57–71.

Bankston, C. A., comp. 1984. *Proc. International Energy Agency Workshop on the Design and Performance of Large Solar Thermal Collector Arrays.* San Diego, CA, June 10–14. Solar Energy Research Institute report SERI/SP-271-2664.

Bansal, N. K., and A. K. Sharma. 1984. Transit theory of a tubular solar energy collector. *Solar Energy* 32: 67–74.

Barnhart, J. S. 1980. ETRANS: An energy transport system optimization code for distributed networks of solar collectors. Pacific Northwest Laboratory report PNL-3327.

Barra, O., M. Conti, E. Santamata, R. Scarmozzion, and R. Visentin. 1977. Shadows' effect in a large scale solar plant. *Solar Energy* 19: 759–762.

Barra, O. A., M. Conti, L. Correra, and R. Visentin. 1978a. Thermal regimes in a primary fluid heated by solar energy in a linear collector. *Il Nuovo Cimento* 1C: 185–195.

Barra, O. A., M. Conti, L. Correra, and R. Visentin. 1978b. Transient temperature variations in the primary network of a solar plant. *Il Nuovo Cimento* 1C: 167–184.

Barra, O. A., and L. Franceschi. 1982. The parabolic trough plants using black body receivers: Experimental and theoretical analyses. *Solar Energy* 28: 163–171.

Beard, J. T., F. A. Iachetta, L. U. Lilleleht, F. L. Huckstep, and W. B. May. 1978. Design and operation influences on thermal performance of "Solaris" solar collector. *ASME J. Eng. for Power* 100: 497–502.

Beard, J. T., F. A. Iachetta, R. F. Messer, F. L. Huckstep, and W. B. May. 1977. Performance and analysis of an open fluid-film solar collector. *Proc. AS/ISES Ann. Meet.* 1: 126–129.

Beckman, W. A. 1968. Radiation and convection heat transfer in a porous bed. *ASME J. Eng. for Power* 90: 51–54.

Behrendorff, M. J., and R. I. Tanner. 1982. Transient performance of evacuated tubular solar collectors. *J. Solar Energy Eng.* 104: 326–332.

Bennet, D. L., and J. C. Chen. 1980. Forced convective boiling in vertical tubes for saturated pure components and binary mixtures. *AIChE J.* 26: 454–461.

Bevill, B., and H. Brandt. 1968. A solar energy collector for heating air. *Solar Energy* 12: 19–36.

Bhowmik, N. C., and S. C. Mullick. 1985. Calculating of tubular absorber heat loss factor. *Solar Energy* 35: 219–225.

Bird, S. 1978. Modification of the JPL solar thermal simulation code for use in the PNL small solar thermal power system analysis. Battelle Pacific Northwest Laboratory, Richland, WA.

Black, A. W., and E. M. Sparrow. 1967. Experiments on turbulent heat transfer in a tube with circumferentially varying thermal boundary conditions. *J. Heat Transfer* 89: 258–268.

Blanpied, M., G. Clark, and J. S. Cummings. 1982. Effect of tilt angle on infrared radition received on tilted surfaces. *Annual Meeting American Solar Energy Society*, vol. 5, pt. 2 of 3, pp. 673–678.

Bliss, R. W., Jr. 1959. The derivations of several "plate-efficiency factors" useful in the design of flat-plate solar heat collectors. *Solar Energy* 3, 4: 55–64.

Bliss, R. W., Jr. 1955. Multiple gauge flat plate solar air heaters. *Proc. World Symp. on Applied Solar Energy*. Phoenix, AZ, pp. 151–158.

Blum, H. A., J. M. Estes, and E. E. Kerlin. 1973. Design and feasibility of flat plate solar collectors to operate at 100–150°C. Paper no. E18. International Solar Energy Society Conference, Paris.

Bohn, M. S. 1985. Direct absorption receiver. *Proc. Solar Thermal Research Program Annual Conference*. SERI/CP-251-2680, DE85002938, pp. 89–96.

Bol, K., and M. Lang. 1978. Two-phase working fluid in solar collectors. AIAA paper N354.

Borgese, D., G. Dinelli, J. J. Faure, J. Gretz, and G. Schober. 1984. Eurelios, the 1-MW(el) heliostatic power plant of the European community program. *ASME J. Solar Energy Eng.* 106: 66–77.

Boyd, D. A., R. Gajewski, and R. Swift. 1976. A cylindrical blackbody solar energy receiver. *Solar Energy* 18: 395–401.

Buchberg, H., I. Catton, and D. K. Edwards. 1976. Natural convection in enclosed spaces—A review of applications to solar energy collection. *J. Heat Transfer* 98, 2: 182–188.

Buchberg, H., and D. K. Edwards. 1976. Design considerations for solar collectors with cylindrical glass honeycomb. *Solar Energy* 18: 182–188.

Buchberg, H., O. A. Lalude, and D. K. Edwards. 1971. Performance characteristics of rectangular honeycomb solar—Thermal converters. *Solar Energy* 13: 193–221.

Budin, R., and L. Budin. 1982. A mathematical model of shading calculation. *Solar Energy* 29, 4: 339–349.

Cane, R. L. D., K. G. T. Hollands, G. D. Raithby, and T. E. Unny. 1977. Free convection heat transfer across inclined honeycomb panels. *J. Heat Transfer* 99: 86–91.

Catton, I. 1978. Natural convection in enclosures. *6th International Heat Transfer Conference*. Montreal, Canada, pp. 13–30.

Chan, A. M. C., and S. Banerjee. 1979a. Three-dimensional numerical analysis of transient natural convection in rectangular enclosures. *J. Heat Transfer* 101: 114–119.

Chan, A. M. C., and S. Banerjee. 1979b. A numerical study of three-dimensional roll cells within rigid boundaries. *J. Heat Transfer* 101: 233–237.

Chao, P., H. Ozoe, and S. W. Churchill. 1981. The effect of a non-uniform surface temperature on laminar natural convection in a rectangular enclosure, *Chem. Eng. Commun.* 9: 245–254.

Chao, P. K., H. Ozoe, N. Lior, and S. W. Churchill. 1983a. The effect of partial baffles on natural convection in an inclined rectangular enclosure. Part I: Experimental observations. *Chem. Eng. Fund.* 2, 2: 39–49.

Chao, P. K., H. Ozoe, N. Lior, and S. W. Churchill. 1983b. The effect of partial baffles on natural convection in an inclined rectangular enclosure. Part II: Numerical solution. *Chem. Eng. Fund.* 2, 2: 38–49.

Charters, W. W. S., and L. F. Peterson. 1972. Free convection suppression using honeycomb cellular materials. *Solar Energy*, 13: 353–361.

Chen, J. C. 1966. Correlation for boiling heat transfer to saturated fluids in convective flow. *Ind. Eng. Chem. Process Design Develop.* 5: 322–329.

Cheng, K. C., and S. W. Hong. 1972. Effect of tube inclination on laminar convection in uniformly heated tubes for flat-plate solar collectors. *Solar Energy* 13: 363–371.

Chiang, C. J. 1982. Thermal receiver designs for line-focusing solar collectors. Sandia National Laboratories report. SAND81-1862.

Chiou, J. P. 1982. The effect of nonuniform fluid flow distribution on the thermal performance of solar collector. *Solar Energy* 29, 6: 487–502.

Chiou, J. P. 1980. The effect of longitudinal heat conduction on the thermal performance of the flat plate solar collector. ASME paper 80-C2/Sol-5.

Chiou, J. P. 1979. Noniterative solution of heat transfer equation of fluid flow in solar collector. ASME paper 79-WA/Sol-24, New York, 6 pp.

Chiou, J. P., M. M. El-Wakil, and J. A. Duffie. 1965. A slit-and expanded aluminum-foil, matrix solar collector. *Solar Energy* 9, 2: 73–80.

Chow, S. P., G. L. Harding, B. Window, and K. J. Cathro. 1984. Effect of collector components on the collection efficiency of tubular evacuated collectors with diffuse reflectors. *Solar Energy* 32: 251–262.

Christopher, D. M., and J. T. Pearson. 1982. Convectice heat transfer in a louvered air-heating solar collector. ASME paper 82-HT-41, 9 pp.

Clausing, A. M. 1983. Convective losses from cavity solar receivers—Comparisons between analytical predictions and experimental results. *J. Solar Energy Eng.* 105: 29–33.

Clausing, A. M., L. D. Lister, and J. M. Waldvogel. 1986. An experimental investigation of convective losses from cavity solar receivers. *ASES Ann. Meet.*, pp. 248–251.

Cobble, M. H. 1964a. Irradiation into transparent solids and the thermal trap effect. *Franklin Institute J.* 278: 383–393.

Cobble, M. H. 1964b. Heating a fluid by solar radiation. *Solar Energy* 8, 2: 65–68.

Cobble, M. H., P. C. Fang, and E. Lumsdaine. 1966. Verification of the theory of the thermal trap. *Franklin Institute J.* 282: 102–107.

Cole-Appel, B. E., and R. D. Haberstroh. 1976. Performance of air-cooled flat plate collectors. *Proc. ISES/SESC Joint Meeting "Sharing the Sun"* 2: 94–106.

Cole-Appel, B. E., G. O. G. Löf, and L. E. Shaw. 1977. Performance of air-cooled flat plate collectors. *Proc. 1977 AS/ISES Ann. Meet.*, pp. 2.6–2.10.

Cooper, P. I., and R. V. Dunkle. 1981. A Non-linear flat-plate collector model. *Solar Energy* 26: 133–140.

Culham, R., and P. Sauer. 1984. The effects of unbalanced flow on the thermal performance of collector arrays. *ASME J. Solar Energy Eng*, vol. 106, pp. 165–170.

Davis, S. B. 1982. Ceramic high temperature receiver. *Proc. Parabolic Dish Solar Thermal Power Annual Program Review*. Report DOE/JPL-1060-52, pp. 247–256.

Deanda, L. E., and M. Faust. 1981. Steam rankine solar receiver, phase 2. Final Technical Report. Report NASA-CR-168656; NAS 1 26-168656, 95 pp.

Dellin, T. A., M. J. Fish, and C. L. Yang. 1981. User's manual for DELSOL2: A computer code for calculating the optical performance and optimal system design for solar-thermal central-receiver plants. Sandia National Laboratories report SAND-81-8237, 76 pp.

de Graaf, J. G. A., and E. F. M. Van der Held. 1954. The relation between the heat transfer and the convection phenomena in enclosed plane air layers. *Appl. Sci. Res.* A4: 460–461.

de Graaf, J. G. A., and E. F. M. Van der Held. 1953. The relation between the heat transfer and the convection phenomena in enclosed plain air layers. *Appl. Sci. Res.* A3: 393–409.

de Ron, A. J. 1980. Dynamic modelling and verification of a flat-plate solar collectors. *Solar Energy* 24, 2: 117–128.

de Vahl Davis, G. 1986. Finite difference methods for natural and mixed convection in enclosures. *Heat Transfer 1986, Proc. Eighth International Heat Transfer Conf.*, Washington, DC: Hemisphere, pp. 101–109.

de Winter, F. 1975. Solar energy and the flat plate collector—An anotated bibliography. ASHRAE report S-101, New York (now Atlanta). February 1975.

Diab, M. R., J. T. Pearson, and R. Viskanta. 1980. A two-dimensional analysis of flat plate air-heating solar collectors. ASME paper no. 80-HT-117. *Joint ASME/AIChE Nat. Heat Transfer Conf.*, Orlando, FL, 9 pp.

Diaz, L. A., and N. V. Suryanarayana. 1981. Temperature variation in the absorber plate of an air heating flat plate collector. *J. Solar Energy Engineering* 103, 2: 153–157.

Dietz, A. G. H. 1963. Diathermaneous materials and properties of surfaces. In A. M. Zarem and D. D. Erway, eds., *Introduction to the Utilization of Solar Energy*. New York: McGraw-Hill, ch. 4.

DOE-2 Reference Manual Version 2.1A. 1981. Los Alamos Scientific Laboratory report LA-7689-M.

Downing, R. C., and V. H. Waldin. 1980. Phase-change in solar hot water heating using R-11 and R-114. *Trans. ASHRAE* 86, pt. 1: 848–856.

Duffie, J. A., and W. A. Beckman. 1980. In *Solar Engineering of Thermal Processes*. New York: Wiley, ch. 8.

Duffie, J. A., and W. A. Beckman. 1974. In *Solar Energy Thermal Processes*, New York: Wiley, chs. 4–8.

Dunkle, R. V., and E. T. Davey. 1970. Flow distribution in solar absorber banks. *Proc. ISES Conf.*, Melbourne, Australia.

Dunn, J. R., and F. N. Vafaie. 1981. Forced convection heat transfer for two-phase helical flow in a solar receiver. ASME, paper 81-WA/HT-13. Winter Annual Meeting, 9 pp.

Durrant, O. W., T. J. Capozzi, and R. H. Best. 1982. The development and design of steam/water central solar receivers for commerical application. *J. Solar Energy Eng.* 104: 173–181.

Eaton, C. B., and H. A. Blum. 1975. The use of moderate vacuum environment as a means of increasing the collection efficiencies and operating temperatures of flat plate solar collectors. *Solar Energy* 17: 151–158.

Eck, T. F., T. L. Logee, and S. M. Rossi. 1984. Performance of collectors in the national solar data network. *Proc. International Energy Agency Workshop on the Design and Performance of Large Solar Thermal Collector Arrays*, C. A. Bankston, comp. Solar Energy Research Institute report SERI/SP-271-2664. San Diego, CA, pp. 116–135.

Edwards, D. K. 1977. *Solar Collector Design*. Philadelphia: Franklin Institute Press.

Edwards, D. K. 1977. Solar absorption by each element in an absorber-coverglass array. *Solar Energy* 19, 4: 401–402.

Edwards, D. K., J. N. Arnold, and I. Catton. 1976. End-clearance effects on rectangular-honeycomb solar collectors. *Soler Energy* 18: 253–257.

Edwards, D. K., J. N. Arnold, and P. S. Wu. 1979. Correlations for natural convection through high L/D rectangular cells. *J. Heat Transfer* 101: 741–743.

Edwards, D. K., snd S. J. Rhee. 1981. Experimental correction of instantaneous collector efficiency for transient heating or cooling. *Solar Energy* 26: 267–270.

Edwards, D. K., and W. M. Sun. 1971. Effect of wall radiation on thermal instability in a vertical cylinder. *Int. J. Heat Mass Transfer* 14: 15–18.

El Gabalawi, N., G. Hill, J. Bowyer, and M. Slonski. 1978. A modularized computer simulation program for solar thermal power plants. Jet Propulsion Laboratory report JPL-5102-80.

Elsayed, M. M. 1984. Calculation of radiation properties of a stack of partially transparent plates with and absorber plate. *ASME J. Solar Energy Eng.* 106: 370–372.

Elsherbiny, S. M., K. G. T. Hollands, and G. D. Raithby. 1980. Free convection across inclined air layers with one surface V-corrugated. *J. Heat Transfer* 100: 410–415.

Elsherbiny, S. M., G. D. Rathby, and K. G. T. Hollands. 1982. Heat transfer by natural convection across vertical and inclined air layers. *J. Heat Transfer* 104: 96–102.

Emmet, W. L. R. 1911 (January 3). Apparatus for utilizing solar heat. U.S. patent no. 980,505.

Farouk, B., and S. I. Guceri. 1982. Laminar and turbulent natural convection in the annulus between horizontal concentric cylinders. *J. Heat Transfer* 104: 631–636.

Felland, J. R., and D. K. Edwards. 1978. Solar and infrared radiation properties of parallel plate honeycomb. *AIAA J. Energy* 2, 5: 309–317.

Felske, J. D. 1979. Analysis of an evacuated cylindrical solar collector. *Solar Energy* 21: 567–570.

Finegold, J. G., and F. Ann Herlevich. 1980. BALDR-1: A solar thermal system simulation. Solar Energy Research Institute report SERI/TP-351-464, 6 pp.

Fishenden, M., and O. A. Saunders. 1950. *Introduction to Heat Transfer*. Oxford: Oxford University Press.

Fraikin, M. P., T. J. Portier, and C. J. Fraikin. 1980. Application of a k-δ turbulence model to an enclosed buoyancy-driven recirculating flow. ASME paper 80-HT-68.

Francey, J. L., and J. Papaioannou. 1985. Wind related heat losses of a flat plate collector. *Solar Energy* 35: 15–19.

Francia, G. 1962. A new collector of solar radiant energy-theory and experimental verification—Calculation of the efficiencies. SAE paper no. 594B. National Aerospace Engineering and Manufacturing Meeting, Los Angeles, CA, October 8–12, 19 pp.

Fujita, T., W. Revere, J. Biddle, H. Awaya, and D. DeFranco. 1982. Design options for low cost thermal transport piping networks for point focusng parabolic dish solar thermal systems. *Proc. ASES Ann. Meet.*, pp, 339–343.

Garg, H. P. 1982. *Treatise on Solar Energy: Fundamentals of Solar Energy*. Vol. 1. New York: Wiley.

Garg, H. P., R. A. Shukla, R. C. Agnihotri, and S. Chakravertty, and Indrajit. 1983. Study of transmittance in an evacuated flat plate collector with double cylindrical glass cover. *J. Energy Convers. Mgmt.* 23, 1: 33–36.

Garg, H. P., V. K. Sharma, and B. Bandyopadhyay. 1982. Transient analysis of a solar air heater of the second kind. *Energy Convers. Mgmt.* 22: 47–53.

Garg, H. P., and Datta, G. 1984. The top loss calculation for flat plate solar collectors. *Solar Energy* 32, 1: 141–143.

Gilleland, F. W. 1980. Insulation for solar collectors important to heat loss control. *Solar Eng.*: 20–21.

Graham, B. J. 1979. A survey and evaluation of current design of evacuated collectors. Final technical report ALO/5350-1 from Trident Engineering Associates to the USDOE, 178 pp.

Grassie, S. L., and N. R. Sheridan. 1977. The use of planar reflectors for increasing the energy yield of flat-plate collectors. *Solar Energy* 19: 663–668.

Grilikhes, V. A., and F. V. Obtemperanskii. 1969. Analysis of radiative heat exchange processes in cylindrical cavity-type collectors of solar power plants. *Geliotekhnika* 5: 40–48.

Guthrie, K. I., and W. W. S. Charters. 1982. An evaluation of a transverse slatted flate plate collector. *Solar Energy* 28: 89–97.

Harris, J. A., and T. G. Lenz. 1985. Thermal performance of solar concentrator/cavity receiver systems. *Solar Energy* 34: 135–142.

Harrison, T. D. 1982. Program for predicting thermal performance based on test data of low- to medium-temperature line-focusing, concentrating solar collectors. Sandia National Laboratories report SAND-82-0092, 20 pp.

Hasatani, M., N. Arai, Y. Bando, and H. Nakamura. 1982. Collection of thermal radiation by a semitransparent fluid layer flowing in an open channel. *Heat Transfer-Japanese Research* 11: 17–30.

Hassan, K. 1979. Heat transfer through collector glass covers. *Proc. ISES Silver Jubilee Congress*, K. Boer and B. Glenn, eds. Elmsford, NY: Pergamon, pp. 312–316.

Hatfield, D. W., and D. K. Edwards. 1982. Effects of wall radiation and conduction on the stability of a fluid in a finite slot heated from below. *Int. J. Heat Mass Transfer* 25: 1363–1376.

Hewitt, H. C., and E. I. Griggs. 1979. Method to determine and optimal air-flow rate in solar collectors. *J. Energy* 3, 6.

Hewitt, H. C., B. K. Parekh, and G. L. Askew. 1978. Test for determining a best flow rate through solar collectors. *J. Energy* 2, 6: 342–345.

Hildebrandt, A. F., and S. Dasgupta. 1980. Survey of Power Tower Technology. *ASME J. Solar Energy Eng.* 102: 91–104.

Hjertager, B. H., and B. F. Magnussen. 1977. Numerical prediction of three-dimensional turbulent buoyant flow in a ventilated room. In D. B. Spalding and N. Afgan, eds., *Heat Transfer and Turbulent Buoyant Convection*, vol. 2. Washington, DC: Hemisphere, pp. 429–442.

Hoffman, E. W., and V. C. Flannery. 1985. Investigation of flow distribution in solar collector arrays. *Proc. ISES Conf.*, Montreal, Canada, pp. 1008–1012.

Hollands, K. G. T., and K. Iynkaran. 1985. Proposal for a compound-honeycomb collector. *Solar Energy* 34: 309–316.

Hollands, K. G. T., G. D. Raithby, F. B. Russell, and R. G. Wilkinson. 1984. Coupled radiative and conductive heat transfer across honeycomb panels and through single cells. *Int. J. Heat Mass Transfer* 27: 2119–2131.

Hollands, K. G. T. 1973. Natural convection in horizontal thin-walled honeycomb panels. *J. Heat Transfer* 95: 439–444.

Hollands, K. G. T. 1965. Honeycomb devices in flat-plate solar collector. *Solar Energy* 9, 3: 159–164.

Hollands, K. G. T., and L. Konicek. 1965. Experimental study of the stability of differentially heated inclined air layers. *J. Heat Transfer* 87: 1467–1476.

Hollands, K. G. T. 1963. Directional selectivity, emittance, and absorptance properties of vee corrugated specular surfaces. *Solar Energy* 7, 3: 108–116.

Hollands, K. G. T., and E. C. Shewen. 1981. Optimization of flow passage geometry for air-heating, plate-type solar collectors. *J. Solar Energy Engg.* 103, 4: 323–330.

Hollands, K. G. T., T. E. Unny, G. D. Raithby, and L. Konicek. 1976. Free convection heat transfer across inclined air layers. *J. Heat Transfer* 98, 2: 189–193.

Hollands, K. G. T., and J. L. Wright. 1983. Heat loss coefficients and effective products for flat-plate collectors with diathermanous covers. *Solar Energy* 30, 3: 211–216.

Honeywell. 1977. Jet impingement solar air heater. Report on USDOE contract EY-76-C-02-2929 "Low Cost Solar Air Heater."

Hoogendorn, C. J. 1986. Natural Convection in Enclosures. *Heat Transfer 1986, Proc. 8th International Heat Transfer Conference*, vol. 1. Washington, DC: Hemisphere, pp. 111–120.

Hoogendron, C. J. 1985. Natural convection supression in solar collectors. In S. Kakac, W. Aung, and R. Viskanta, eds., *Natural Convection Fundamentals and Applications*. Washington, DC: Hemisphere, pp. 940–960.

Hottel, H. C., and D. D. Erway. 1963. Collection of solar energy. In A. M. Zarem and D. D. Erway, eds., *Introduction to the Utilization of Solar Energy*, New York: McGraw-Hill, ch. 5.

Hottel, M. C., and B. B. Woertz. 1942. Performance of flat plate solar-heat collectors. *Trans. ASME* 64: 91–104.

Howell, J., B. R. Bannerot, and G. C. Vilet. 1982. *Solar-Thermal Energy Systems: Analysis and Design*. New York: McGraw-Hill, chs. 4–5.

Hruby, J. M. 1985. Solid particle receivers: a status report. *Proc. Solar Thermal Research Program Annual Conference*. SERI/CP-251-2680, DE85002938, pp. 79–87.

Humphrey, J. A. C., and E. W. Jacobs. 1981. Free-forced laminar flow convective heat transfer from a square cavity in a channel with variable inclination. *Int. J. Heat Mass Transfer* 24: 1589–1597.

Hunt, A. J. 1978. Small particle heat exchangers. Lawrence Berkeley Laboratory report LBL-7841.

Hunt, A. J., and C. T. Brown. 1983. Solar test results of an advanced direct absorption high temperature gas receiver (SPHER). *Proc. 8th Biennial Congress of ISES*, vol. 2, Perth, Australia, pp. 959–963 (also Report LBL-16497).

Hunt, A. J., Ayer, and P. Hull. 1985. Direct radiant heating of particle suspensions. *Proc. Solar Thermal Research Program Annual Conference*. SERI/CP-251-2680, DE85002938, pp. 139–148.

Hurtado, P., and M. Kast. 1984. Experimental study of direct in-situ generation of steam in a line-focus solar collector. Report DOE/SF/11946-T1 (DE85000497), 71 pp.

Iyican, L., L. C. Witte, and Y. Bayazitoglu. 1981. Natural convection and sidewall losses in trapezoidal groove collectors. *J. Solar Energy Eng.* 103: 167–172.

Janke, S. H. 1983. A three-dimensional model for determining the solar radiation absorbed in a transparent circular cylinder containing a blackened fluid. *Progress in Solar Energy*, vol. 6. American Solar Energy Society Annual Meeting, pp. 459–464.

Jones, G. F. 1981. Heat transfer and flow distribution within radiant-convective finned-tube manifold assemblies. Ph.D. dissertation. Dept. Mech. Engrg. & Appl. Mech., University of Pennsylvania, Philadelphia.

Jones, G. F., and N. Lior. 1979. Optimal insulation for solar heating system pipes and tanks. *Energy*, 4: 593–621.

Jones, G. F., and N. Lior. 1978. Isothermal flow distribution in solar collectors and collector manifolds. *Proc. Annual Meeting of the AS/ISES*, vol. 2.1, Denver, co, pp. 362–372.

Jones, G. F., and N. Lior. 1987. Conjugate heat transfer and flow distribution in an essembly of manifolded finned tubes. ASME Paper.

Jones, R. E., and J. F. Burkhart. 1981. Shading effects of collector rows tilted toward the equator. *Solar Energy* 26: 563–565.

Kaehn, H. D., M. Geyer, D. Fong, F. Vignola, and D. K. McDaniels. 1978. Experimental evaluation of the reflector-collector system. *Proc. AS/ISES Ann. Meet.*, vol. 2.1, pp. 654–659.

Kamminga, W. 1985. The approximate temperature within a flat-plate solar collector under transient conditions. *Int. J. Heat Mass Transfer* 28: 433–440.

Kim, D. M., and R. Viskanta. 1984. Effect of wall heat conductance and radiation on natural convection in a rectangluar cavity. *Num. Heat Transfer* 7: 449–470.

Kim, D. M., and R. Viskanta. 1985. Effect of wall heat conduction on natural convection in a square enclosure. *J. Heat Transfer* 107: 139–146.

Kind, R. J., D. H. Gladstone, and A. D. Moizer. 1983. Convective heat losses from flat-plate solar collectors in turbulent winds. *J. Solar Energy Eng.* 105: 80–85.

Kind, R. J., and D. Kitaljevich. 1985. Wind-induced heat losses from solar collector arrays on flat-roofed buildings. *ASME J. Solar Energy Eng.* 107: 335–342.

Klein, S. A., J. A. Duffie, and W. A. Beckman. 1973. Transient considerations of flat-plate solar collectors. *ASME J. Eng. for Power* 96A: 109–113.

Knowles, G. R., and E. M. Sparrow. 1979. Local and average heat transfer characteristics for turbulent airflow in an asymmetrically heated tube. *J. Heat Transfer* 101: 635–641.

Kovarik, M. 1978. Optimal distribution of heat conducting material in the finned pipe solar energy collector. *Solar Energy* 21: 477–484.

Kovarik, M., and P. F. Lesse. 1976. Optimal control of flow in low temperature solar heat collectors. *Solar Energy* 18: 431–435.

Kraabel, J. S. 1983. An experimental investigation of the natural convection from a side-facing cubical cavity. *ASME-JSME Thermal Engineering Joint Conference*, vol. 1, pp. 299–306.

Kreith, F., and J. F. Kreider. 1981. Nonconcentrating solar collectors. In J. F. Kreider and F. Kreith eds., *Solar Energy Handbook*. New York: McGraw-Hill.

Kreith, F., and J. F. Kreider. 1978. *Principles of Solar Engineering*. Washington DC: Hemisphere.

Kreith, F., and R. Anderson. 1985. Natural convection in solar systems. In S. Kakac, W. Aung and R. Viskanta, eds., *Natural Convection: Fundamentals and Applications*. Washington, DC: Hemisphere, pp. 881–939.

Kuehn, T. H., and R. J. Goldstein. 1980. A parametric study of Prandtl number and diameter ratio effects on natural convection heat transfer in horizontal cylindrical annuli. *J. Heat Transfer* 102: 768–770.

Kuehn, T. H., and R. J. Goldstein. 1978. An experimental study of natural convection heat transfer in concentric and eccentric horizontal cylinders. *J. Heat Transfer* 100: 635–640.

Kuehn, T. H., and R. J. Goldstein. 1976. Correlating equations for natural convection heat transfer between horizontal circular cylinders. *Int. J. Heat Mass Transfer* 19: 1127–1134.

Kuehn, T. H., and R. J. Goldstein. 1976. An experimental and theoretical study of natural convection in the annulus between horizontal concentric cylinders. *J. Fluid Mech.* 74: 695–719.

Kugath, D. A., G. Drenker, and A. A. Koenig. 1979. Design and development of a paraboloidal dish solar collector for intermediate temperature service. *Proc. ISES Silver Jubilee*, vol. 2, Atlanta, GA, pp. 449–453.

Lalude, O., and H. Buchberg. 1971. Design and application of honeycomb porous-bed solar-air heaters. *Solar Energy* 13: 223–242.

Landstrom, D. K., G. H. Stickford, Jr., S. G. Talbert, and R. E. Hess. 1978. Development of a low-temperature, low-cost solar collector using a black-liquid concept. *Proceedings of the 1978 Annual Meeting, American Section of the International Solar Energy Society, Inc.*, vol. 2.1, Denver, CO, pp. 228–233.

Landstrom, D. K., S. G. Talbert, and V. D. McGinniss. 1980. Development of a low-temperature, low-cost black liquid solar collector phase II. *Proc. Ann. DOE Active Solar Heating and Cooling Contractors' Review Meeting.*

Lansing, F. L. 1979. HEAP: Heat energy analysis program—A computer model simulating solar receivers. Report DOE/JPL-1060-13 (JPL publication 79-3).

Lansing, F. L., V. Clarke, and R. Reynolds. 1979. A high performance porous flat plate solar collector. *Energy* 4: 685–694.

Larson, D. C. 1980a. Optimization of flat-plate collector-flat mirror systems. *Solar Energy* 24: 203–207.

Larson, D. C. 1986b. Concentration ratios for flat plate solar collectors with adjustable flat mirrors. *AIAA J. Energy* 4: 170–175.

Leary, P. L., and J. D. Hankins. 1979. A user's guide for MIRVAL—A computer code for comparing designs of heliostat-receiver otpics for central receiver solar power plants. Sandia National Laboratories Report SAND-77-8280.

Lee, B. E. 1987. Wind flow over roof-mounted flat plate collectors. *Solar Energy* 38: 335–340.

Lee, T. S., N. E. Wijeysundera, and K. S. Yeo. 1984. Free convection fluid motion and heat transfer in horizontal concentric and eccentric cylindrical collector systems. In D. Y. Lowani, ed., *Solar Engineering 1984*. ASME, New York, pp. 194–200.

Lewandowski, A. A. 1985. Direct absorption receiver system studies. *Proc. Solar Thermal Research Program Annual Conference*. SERI/CP-251-2680, DE85002938, pp. 107–117.

Lin, N. N., and A. Bejan. 1983. Natural convection in a partially divided enclosure. *Int. J. Heat Mass Transfer* 26: 1867–1878.

Lior, N. 1986. Natural convection in high technology applications. In W.-J. Yang and Y. Mori, eds., *Heat Transfer in High Technology and Power Engineering*. Washington, DC: Hemisphere, pp. 573–596.

Lior, N., J. O'Leary, and D. Edelman. 1977. Optimized spacing between rows of solar collectors. *Proc. Annual Meeting, American Section of the International Solar Energy Society*, vol. 1, Orlando, FL, pp. 3-15–3-20.

Lior, N., H. Ozoe, P. Chao, G. F. Jones, and S. W. Churchill. 1983. Heat transfer considerations in the use of new energy resource. In T. Mizushina and W.-J. Yang, eds., *Heat Transfer in Energy Problems*. New York: Hemisphere/Springer-Verlag, pp. 175–188.

Lior, and N., and A. P. Saunders. 1973. Solar collector performance studies. University of Pennsylvania report NSF/RANN/SE/G127976/TR73/1 to NSF/RANN, 114 pp.

Lior, N., and R. N. Segall. 1986. Heat transfer with air flow across an array of plates with angle of attack. AIAA paper 86-1365. AIAA/ASME 4th Thermophysics and Heat Transfer Conference, Boston, MA.

Lipps, F. W. 1983. Geometric configuration factors for polygonal zones using Nusselt's unit sphere. *Solar Energy* 30: 413–419.

Liu, C. H., and E. M. Sparrow. 1980. Convective-radiative interaction in a parallel plate channel—Application to air-operated solar collectors. *Int. J. Heat Mass Transfer* 23: 1137–1146.

Liu, B. Y. H., and R. C. Jordan. 1965. Performance and evaluation of concentrating solar collectors for power generation. *Trans. ASME J. Eng. for Power* 87: 1–7.

Liu, B. Y. H., and J. C. Jordan. 1963. A rational procedure for predicting the long-term average performance of flat-plate solar-energy collectors. *Solar Energy* 7, 2: 53–74.

Liu, B. Y. H., and R. C. Jordan. 1961. Daily insolation on surfaces tilted toward the equator. *ASHRAE J.* 3: 53–59.

Löf, G. O. G., M. M. El Wakil, and J. A. Duffie. 1961. The performance of Colorado solar house. *United Nations Conference on New Sources of Energy*, Rome.

Löf, G. O. G. 1980. Flat-plate and nonconcentrating collectors. In W. C. Dickinson and P. N. Cheremisinoff, eds., *Solar Energy Technology Handbook*, part A. New York: Marcel Dekker, ch. 9.

Lou, Y. S., and T. K. Shih. 1972a. Nonlinear heat conduction in rarefied gases. *AIAA J.* 10: 1149–1150.

Lou, Y. S., and T. K. Shih. 1972b. Nonlinear heat conduction in rarefied gases confined between concentric cylinders and spheres. *Phys. Fluids* 15: 785–788.

Lumsdaine, E. 1969. Solar Heating of a fluid through a semi-transparent plate: Theory and experiment. *Solar Energy* 12: 457–467.

Lumsdaine, E. 1970. Transient solution and criteria for achieving maximum fluid temperature in solar energy applications. *Solar Energy* 13: 3–19.

Mansfield, R. G., and A. Eden. 1978. The application of thermography to large arrays of solar energy collectors. *Solar Energy* 21: 533–537.

Marcus, S. L. 1983. An approximate method for calculating the heat flux through a solar collector honeycomb. *Solar Energy* 30, 2: 127–131.

Markatos, N. C., and K. A. Pericleous. 1984. Laminar and turbulent natural convection in an enclosed cavity. *Int J. Heat Mass Transfer* 27: 755–772.

Marshall, K. N., R. K. Wedel, and R. E. Dammann. 1976. Development of plastic honeycomb flat-plate solar collectors. Final report SAN/1081-76/1 to the USDOE. Lockheed Missile Co. LMSC/D462879, 178 pp.

Mather, G. R., Jr. 1982. Transient response of solar collectors. *ASME J. Solar Energy Eng.* 104: 165–172.

May, E. K., and L. M. Murphy. 1983. Performance benefits of the direct generation of steam in line-focus solar collectors. *J. Splar Energy Eng.* 105, 2: 126–133.

McAdams, W. H. 1954. *Heat Transmission*. 3d ed. New York: McGraw-Hill.

McCumber, W. H., and M. W. Weston. 1979. The analysis and comparison of actual to predicted collector array performance. *IBM J. R&D* 23: 239–358.

McDaniels, D. K., D. H. Lowndes, H. Mathew, J. Reynolds, and R. Gray. 1975. Enhanced solar energy collection using reflector-solar thermal collector combinations. *Solar Energy* 17: 277–283.

McKinnon, R. A., A. D. Turrin, and G. L. Schrenk. 1965. Cavity receiver temperature analysis. Paper no. 65–470. AIAA Second Annual Meeting, 24 pp.

McMordie, R. K. 1984. Convection heat loss from a cavity receiver. *J. Solar Energy Eng.* 106, 1: 98–100.

McMurrin, J. C., N. A. Djordjevic, and H. Buchberg. 1977. Performance measurements of a cylindrical glass honeycomb solar collector compared with predictions. *J. Heat Transfer* 99C, 2: 169–173.

Menuchin, Y., S. Bassler, G. F. Jones, and N. Lior. 1981. Optimal flow configuration in solar collector arrays. *Proc. of the 1981 Annual Meeting of the AS/ISES*, vol. 4.1, Philadelphia, PA, pp. 616–620.

Meyer, B. A., M. M. El-Wakil, and J. W. Mitchell. 1979. Natural convection heat transfer in small and moderate aspect ratio enclosures—An application to flat plate collectors. *J. Heat Transfer* 101: 655–659.

Meyer, B. A., J. M. Mitchell, and M. M. El-Wakil. 1980. Convective heat transfer in trough and CPC collectors. *Proc. Ann Meet. AS/ISES*, vol. 3.1, pp. 437–440.

Miller, K. W. 1943 (July). Solar heat trap. Memorandum to the Office to Production, Research and Development, WPB.

Minardi, J. E., and H. N. Chuang. 1975. Performance of a "black" liquid flat-plate solar collector. *Solar Energy* 17: 179–183.

Mirenayat, J. 1981. Experimental study of heat loss through natural convection from an isothermal cubic open cavity. SANDIA Report 81-8014. Sandia National Laboratories, Livermore, CA, pp. 165–174.

Mitalas, G. P., and D. G. Stephenson. 1962. Absorption and transmission of thermal radiation by single and double glazed windows. Research paper no. 173. Division of Building Research of National Research Council, Ottawa, Canada.

Morcos, S. M., and M. M. M. About-Ellail. 1983. Buoyancy effects in the entrance region of an inclined multirectangular-channel solar collector. *J. Solar Energy and Eng.* 105, 2: 157–162.

Morrison, G. L., and D. B. J. Ranatunga. 1980. Transient response of thermosyphon solar collectors. *Solar Energy* 24: 55–61.

Mull, W., and H. Reiher. 1930. Der Warmeschutz von Luftschichten. *Beiheft zum Gesund. Ing. Beiheft*, vol. 28, no. 1, 26 pp.

Nahar, N. M., and H. P. Garg. 1980. Free convection and shading due to gap spacing between an absorber plate and the cover glazing in solar energy flat-plate collectors. *Applied Energy*, 7, 1-3: 129–145.

Naidu, M. G., and J. P. Agarwal. 1981. A theoretical study of heat Transfer in a flat-plate solar collector. *Solar Energy*, 26, 4: 313–323.

Nansteel, M., and R. Greif. 1984. An investigation of natural convection in enclosures with two- and three-dimensional partitions. *Int. J. Heat Mass Transfer* 27: 561–571.

Oliphant, M. V. 1979. Measurement of wind speed distributions across a solar collectors. *Solar Energy* 24: 403–405.

Onur, N., and J. C. Hewitt, Jr. 1980. A study of wind effects on collector performance, ASME paper 80-C2/Sol-4, 6 pp.

Ostrach, S. 1982. Natural convection heat transfer in cavities and cells. *7th Int. Heat Transfer Conf.*, vol. 1. Munich, pp. 365–379.

Ostrach, S. 1972. Natural convection in enclosures. In J. P. Hartnett and T. F. Irvine, Jr., eds., *Advances in Heat Transfer*, vol. 8. New York: Academic Press, ch. 12.

Ota, T., and N. Kon. 1979. Heat transfer in the separated and reattched flow over blunt flat plates—Effects of nose shape. *Int. J. Heat Mass Transfer* 22: 197–206.

Ozoe, H., P. K.-B Chao, S. W. Churchill, and N. Lior. 1983. Laminar natural convection in an inclined rectangular box with the lower surface half heating and half insulated, *ASME J. Heat Transfer* 105: 425–442.

Ozoe, H., K. Fujii, N. Lior, and S. W. Chruchill. 1982. A theoretically based correlation for natural convection in horizontal rectangular enclosures heated from below with arbitrary aspect ratios. Paper no. NC23. *7th International Heat Transfer Conference*, vol. 2. Munich. Washington, DC: Hemisphere, pp. 257–262.

Ozoe, H., K. Fujii, N. Lior, and S. W. Churchill. 1983b. Long rolls generated by natural convection in an inclined rectangular enclosure. *Int. J. Heat Mass Transfer* 26, 10: 1427–1438.

Ozoe, H., K. Fujii, T. Shibata, H. Kuriyama, and S. W. Churchill. 1985a. Three-dimensional numerical analysis of natural connvectional in a spherical annulus. *Numerical Heat Transfer* 8: 383–406.

Ozoe, H., A. Mouri, M. Hiramitsu, S. W. Churchill, and N. Lior. 1986. Numerical calculation of three-dimensional turbulent natural convection in a cubical enclosure using a two-equation model for turbulence. *J. Heat Transfer* 108: 806–813.

Ozoe, H., A. Mouri, M. Ohmuro, S. W. Churchill, and N. Lior. 1985b. Numerical calculations of laminar and turbulent natural convection of water in rectangular channels heated and cooled isothermally on the opposing vertical walls. *Int. J. Heat Mass Transfer* 28, 1: 125–138.

Ozoe, H., M. Ohmuro, A. Mouri, S. Mishima, H. Sayama, and S. W. Churchill. 1983a. Laser-doppler measurements of the velocity along a heated vertical wall of a rectangular enclosure. *J. Heat Transfer* 105: 782–783.

Ozoe, H., T. Okamoto, and H. Sayama, and S. W. Churchill. 1978. Natural convection in doubly inclined rectangular boxes. *Proc. 6th International Heat Transfer Conference*, vol. 2. Washington, DC: Hemisphere, pp. 293–298.

Ozoe, H., H. Sayama, and S. W. Churchill. 1977a. Natural convection patterns in along inclined rectangular box heated from below. Part I: Three dimensional photography. *Int. J. Heat Mass Transfer* 20: 123–129.

Ozoe, H., H. Sayama, and S. W. Churchill. 1977b. Natural convection patterns in along inclined rectangular box heated from below. Part II: Three dimensional numerical results. *Int. J. Heat Mass Transfer* 20: 131–139.

Ozoe, H., T. Shibata, and S. W. Churchill. 1981a. Natural convection in an inclined circular cylindrical annulus heated and cooled on its end plates. *Int. J. Heat Mass Transfer* 24: 727–737.

Ozoe, H., K. Yamamoto, and S. W. Churchill. 1979. Three-dimensional numerical analysis of natural convection in an inclined channel with a square cross section. *AIChE J.* 25: 709–716.

Ozoe, H., K. Yamamoto, S. W. Churchill, and H. Sayama. 1976. Three-dimensional, numerical analysis of laminar natural convection in a confined fluid heated from below. *J. Heat Transfer* 98C: 202–207.

Parmelee, G. V. 1955. Application of thermal circuit analysis to collector design. *Trans. Int. Conf. on the Use of Solar Energy: The Scientific Basic*, vol. 2, pt. 1, Sec. A. Tucson, AZ, pp. 115–119.

Pellette, P., M. Cobble, and P. Smith. 1968. Honeycomb thermal trap. *Solar Energy* 12: 263–265.

Perrot, M., P. Gallet, G. Peri, J. Desautel. and M. Touchaies. 1967. Les structures cellulaires antirayonnantes et leurs applications industrielles. *Solar Energy* 11, 1: 34–38.

Phillips, W. F. 1982. A simplified nonlinear model for solar collectors. *Solar Energy* 29, 1: 77–82.

Phillips, W. F. 1979. The effects of axial conduction on collector heat removal factor. *Solar Energy* 23: 187–191.

Pons, R. L. 1980. Optimization of a point-focusing distributed receiver solar thermal electric system. *ASME J. Solar Energy Eng.* 102: 272–280.

Rabl, A. 1985. *Active Solar Collectors and Their Applications*. Oxford: Oxford University Press.

Rabl, A. 1981. Yearly average performance of the principal solar collector types. *Solar Energy* 27, 3: 215–234.

Rabl, A. 1981. Intermediate concentrating solar collectors. In J. F. Kreider and F. Kreith, eds., *Solar Energy Technology Handbook*. New York: McGraw-Hill, ch. 8.

Rabl, A. 1980. Concentrating collectors. In W. C. Dickinson and P. N. Cheremisinoff, eds., *Solar Energy Technology Handbook*, pt. A. New York: Marcel Dekker, ch. 10.

Rao, P. P., J. E. Francis, and T. J. Love, Jr. 1977. Two-dimensional analysis of a flat-plate solar collector. *J. Energy* 1, 5: 324–328.

Ratzel, A. C., C. E. Hickox, and D. K. Gartling. 1979. Techniques for reducing thermal conduction and natural convection heat losses in annular receiver geometries. *J. Heat Transfer* 101: 108–113.

Rice, M. P., M. F. Modest, D. N. Borton, and W. E. Rogers. 1981. Heat transfer analysis of receivers for a solar concentrating collectors. *Chem. Engineering Comm.*, 8, 4-6: 353–365.

Robinson, H. E., and F. J. Powlitch. 1954. Thermal insulation values of airspaces. U. S. Housing and Home Finance Agency, Division of Housing, research paper no. 32. April.

Rozkov, I. A. 1977. Study of the characterictics of convective heat transfer in cylindrical solar receivers. *Geliotekhnika* 2: 56–63.

Safdari, Y. B. 1966. Radiation heating through transparent and opaque walls. *Solar Energy* 10: 53–58.

Saito, A., Y. Utaka, T. Tsuchio, and K. Katayama. 1984. Transient response of flat plate solar collector for periodic solar intensity variation. *Solar Energy* 32, 1: 17–23.

Saltiel, C. J., and M. Sokolov. 1982. Thermal and optical analysis of an evacuated circular cylindrical concentrating collector. *Solar Energy* 29: 391–396.

Samano, A., and A. Fernandez. 1983. Simulation of a solar evacuated collector with black fluid. *Progress in Solar Energy*, vol. 6. American Solar Energy Society Annual Meeting, pp. 377–380.

Sassi, G. 1983. Some notes on shadow and blockage effect. *Solar Energy* 31, 3: 331–333.

Sateunanathan, S., and P. Gandhidasan. 1981. The role of rear insulation in the performance of flat plate solar collectors. *ASME J. Solar Energy and Engineering* 103: 282–284.

Savery, C. W., P. M. Anderson, and D. C. Larson. 1976. Double-exposure collectors with mirrors for solar-heating systems. ASME paper no. 76-WA/HT-16, 8 pp.

Savery, C. W., and D. C. Larson. 1978. Performance of a double-exposure solar collector. *Proc. AS/ISES Ann. Meet.*, vol. 2.1, pp. 649–653.

Schmidt, R. R., and E. M. Sparrow. 1978. Turbulent flow of water in a tube with circumferentially nonuniform heating, with or without buoyancy. *J. Heat Transfer* 100: 403–409.

Seitel, S. C. 1975. Collector performance enhancement with flat reflectors. *Solar Energy* 17: 291–295.

Selcuk, M. K. 1971. Thermal and economic analysis of the overlapped-glass plate solar-air heater. *Solar Energy* 13: 165–191.

Selcuk, M. K. 1964. Flat-plate solar collector performance at high temperatures. *Solar Energy* 8, 2: 57–62.

Shah, R. K., and A. L. London. 1978. *Laminar Flow Forced Convection in Ducts*. New York: Academic Press.

Sharafi, A. Sh., and A. G. Mukminova. 1975. Procedure for computing the reflectively, absorptivity, and transmission coefficient for radiant energy in multillayer systems with varying optical properties. *Geliotekhnika* 11: 2: 50–54.

Shoemaker, M. J. 1961. Notes on a solar collector with unique air permeable media. *Solar Energy* 5, 4: 138–141.

Shouman, A. R., and I. A. Tag. 1981. Method of testing solar collectors for determination of heat transfer parameters. Solar Energy Division of the ASME, Winter Annual Meeting, Washington, DC, November 15–20, pp. 1–5.

Shurcliff, W. A. 1974. Transmittance and reflection loss of multi-plate planar window of a solar-radiation collector: Formulas and tabulation of results for the case $n = 1.5$. *Solar Energy* 16: 149–154.

Siebers, D. L., and J. S. Kraabel. 1984. Estimating convective energy losses from solar central [illegible]

Solar Thermal Collector Arrays, C. A. Bankston, comp. Solar Energy Research Institute report SERI/SP-271-2664. San Diego, CA, pp. 401–440.

Siebers, D. L., R. L. Moffat, and R. G. Schwindt. 1983. Experimental, variable properties natural convection from a large vertical flat surface. *ASME-JSME Thermal Engineering Joint Conference*, vol. 3, pp. 269–275.

Siebers, D. L., R. G. Schwindt, and R. J. Moffat. 1982. Experimental mixed convection from a large vertical plate in a horizontal flow. *Proc. 7th International Heat Transfer Conference*. Paper MC 13.

Siebers, D. L., and R. Viskanta. 1979. Thermal analysis of some flat-plate solar collector designs for improving performance. *J. Energy* 3: 8–15.

Siebers, D. L., and R. Viskanta. 1978. Some aspects of the transient response of a flat plate solar energy collector. *Energy Conversion* 18: 135–139.

Siegel, R. 1973. Net radiation method for transmission through partially transparent plates. *Solar Energy* 15: 273–276.

Siegel, R., and J. R. Howell. 1981. *Thermal Radiation Heat Transfer*. 2d ed. New York: McGraw-Hill.

Smart, D. R., K. G. T. Hollands, and G. D. Raithby. 1980. Free convection heat transfer across rectangular-called diathermaneous honeycomb. *J. Heat Transfer* 102: 75–80.

Sobin, A., W. Wagner, and C. R. Easton. 1974. Central collector solar energy receivers. *Proc. AS/ISES Ann. Meet.*, 44 pp.

Soin, R. S., K. Sangameswar Rao, D. P. Rao, and K. S. Rao. 1979. Performance of flat plate solar collector with fluid undergoing phase change. *Solar Energy* 23: 69–73.

Souka, A. F. 1965. Double exposure flat-plate collector. *Solar Energy* 9, 3: 117–118.

Souka, A. F., and H. H. Safwat. 1969. Theoretical evaluation of the performance of a double exposure flat-plate collector using a single reflector. *Solar Energy* 12: 347–352.

Souka, A. F., and H. H. Safwat. 1966. Determination of the optimum orientation for the double exposure flat-plate collector and its reflectors. *Solar Energy* 10: 170–174.

Sparrow, E. M., W. J. Bifano, and J. A. Healy. 1972. Efficiencies of honeycomb absorbers of solar radiation. *J. Spacecraft* 9, 2: 67–68.

Sparrow, E. M., and R. J. Kroweeh. 1977. Circumferential variations of bore heat flux and outside surface temperature for a solar collector tube. *J. Heat Transfer* 99, 3: 360–366.

Sparrow, E. M., J. S. Nelson, and W. Q. Tao. 1982. Effect of leeward orientation, adiabtaic framing surfaces, and eaves on solar-collector-related heat transfer coefficients. *Solar Energy* 29: 33–41.

Sparrow, E. M., J. W. Ramsey, and E. A. Mass. 1979. Effect of finite width on heat transfer and fluid flow about an inclined rectangular plate. *J. Heat Transfer* 101: 199–204.

Sparrow, E. M. and K. K. Tien. 1977. Forced convection heat transfer at an inclined and yawed square plate—Applications to solar collectors. *J. Heat Transfer* 99, 4: 507–512.

Speyer, E. 1965. Solar energy collection with evacuated tubes. *Trans. ASME J. Eng. for Power*, pp. 270–276.

Stephenson, D. G. 1965. Equations for solar heat gain through windows. *Solar Energy* 9, 2: 81–86.

Stokes, G. G. 1982. On the intensity of the light reflected from or transmitted through a pile of plates. *Proc. Roy. Soc. (London)* 11: 545–556.

Strumpf, H. J., D. M. Kotchick, and M. G. Coombs. 1982. High temperature ceramic heat exchanger element for a solar thermal receiver. *Parabolic Dish Solar Thermal Power Annual Program Review Proceedings*. Report DOE/JPL-1060-52, pp. 233–246. Reprinted in *J. Solar Energy Eng.* 104: 305–309.

Swartman, R. K., and O. Ogunlade. 1966. An investigation on packed-bed collectors. *Solar Energy*

Symons, J. G. 1982. The solar transmittance of some convection suppression devices for solar energy applications: An exerimental study. *ASME J. Solar Energy Eng.* 104: 251–256.

Symons, J. G., and R. Gani. 1079. Thermal performance predictions and sensitivity analysis for high temperature flat-plate solar collectors. *Solar Energy* 24: 407–410.

Symons, J. G., and M. K. Peck. 1984. An overview of the CSIRO project on adavnced flat-plate solar collectors. *Proc. ISES Biennial Meeting, Solar World Congress*, vol. 2. Perth, Australia, 1983, New York: Pergamon, pp. 748–752.

Tabor, H. 1969. Cellular insulation (honeycombs). *Solar Energy* 12: 549–552.

Tabor, H. 1966. Mirror boosters for solar collectors. *Solar Energy* 10: 111–118.

Tabor, H. 1958. Radiation, convection and conduction coefficients in solar collectors. *Bull. Res. Counc. of Israel* 6C: 155–176.

Tabor, H. 1955. Solar Energy collector design with special reference to selective radition. *Bull. Res. Counc. of Israel* 5C: 5–27.

Tan, H. M., and W. W. S. Charters. 1970. An experimental investigation of forced-convective heat transfer for fully-developed turbulent flow in a rectangular duct with asymmetric heating. *Solar Energy* 13: 121–125.

Test, F. L. 1976. Parametric study of flat plate solar collectors. *Energy Conversion* 16: 23–33.

Test, F. L., and R. C. Lessman. 1980. An experimental study of heat transfer during forced convection over a rectangular body. *J. Heat Transfer* 102: 146–151.

Test, F. L., R. C. Lessman, and A. Johary. 1981. Heat transfer during wind flow over rectangular bodies in the natural environment. *J. Heat Transfer* 103: 262–267.

Thodos, G. 1976. Predicted heat transfer performance of an evacuated glass-jacketed CPC receiver: Countercurrent flow design. Argonne National Laboratory Report ANL-76-67, 37 pp.

Thomas, J. R. 1981. Helium penetration in evacuated solar collectors: Theory and the effect on their performacne. *ASME J. Solar Energy Eng.* 103: 175–177.

Thomas, J. R., Jr. 1979. Heat conduction through partial vacuum. U.S.D.O.E. Report ALO/5367-1.

Thomas, J. R., Jr., T. S. Chang, and C. E. Siewert. 1973. Heat transfer between parallel plates with arbitrary surface accommodation. *Phys. Fluids* 16: 2116–2120.

Tien, C. L., and W. W. Yuen. 1975. Radiaton characteristics of honeycomb solar collectors. *Int. J. Heat Mass Transfer* 18: 1409–1413.

Trentelman, J., and P. H. Wojciechowski. 1977. Performance analysis of a black liquid absorbing collector (BLAC). *Proc. AS ISES Ann. Meet.* 1: 1-21–1-25.

Truncellito, N., H. Yeh, and N. Lior. 1987. Numerical solutions of turbulent convection over a flat plate with angle of attack. *J. Heat Transfer* 109: 238–242. Longer version also published as ASME paper 83-WA HT-3, 1983.

U.S. Post Office. 1969. Computer program for analysis of energy utilization in post facilities Report on Project No. 67138.

Umarov, G. Ya., R. R. Avezov, F. Soatov, and K. Babakulov. 1975. Determination of the entrance coefficient for solar rays transmitted by a glazed cover of a hot-box type solar installation. *Geliotekhnika* 11, 3-4: 70–73.

Vaxman, M., and M. Sokolov. 1985. Analysis of a free-flow solar collector. *Solar Energy* 35: 287–290.

Veinberg, V. B. 1959. *Optics in Equipment for the Utilisation of Solar Energy*. State Publishing House of Defense Ministry, Moscow. Trans. by U.S. Department of Army Intelligence Translation 44787, or USAEC Translation AEC-tr-4471.

Viskanta, R. 1982. Radiation heat transfer: interaction with conduction and convection and approximate methods in radiation. *7th Int. Heat Transfer Conf.*, vol. 1. Munich. pp. 103–121.

Viskanta, R., and E. E. Anderson. 1975. Heat transfer in semitransparent solids. In *Advances in Heat Transfer*, vol. 11. New York: Academic Press, pp. 317–441.

Viskanta, R., D. L. Siebers, and R. P. Taylor. 1978. Radiation characteriestics of multiple-plate glass system. *Int. J. Heat Mass Transfer* 21: 815–818.

Vittitoe, C. N., and F. Biggs. 1981. User's guide to HELIOS: A computer program for modeling the optical behavior of reflecting solar concentrators. Part I. Introduction and code input. Sandia National Laboratory Report SAND-81-1180, 64 pp.

Wagner, T. R., W. G. Houf, and F. P. Incropera. 1980. Radiative property measurements for india ink suspensions of varying concentrations. *Solar Energy* 25, 6: 549–554.

Wang, K. Y., and F. Kreith. 1986. Heat transfer research for high temperature solar energy systems. In W.-J. Yang and Y. Mori, eds., *Heat Transfer in High Technology and Power Engineering*. Washington, DC: Hemishpere, pp. 541–560.

Wang, Y. Y., and R. J. Copeland. 1984. Heat transfer in a solar radiation absorbing molten salt film flowing over an insulated substrate. ASME Paper 84-WA/Sol-22.

Webb, B. W., and R. Viskanta. 1985. Analysis of heat transfer and solar radiation absorption in an irradiated thin, falling molten salt film. *ASME J. Solar Energy Eng.* 107: 113–119.

Whillier, A. 1953. Solar energy collection and its use in house heating. Sc.D. dissertation. Department of Mechanical Engineering, MIT.

Whillier, A. 1963. Plastic covers for solar collectors. *Solar Energy* 7, 3: 148–151.

Whillier, A. 1964a. Performance of black-painted solar air heaters of conventional design. *Solar Energy* 8, 1: 31–37.

Whillier, A. 1964b. Thermal resistance of the tube-plate bond in solar heat collectors. *Solar Energy* 8, 3: 95–98.

Whillier, A., and G. Saluja. 1965. Effects of materials and construction details on the thermal performance of solar water heaters. *Solar Energy* 9, 1: 21–26.

Whillier, A. 1967. Design factors influencing collector performance. In R. C. Jordan, ed., *Low Temperature Engineering Application of Solar Energy*. New York: ASHRAE, ch. 3.

Whillier, A. 1977. Prediction of performance of solar collectors. In R. C. Jordan and B. Y. H. Liu, eds., *Applications of Solar Energy for Heating and Cooling of Buildings*. NY: ASHRAE GRP 170, ch. 8.

Wideman, D. C., and J. R. Thomas, Jr. 1980. Heat conduction in rarefied gases between concentric cylinders: Arbitrary accommodation on both surfaces. *Proc. 1980 Heat Transfer and Fl. Mech. Inst.*, Gerstein and Choudhury, eds., Stanford: Standford University Press, pp.186–195.

Wijeysundera, N. E. 1978. Comparison of transient heat transfer models for flat plate collectors. *Solar Energy* 21, 6: 517–521.

Wijeysundera, N. E. 1976. Response time of solar collectors. *Solar Energy* 21: 517–521.

Wijeysundera, N. E. 1975. A net radiation method for the transmittance and absorptivity of a series of parallel regions. *Solar Energy* 17: 75–77.

Window, B., and L. M. Bassett. 1981. Optical collection efficiencies of arrays of tubular collectors with specular reflectors. *Solar Energy* 26: 341–346.

Window, B., and J. Zybert. 1981. Optical collection efficiencies of arrays of tubular collectors with diffuse collectors. *Solar Energy* 26: 325–331.

Winn, R. C., and C. B. Winn. 1981. Optimal control of mass flow rates in flat plate solar collectors. *ASME J. Solar Energy Eng.* 103: 113–120.

Wolf, D., A. I. Kudish, and A. N. Sembira. 1981. Dynamic simulation and parametric sensitivity studies on a flat-plate solar collector. *Energy* 6, 4: 333–351.

Wu, T. V. Narayana, and D. N. Gorman. 1983. Conceptual design of an advanced wateŋ receiver for a solar thermal central power system. *J. Solar Energy Engineering* 105: 34–4

Yang, and J. R. Lloyd. 1985. Natural convection-radiation interaction in enclosures. In S. Ka/. Aung, and R. Viskanta, eds., *Natural Convection Fundamentals and Applications*. Wash, DC: Hemisphere, pp. 381–410.

Yeh, and M. Wiener. 1984. Thermal analysis of a solar advanced water/steam receiver. *ASM lar Energy Engineering* 106: 44–49.

Young, W. W., and P. I. Mattox. 1978. Evaluation of heat transfer enhancement in air heatinctors. *Proc. 3rd Ann. Solar Heating and Cooling Cotractors' Meeting*, pp. 106–107.

Young, W. W., W. Schultz, and R. Barber. 1979. Solar collector fluid parameter study. Natiorcau of Standards Report NBS-GCR 79-184.

Yuen, . F. J. Miller, and A. J. Hunt. 1986. Heat transfer characteristics of small particle/gas mixtur. *Comm. Heat Mass Transfer* 13: 145–154.

Zhang, and Z. Lavan. 1983. Thermal performance of a serpentine absorber plate. *Progress in Solagy*, vol. 6. Proc. ASES Annual Meeting, pp. 471–476.

Zukau. 1985. *Heat Transfer of a Cylinder in Crossflow*. Washington, DC: Hemisphere.