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Thermal resistance of a solar-energy collector absorber under a non-uniform flux distribution

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Abstract—A generalized method for determining thermal resistances of media experiencing non-uniform flux distributions has been evolved. The method was applied to the analysis of heat transfers in a finned tube, employed as an absorber in a solar-energy collector. The present study demonstrates that, under non-uniform flux distribution, the values of thermal resistances predicted when assuming a uniform flux distribution are generally lower than the actual values encountered. However, for a finned-tube absorber in a solar-energy collector, this is compensated (either partly or overwhelmingly, depending on the particular characteristics of the collector), by the counteracting effect of the distributed nature of the device, which results in an axial variation of temperature.

1. INTRODUCTION

In most solar-energy collectors a temperature difference ensues during operation between the absorber surface, where the solar radiation is collected, and the working fluid. This is due to the finite thermal resistance of the elements through which the collected flux passes before reaching the working fluid. The effect of this temperature drop on the collector's thermal performance can be expressed by an efficiency factor F', which is defined as the ratio of the actual useful energy gain by the working fluid to the gain that would result if the collector's absorbing surface had been at the same local fluid temperature [1]. The presence of the F' factor in the Hottel–Whillier–Bliss expression of the collector efficiency accounts for this effect [1], i.e.

$$\eta = F'\eta_0 - F'U_L(\bar{T}_f - T_{amb}). \qquad (1)$$

The factors affecting F' are considered in this study. In particular the case of a finned absorber, where the effect of this factor becomes of increasing importance, has been analysed.

2. ANALYSIS

For most geometries the collector efficiency factor, F', can be expressed as the ratio of two thermal resistances [1], namely

$$F' = \frac{1/U_L A_r}{1/U_0 A_r} \qquad (2)$$

where U_L and U_0 are the heat-transfer coefficients from the absorber and the working fluid, respectively, to the ambient environment. The present analysis is developed for absorber geometries for concentrating solar-energy collectors, although the results are applicable equally to non-concentrating collectors.

The half cross-section of a finned absorber is shown in Fig. 1. The particular geometry illustrated has been proposed [2] as a more efficient alternative absorber configuration for a CPC collector than a tubular absorber (i.e. a tube of diameter D in Fig. 1). With respect to its thermal resistance, the particular shape of the fin shown in Fig. 1 is irrelevant, its length w_f and thickness δ_f being the only parameters to be considered. Thus, the problem is equivalent to that of a flat fin of the same length and thickness (this is depicted by dashed lines in Fig. 1).

Duffie and Beckman [1] provided analytical expressions for F' of a finned absorber which have the following inherent simplifying assumptions:

(i) the variation of temperature in the axial direction along the fluid flow path was neglected and an average temperature was taken;

(ii) temperature in the radial direction of the tube wall was constant.

Wijeysundera [3] studied the problem without making the first assumption for various absorber geometries; in all cases the resulting value of F' was up to 4% greater than that given by Duffie and Beckman [1]. In the case of a finned tube, the assumption of a constant tube temperature is not realistic and will result in more favourable (i.e. higher) values for the F' factor. This has been quantified in the present study through a more detailed examination of the heat transfer in the cross-section shown in Fig. 1. The axial variation of temperature has not been considered, so the first assumption still holds.

The collector efficiency factor is analysed in the

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NOMENCLATURE

- A_r absorber area of the collector [m^2]
- C ratio of the aperture area to the absorber area in a concentrating collector
- D, D_i exterior and interior diameters of a tubular absorber, respectively (Fig. 1) [m]
- d, d_i exterior and internal diameters of a tube in a finned absorber, respectively (Fig. 1) [m]
- d_{in} interior diameter of the fluid layer (Fig. 4) [m]
- F fin efficiency (equation (9))
- F', F'_{cor} collector efficiency factors for a uniform and a non-uniform flux distribution, respectively
- f corrective factor for thermal resistances
- h, h' heat-transfer coefficients under a uniform and a non-uniform flux distribution, respectively [W m^{-2} K^{-1}]
- k thermal conductivity [W m^{-1} K^{-1}]
- n integer
- Q steady-state rate of heat transfer [W]
- $\dot{q}$ steady-state heat flux [W m^{-2}]
- $\dot{q}_L$ steady-state rate of local heat losses (Fig. 2) [W m^{-2}]
- R, R' thermal resistances under a uniform and non-uniform flux, respectively [K W^{-1}]
- S area [m^2]
- s infinitesimal increment of area [m^2]
- T temperature [K]
- $\Delta T(\phi)$ local temperature difference between a point on the external tube surface at an angular position ϕ and the mean fluid temperature [K]
- U_L, U_0 heat-loss coefficients between the absorber surface and the environment, and the working fluid and the environment, respectively [W m^{-2} K^{-1}]
- w width [m].

Greek symbols

- β non-dimensional coefficient (equation (26))
- δ thickness [m]
- η, η_0 instantaneous and optical efficiencies of the collector, respectively
- ϕ angle (Fig. 3) [deg]
- $\phi_{0,1}$, $\phi_{0,2}$ angles defining the arc over which the flux is delivered to the tube (Fig. 3) [deg].

Subscripts

- amb ambient environment
- b bond
- cor corrected
- equ equivalent
- f fluid
- fin fin
- i interior
- min minimum
- t tube
- u infinitesimal element
- 0, 1, 2 integers used as subscripts.

Superscript

- mean values.

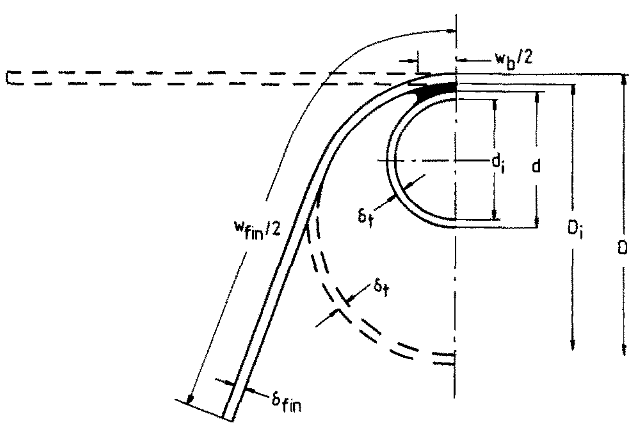

FIG. 1. Cross-section of a finned absorber suitable for a CPC collector. The equivalent (i.e. of the same absorbing surface) tubular absorber is also shown as dashed lines.

following sections for two absorber configurations: (i) a tubular absorber; (ii) a finned absorber.

2.1. *Tubular absorber*

For this case, a uniform flux distribution along the absorber circumference is assumed, F' can be expressed analytically as

$$F' = \frac{R_L}{R_L + R_t + R_f} \tag{3}$$

where

$$R_L = 1/U_L C A_r \tag{4}$$

$$R_t = \frac{D \ln(D/D_i)}{2k_t A_r} \tag{5}$$

$$R_f = \frac{D}{D_i h_f A_r}. \tag{6}$$

The thermal resistances R_L, R_t and R_f are based on the absorber area A_r of the collector. However, the overall heat-loss coefficient U_L is normally based on the aperture area, then the concentration ratio, C, of the collector appears in equation (4): $C = 1$ for flat-plate collectors.

2.2. *Finned absorber*

For the geometry shown in Fig. 1 and for a uniform flux distribution across the top surface of the fin, the efficiency factor of the collector can be expressed as

$$F' = \frac{R_L}{R_L + R_{fin} + f_b R_b + f_t(R_t + R_f)}. \tag{7}$$

The various thermal resistances appearing in equation (7) are all based on the absorber area, A_r, of the collector and are considered below.

The thermal resistance R_L between the absorber fin and the environment is given by equation (4). The thermal resistance of the free length of the fin is given by

$$R_{fin} = \frac{(w_{fin} - w_b)(1-F)}{w_b + (w_{fin} - w_b)F}(1/U_L C A_r) \tag{8}$$

where the fin efficiency for a constant thickness fin is given [1] by

$$F = \frac{\tanh[(U_L/k_f\delta_{fin})^{1/2}(w_{fin} - w_b)/2]}{(U_L/k_f\delta_{fin})^{1/2}(w_{fin} - w_b)/2}. \tag{9}$$

The terms R_b, R_t and R_f in equation (7) correspond to the thermal resistances of the bond, the tube and the working fluid, respectively, calculated under the assumption of a uniform-temperature distribution (i.e. the temperatures at the interfaces of fin–bond, bond–tube and tube–working fluid are constant). These terms can be expressed analytically as

$$R_b = \frac{w_{fin}\delta_b}{w_b k_b A_r} \tag{10}$$

$$R_t = \frac{w_{fin} \ln(d/d_i)}{2\pi k_t A_r} \tag{11}$$

$$R_f = \frac{w_{fin}}{\pi d_i h_f A_r}. \tag{12}$$

The non-uniform temperature distribution in the considered problem is taken into account by the adoption of two corrective factors, f_b and f_t, applied to the values provided by equations (10)–(12). Because in most practical cases the term R_t is negligible, the thermal resistances R_t and R_f can be considered together, and so a single corrective factor, f_t, has been employed.

When $f_b = f_t = 1$, equation (7) corresponds to that given by Duffie and Beckman [1], although the term R_t was omitted by Duffie and Beckman as negligible. Evaluating the corrective factors f_b and f_t allows the overall effect of the assumption for a uniform temperature distribution on F' to be quantified.

3. THERMAL RESISTANCES FOR NON-UNIFORM HEAT FLUXES

Consider the heat transfer for the general case shown in Fig. 2: the heat flux $\dot{Q}_1$, is delivered to the surface S_1 of a body of an arbitrary geometry and part of this flux $\dot{Q}_2$ is carried away from the surface S_2. The distributions of the heat fluxes on both surfaces S_1 and S_2 are arbitrary and heat transfer between these two surfaces may occur by conduction, convection and/or radiation. Heat is also dissipated over the rest of the external surface of the body at a local rate, $\dot{q}_L$. No internal heat generation occurs.

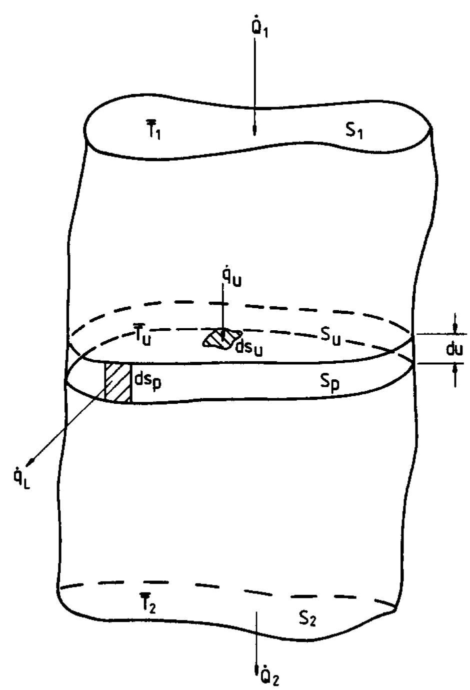

FIG. 2. Schematic diagram of a general mode of heat transfer within a body of arbitrary geometry.

Under steady-state conditions and for a segment of infinitesimal thickness du (Fig. 2), the heat flux $\dot{Q}_u$ entering this segment is

$$\dot{Q}_u = \int_{S_u} \dot{q}_u \, \mathrm{d}s_u. \tag{13}$$

The heat flux leaving this segment is

$$\dot{Q}_{u+\mathrm{d}u} = \dot{Q}_u - \int_{S_p} \dot{q}_\mathrm{L} \, \mathrm{d}s_p. \tag{14}$$

The thermal resistance of the considered element is defined as

$$\mathrm{d}R_u = \frac{\mathrm{d}\bar{T}_u}{(\dot{Q}_u + \dot{Q}_{u+\mathrm{d}u})/2} \tag{15}$$

where $\mathrm{d}\bar{T}_u = \bar{T}_u - \bar{T}_{u+\mathrm{d}u}$ is the difference between the mean temperatures of the upper and lower surfaces of the considered element. The total thermal resistance between surfaces S_1 and S_2 can be expressed as

$$R'_{12} = \int_{u_1}^{u_2} \frac{\mathrm{d}\bar{T}_u}{\dot{Q}_u - \frac{1}{2}\int_{S_p} \dot{q}_\mathrm{L} \, \mathrm{d}s_p} \tag{16}$$

where the boundary conditions applied are

$$\left.\begin{aligned} \dot{Q}_{u=u_1} &= \dot{Q}_1 \\ \dot{Q}_{u=u_2} &= \dot{Q}_2. \end{aligned}\right\} \tag{17}$$

Equation (16) provides the thermal resistance for the generalized case of heat transfers within one or more media. However, its application in a practical case involves considerable difficulties in calculation. If the side surface of the body is assumed to be perfectly insulated (i.e. $\dot{q}_\mathrm{L} = 0$, so $\dot{Q}_1 = \dot{Q}_2 = \dot{Q}$), then

$$R'_{12} = (\bar{T}_1 - \bar{T}_2)/\dot{Q}. \tag{18}$$

In the case of $\dot{q}_\mathrm{L} \neq 0$, equation (18) will give an overestimation of the thermal resistance involved if $\dot{Q} = \dot{Q}_2$ and an underestimation for $\dot{Q} = \dot{Q}_1$. If $\dot{q}_\mathrm{L}$ is varying between S_1 and S_2 such that a mean heat flux $\dot{Q} = (\dot{Q}_1 + \dot{Q}_2)/2$ ensues between $\dot{Q}_1$ and $\dot{Q}_2$, then by employing this latter value a realistic estimate for R'_{12} close to that provided by equation (16) can be obtained easily. This value can be compared with the thermal resistance R_{12} obtained by assuming a uniform flux distribution. The value of R_{12} can be derived analytically for simple geometries, e.g. for uniform heat conduction through a slab of thickness δ when $S_1 = S_2 = S$, it is $R_{12} = \delta/kS$. A corrective factor f is then defined as

$$f = R'_{12}/R_{12}. \tag{19}$$

The thermal resistance R_{12} is related to the overall heat-transfer coefficient h_{12} between surfaces S_1 and S_2 by

$$R_{12} = 1/h_{12}S \tag{20}$$

where $S = S_1$ or S_2 is the surface on which h_{12} is based. The corrective factor f given by equation (19) can be used consequently to derive the heat-transfer coefficient h'_{12} encountered under a non-uniform flux distribution as

$$h'_{12} = h_{12}/f. \tag{21}$$

Supposing that the heat-transfer coefficient h_{12} is known for a uniform flux distribution, equation (21) can then be used to provide the value of this coefficient under a non-uniform flux distribution.

4. THE ACTUAL THERMAL RESISTANCE OF THE BOND AND THE FLUID

The corrective factors f_b and f_t express the ratio of the thermal resistance under the actual prevailing flux distribution to the thermal resistance under a uniform flux distribution. These resistances are calculated according to equations (18) and (10)–(12), respectively.

4.1. *The corrective factor f_t*

According to equation (18), the thermal resistance of the tube–fluid subsystem (Fig. 1) is

$$R'_\mathrm{tf} = (\bar{T}_\mathrm{b\text{-}t} - \bar{T}_\mathrm{f})/\dot{Q} \tag{22}$$

where $\bar{T}_\mathrm{b\text{-}t}$ is the mean temperature at the contact arc of the bond–tube, $\bar{T}_\mathrm{f}$ the mean fluid temperature and $\dot{Q}$ the heat flux transferred. The corrective factor f_t is then expressed as

$$f_\mathrm{t} = \frac{R'_\mathrm{tf}}{R_\mathrm{t} + R_\mathrm{f}}. \tag{23}$$

Williams [4] gave for this case an analytical solution for the temperature distribution on the external surface of the tube by

$$\Delta T(\phi) = -\frac{2}{h_\mathrm{tf} d} \mathrm{d}\dot{q}/\mathrm{d}\phi \tag{24}$$

where

$$1/h_\mathrm{tf} = 1/h_\mathrm{f} + \delta_\mathrm{t}/2k_\mathrm{t} \tag{24a}$$

$\Delta T(\phi)$ is the local temperature difference between a point of the external tube surface at an angular position ϕ and the mean fluid temperature (Fig. 3), h_tf the heat-transfer coefficient from the exterior surface of the tube to the working fluid, and

$$\mathrm{d}\dot{q}/\mathrm{d}\phi = 0.5\beta \int_{\phi_{0,1}}^{\phi_{0,2}} \left(\frac{\mathrm{d}q_0}{\mathrm{d}\phi_0}\right) \times \sum_{n=-\infty}^{\infty} \exp\left(-\beta|\phi + 2n\pi - \phi_0|\right) \mathrm{d}\phi_0. \tag{25}$$

$\mathrm{d}\dot{q}/\mathrm{d}\phi$ is the intensity of the flux at an angular position ϕ, with

$$\beta = (h_\mathrm{f} d^2/4k_\mathrm{t}\delta_\mathrm{t})^{1/2}. \tag{26}$$

Equation (24) can be solved by numerical integration to provide the mean temperature $\bar{T}_\mathrm{b\text{-}t}$, by averaging the values it yields within the range of $\phi_{0,1} < \phi < \phi_{0,2}$. However, the actual distribution

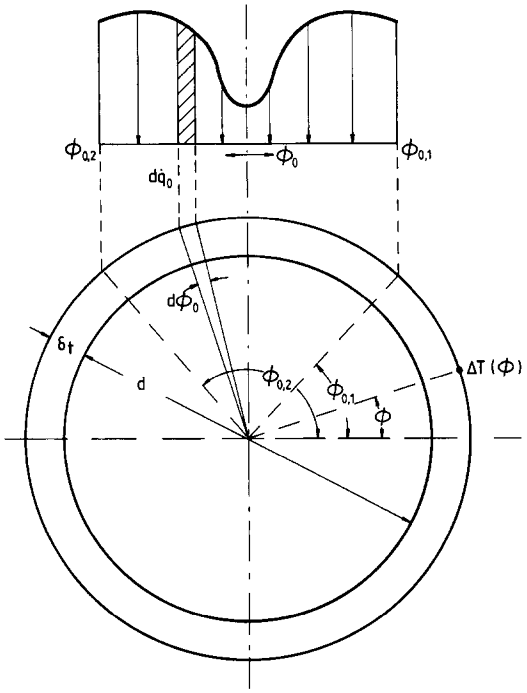

FIG. 3. Cross-section of a tube exposed to a non-uniform flux over the range $\phi_{0,1} < \phi < \phi_{0,2}$. The non-uniform flux distribution prevailing for $w_b/d = 0.75$ is also shown.

$(d\dot{q}_0/d\phi_0)$ of the flux at the bond–tube interface (equation (25)) is not known at this stage. This will be derived in the following section by considering the fin–bond subsystem.

4.2. *The corrective factor f_b*

The subsystem under consideration is shown in Fig. 4. The thermal resistance of the free length of the fin AA′ has been already accounted for by equation (8). In the considered subsystem $AB_1C_1C_2$, the heat flux enters through both surfaces AB_2 and AB_1 and is delivered to surface C_1C_2 (Fig. 4). For the purpose of evaluating this thermal resistance, it is assumed that no thermal losses occur from the fin and the external surfaces B_2C_2 and C_2D of the bond and the tube, respectively (i.e. the entire collected energy is delivered to the working fluid).

Two distinct thermal resistances occur in the considered heat-transfer case, as the subsystem exhibits: (i) a thermal resistance R_1 for the heat flux entering from surface AB_1; (ii) a thermal resistance R_2 for the heat flux entering from surface AB_2 (Fig. 4). The overall thermal resistance derives by proportional summation of R_1 and R_2 as

$$R_b' = (w_b/w_{fin})R_1 + (1 - w_b/w_{fin})R_2 \qquad (27)$$

where the fractions (w_b/w_{fiin}) and $(1 - w_b/w_{fin})$ correspond to the fractions of the heat flux entering from surfaces AB_1 and AB_2, respectively. The corrective factor f_b is then given as

$$f_b = R_b'/R_b. \qquad (28)$$

The evaluation of thermal resistances R_1 and R_2 from equation (18) involves the determination of the temperature distribution over (i) surface AB_1 for a uniform flux distribution over this surface only; (ii) surface AB_2 for a uniform flux distribution over fin surface AA′ only (Fig. 4).

A numerical solution was obtained by using a finite-element computer package [5]. As this could only deal with heat conduction, an 'equivalent' heat conduction problem was devised. In particular, the convection of heat from the tube to the working fluid with a heat-transfer coefficient h_f was replaced by conduction through a solid layer of thickness $(d - d_{in})/2$ (Fig. 4) with an equivalent conductivity of

$$k_{equ} = h_f(d/2)\ln(d/d_{in}). \qquad (29)$$

Although this transformed problem does not represent accurately the heat transfer behaviour inside the tube, it is only with the fin–bond subsystem that we are concerned at this stage, for which this transformation is valid. A value of $d/d_{in} = 8$ has been employed for this transformation.

5. DEDUCTIONS

Results derived from a combined numerical solution of the fin–bond and tube–working fluid subsystems are presented in this section. The values used for various parameters are tabulated in Table 1.

The corrective factor f_b, shown in Fig. 5 for a range of bond widths, exhibits considerably higher values

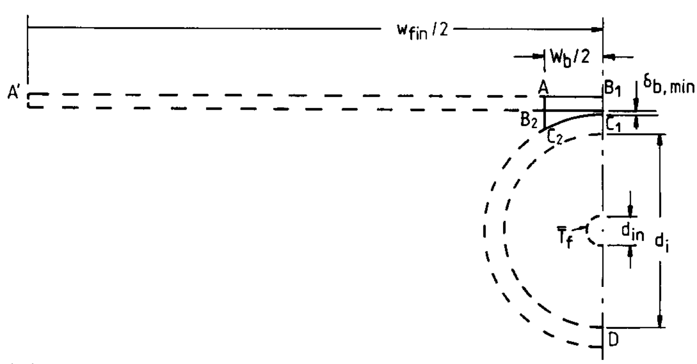

FIG. 4. The bond element considered, consisting of the non-free length of the fin and the bond (solid lines).

Table 1. Values used for various collector parameters

Parameter	Symbol	Value	Unit
Concentration ratio	C	1.55	—
Overall heat-loss coefficient	U_L	5	$W m^{-2} K^{-1}$
Internal diameter of tubular absorber	D_i	25	mm
Internal diameter of finned absorber	d_i	12	mm
Tube thickness in both absorbers	δ_t	1.5	mm
Thermal conductivity of tube	k_t	385†	$W m^{-1} K^{-1}$
Thermal conductivity of fin	k_{fin}	385	$W m^{-1} K^{-1}$
Fin length	w_{fin}	88	mm
Fin thickness	δ_{fin}	1	mm
Base thickness of bond	$\delta_{b,min}$	0.5	mm
Bond width	w_b	0.75d†	mm
Thermal conductivity of bond	k_b	80	$W m^{-1} K^{-1}$
Heat-transfer coefficient for the inner surface of the tube	h_f	variable	$W m^{-2} K^{-1}$
Collected heat flux by the absorber	$\dot{Q}$	745‡	$W m^{-2}$

† Unless otherwise explicitly stated in the text or on the figures.
‡ This value is based on the aperture area of the collector and incorporates optical losses.

than unity. The dependence of the f_b factor on the value of the heat-transfer coefficient inside the tube was found to be very weak, so only a single curve is shown.

The flux distribution over the arc of the tube in contact with the bond was also deduced for all the cases considered and was used subsequently for the numerical integration of equation (25). A typical flux distribution for $w_b/d = 0.75$ is shown in Fig. 3. The profiles of the temperature distribution over the exterior surface of the tube can be seen in Fig. 6 to exhibit a peak at the region corresponding to the contact arc between the tube and the bond. The magnitude of the encountered temperature drop (i.e. local tube temperature minus average fluid temperature) is shown to depend strongly on the heat-transfer coefficient inside the tube.

The variations of the corrective factor f_t, depicted in Fig. 7 for some representative cases, are shown to depend considerably on the conductivity of the tube material. Although f_t is generally smaller than f_b (Figs. 5 and 7), the effect of f_t on the collector efficiency factor is greater, as f_b is associated with a thermal resistance of small magnitude. This is illustrated in Table 2, where the magnitudes of the various thermal resistances appearing in equation (7) are shown for some representative cases. Also shown in Table 2 are the values of the collector efficiency factors F' derived without any correction (i.e. $f_b = f_t = 1$) and F'_{cor}, derived for the actual values of f_b and f_t.

A more comprehensive comparison between F' and F'_{cor} can be seen in Fig. 8: the values of F'_{cor} are invariably lower than those of F'. For the particular values of the parameters used (Table 1), the range of

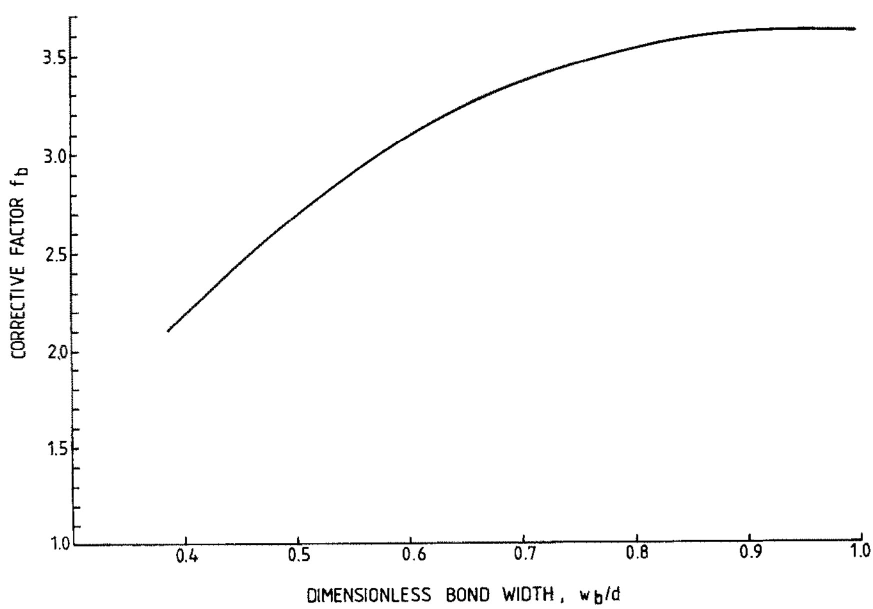

FIG. 5. The corrective factor f_b for various bond widths.

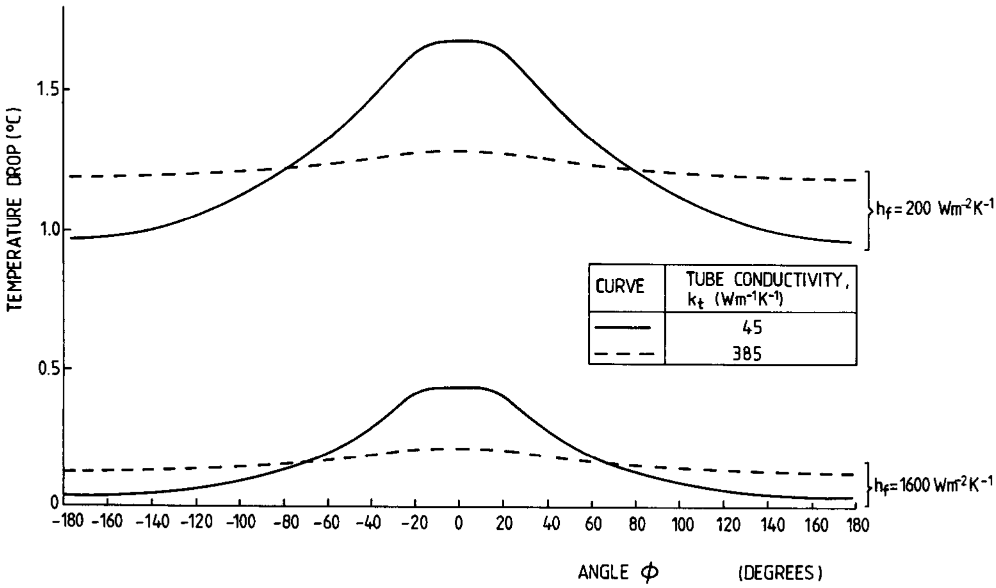

FIG. 6. Temperature distributions over the exterior surface of the tube, shown for two values of the tube conductivity and two values of the heat-transfer coefficient inside the tube.

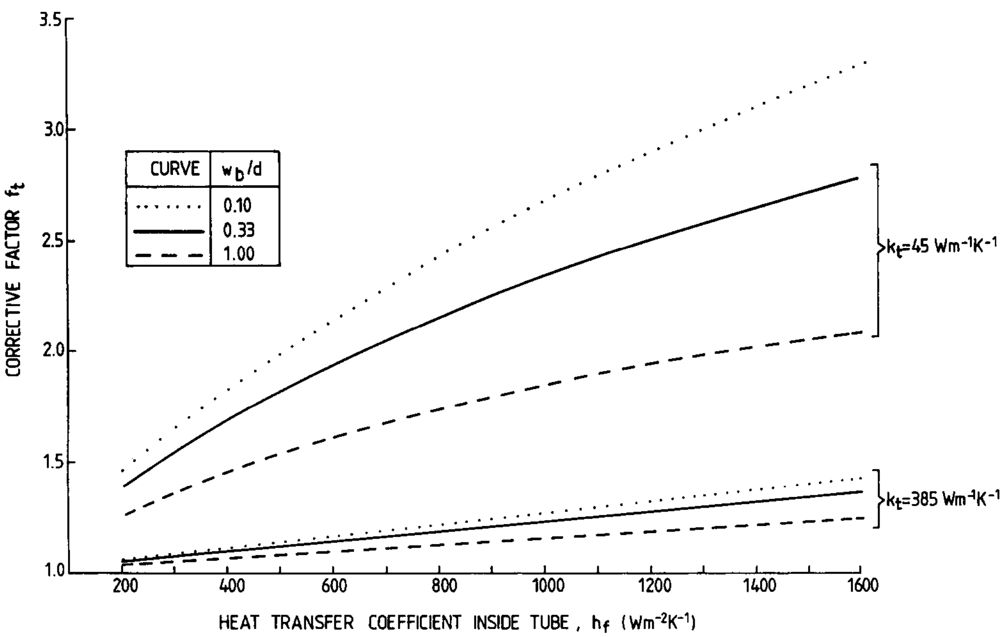

FIG. 7. The corrective factor f_t shown for two values of the tube conductivity and three values of the bond width.

Table 2. Magnitudes of the thermal resistances of the fin–bond and the tube–working fluid subsystems. These results were derived for values of the relevant parameters appearing in Table 1

k_t ($W m^{-1} K^{-1}$)	h_f ($W m^{-2} K^{-1}$)	$f_b R_b$ ($\times 10^{-3} K W^{-1}$)	$f_t(R_t + R_f)$ ($\times 10^{-3} K W^{-1}$)	F'	F'_{cor}
	200	0.51	12.54	0.9068	0.901
385	900	0.51	3.19	0.9701	0.964
	1600	0.51	1.95	0.9786	0.972
	200	0.51	16.21	0.9064	0.878
45	900	0.51	5.58	0.9696	0.947
	1600	0.51	3.91	0.9781	0.958

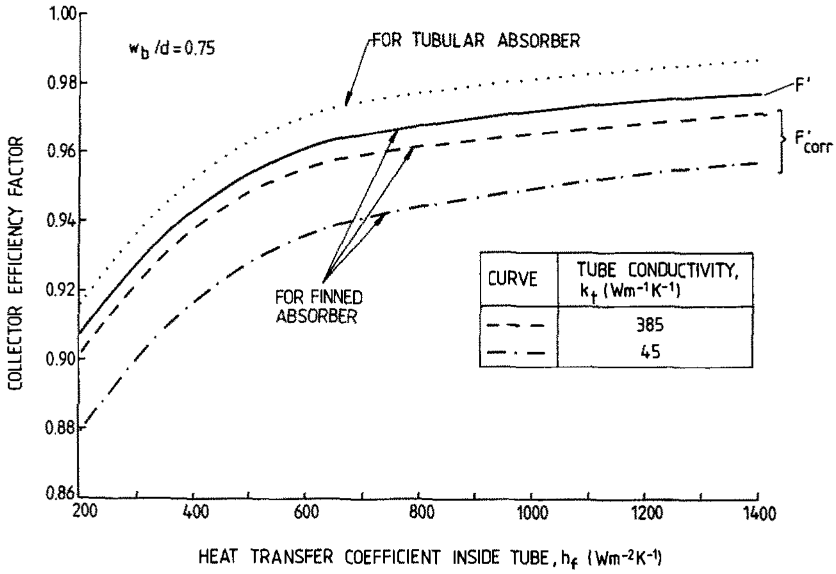

FIG. 8. Collector efficiency factors F' and F'_{cor}, respectively, for a finned absorber as derived (i) with the assumption of uniform flux distribution, (ii) by taking into account the non-uniformity of the flux distribution. The curve corresponding to a tubular absorber is also shown by means of dotted lines.

overestimation resulting without applying a correction lies between 0.5 and 3%. Also shown in Fig. 8 is the collector efficiency factor for a tubular absorber, as derived from equation (3) for the same values of the relevant parameters that appear in Table 1. Although a direct comparison of the F' values is not recommended (the heat-transfer coefficients in the working fluid are not necessarily the same under actual operating conditions), it can be seen, however, that a particular F'_{cor} value for the tubular absorber is higher than the F' value for the finned absorber throughout the range of h_f values depicted.

6. DISCUSSION AND CONCLUSIONS

An analysis of the collector efficiency factors for tubular and finned-tube absorbers in solar-energy collectors has been presented. This was based on an examination of the thermal resistances intervening in the path of the heat flow. The employed method can be applied generally under non-uniform flux distributions.

The results derived demonstrate that, under non-uniform flux distribution, the thermal resistances involved in a heat-transfer problem are generally higher than those corresponding to a uniform flux distribution. In particular, for a solar-energy collector with a finned absorber, the calculation of the collector efficiency factor F' by ignoring the non-uniform flux distribution at the various elements of the absorber yields an *overestimation* of its value by a percentage ranging from 0.5 to 3%. However, an *underestimation* of the F' value occurs [3], by a percentage not exceeding 4%, when the axial, with respect to the working fluid flow, variation of temperature is not taken into account.

Thus, for many finned absorber designs, the counteracting effects in the axial and radial directions cancel each other out. In such cases, equation (7) with $f_b = f_t = 1$ will provide a realistic prediction of the efficiency factor of the collector. However, the validity of such a simplified approach should be verified using the presented analysis for generic absorber designs. In addition, this analysis provides a more comprehensive insight into the overall heat transfers that occur in the absorber of a solar-energy collector.

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RESISTANCE THERMIQUE D'UN COLLECTEUR D'ENERGIE SOLAIRE SOUS UNE DISTRIBUTION DE FLUX NON UNIFORME

Résumé—On traite d'une méthode généralisée pour déterminer des résistances thermiques de milieux subissant des distributions de flux non uniformes. Elle est appliquée à l'analyse des transferts thermiques dans un tube à ailettes utilisé comme collecteur d'énergie solaire. On montre que, pour une distribution de flux non uniforme, les valeurs des résistances thermiques sont supérieures à celles prédites dans le cas d'un flux uniforme. Néanmoins pour un absorbeur à tube aileté ceci est compensé (soit partiellement, soit dépassé suivant les caractéristiques particulières du collecteur) par un effet dû à la variation axiale de la température.

WÄRMELEITWIDERSTAND DES ABSORBERS IN EINEM SONNENKOLLEKTOR BEI UNGLEICHFÖRMIGER WÄRMESTROMVERTEILUNG

Zusammenfassung—Es wurde ein allgemein gültiges Verfahren zur Bestimmung des Wärmeleitwiderstandes bei ungleichförmiger Wärmestromverteilung entwickelt. Das Verfahren wurde auf die Untersuchung der Wärmeleitung in einem berippten Rohr angewandt, das als Absorber in einem Sonnenkollektor installiert ist. Die Untersuchung zeigt, daß der für gleichförmige Wärmestromverteilung berechnete Wärmeleitwiderstand grundsätzlich niedriger ist als der bei ungleichförmiger Wärmestromverteilung tatsächlich ermittelte. Im Absorber eines Sonnenkollektors wird dies jedoch kompensiert (entweder teilweise oder ganz, abhängig von den jeweiligen Eigenschaften des Kollektors) durch den entgegenwirkenden Einfluß der axialen Temperaturverteilung in dem berippten Absorber-Rohr.

ТЕПЛОВОЕ СОПРОТИВЛЕНИЕ АБСОРБЕРА В КОЛЛЕКТОРЕ СОЛНЕЧНОЙ ЭНЕРГИИ ПРИ НЕРАВНОМЕРНОМ РАСПРЕДЕЛЕНИИ ПЛОТНОСТИ ТЕПЛОВОГО ПОТОКА

Аннотация—Разработан обобщённый метод определения теплового сопротивления сред при неравномерном распределении плотности теплового потока. Метод применён при анализе теплопереноса в оребрённой трубе, используемой в качестве абсорбера в коллекторе солнечной энергии. Показано, что при неравномерном распределении плотности теплового потока значения теплового сопротивления, полученные в предположении равномерности распределения, оказываются, как правило, ниже фактических значений. Однако, для абсорбера в форме оребрённой трубы это занижение значений компенсируется (частично или полностью, в зависимости от характеристик коллектора) противоположным влиянием распределительной функции коллектора, которое приводит к осевому выравниванию температуры.