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LOW MASS FLOW RATE IN FLAT-PLATE LIQUID HEATING SOLAR COLLECTORS

Heat transfer problems in flat-plate liquid heating solar collector have been analysed. Some improved collector efficiency criteria have been used. Low mass flow rate (LMFR) of working fluid has been shown as an advantageous operating range for solar collector. The serpentine-flow absorber of the particular type has been proposed as an improved construction for LMFR-collector.

NOMENCLATURE

A – collector area (=absorber area) [m^2]
B – fin length [m]
c_p – fluid specific heat at constant pressure [$J \cdot kg^{-1} \cdot K^{-1}$]
D – outer tube diameter [m]
d – inside tube diameter [m]
d_h – hydraulic diameter [m]
L – absorber length [m]
$\dot{m}_A$ – fluid mass flow rate per collector unit area [$kg \cdot s^{-1} \cdot m^{-2}$]
$\dot{m}_c$ – total collector fluid mass flow rate [$kg \cdot s^{-1}$]
$\dot{m}_t$ – fluid mass flow rate through the single tube [$kg \cdot s^{-1}$]
N – number of covers
q_a – absorbed solar energy per unit area [$W \cdot m^{-2}$]
q_s – irradiance on solar collector surface [$W \cdot m^{-2}$]
s – gap thickness (in parallel plate absorber) [m]
S – gap width (in parallel plate absorber) [m]
T – temperature [K]
T_a – ambient temperature [K]
T_{ab} – absorber mean temperature [K]
T_B – tube wall temperature [K]

T_f – fluid temperature [K]
T_{in} – inlet fluid temperature [K]
T_{out} – outlet fluid temperature [K]
T_s – effective temperature of the surface of the Sun [K]
ΔT – temperature rise (= $T_{out} - T_{in}$) [K]
u – fluid flow velocity [$m \cdot s^{-1}$]
U_b – collector back loss coefficient [$W \cdot m^{-2} \cdot K^{-1}$]
U_L – collector overall loss coefficient [$W \cdot m^{-2} \cdot K^{-1}$]
U_t – collector top loss coefficient [$W \cdot m^{-2} \cdot K^{-1}$]
W – distance between tubes [m]
x,y – rectagular coordinates [m]
α – absorber plate absorptance
α_f – heat transfer coefficient inside channel [$W \cdot m^{-2} \cdot K^{-1}$]
α_w – wind heat transfer coefficient [$W \cdot m^{-2} \cdot K^{-1}$]
β – collector tilt [°]
δ – absorber plate thickness [m]
δ_i – back insulation thickness [m]
ε – absorber plate emissivity
ε_g – cover emissivity
ε_L – correction factor for short channels
λ – thermal conductivity of absorber plate [$W \cdot m^{-1} \cdot K^{-1}$]
λ_i – thermal conductivity of back insulation [$W \cdot m^{-1} \cdot K^{-1}$]
η – collector thermal efficiency
η_b – collector exergetic efficiency
η_{Car} – Carnot cycle efficiency
η_{ef} – collector effectiveness
ρ – fluid mass density [$kg \cdot m^{-3}$]
σ – Stefan-Boltzmann constant [$W \cdot m^{-2} \cdot K^{-4}$]
τ – cover transmittance
$(\tau\alpha)_{ef}$ – effective transmittance-absorptance product
Gr – Grashof number
Nu – Nusselt number
Pr – Prandtl number
Re – Reynolds number

1. INTRODUCTION

In this paper the flat-plate liquid heating solar collector is considered. Collectors of this type have been in common use for many years and are the subject of numerous solar engineering handbooks [1,2].

In flat-plate liquid heating collectors water (or another working fluid) is usually heated up, for domestic uses, to temperatures in general not exceeding 100°C. The determination of the optimum operating range or the optimum operating strategy of solar collectors has been the subject of numerous studies [3,4,5,6,7,8,9,10,11].

In the present work, in order to simplify our considerations, the collector has been separated „in thought" from the other devices, entering into the composition of the solar energy system. Furthermore, the fluid has been assumed to pass through the collector only once (single-pass heating [3,11]).

The quantity commonly used as the measure of the performance of a solar collector is its thermal efficiency [1,2,12,13], which expresses the ratio of the useful collected energy to the available incident energy.

The conception of thermal efficiency is not sufficient for estimating the performance of collectors. A collector reaches the highest value of thermal efficiency when the mass flow rate becomes infinitely large. Then, however, the fluid outlet temperature approaches the fluid inlet temperature and the resulting thermal energy is practically useless.

In the present paper we have tried to find some quantities describing the work of a collector in a more comprehensive manner than by means of its thermal efficiency, and then to determine the general trends in the construction of flat-plate liquid heating solar collectors.

2. FORMULATION OF THE PROBLEM

The quantities describing a collector in a more comprehensive manner than by means of its thermal efficiency may assume different forms. However, it was not possible to include all of them in this paper, and so we decided to limit the analysis to two selected ones.

The first quantity under consideration can be derived by means of a Carnot cycle, which could use the thermal energy of the outlet fluid for the production of mechanical work (Fig.1). Then the efficiency of the whole system (collector + Carnot cycle) – and also a kind of collector „effectiveness" – assumes the following form[1)]

$$\eta_{ef} = \eta\ \eta_{Car} = \eta(1 - T_{in}/T_{out}) \tag{1}$$

where

$$\eta = (\dot{m}_c c_p \Delta T)/(A\ q_s) \tag{2}$$

[1)] In equation (1) instead of the inlet temperature (T_{in}) the ambient temperature (T_a) can be inserted — this question can be controversial.

Similar considerations are carried out for real systems [14,15], which use thermal energy (obtained in a solar device) for the production of electrical or mechanical energy (e.g. solar thermal power plants). However, the quantity η_{Car} was inserted in equation (1) to emphasize the importance of the quality of the thermal energy produced in the solar collector.

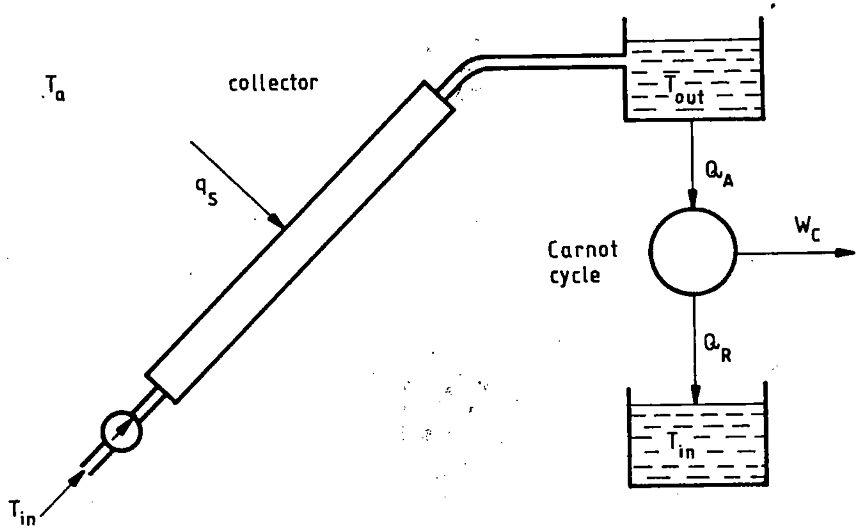

Fig.1. Solar collector with a Carnot cycle for illustrating collector effectiveness (Q_A, Q_R and W_C are the heat added, heat rejected and work connected with the Carnot cycle)

As the second of the above-mentioned quantities we took a quantity which is well-known in solar energy publications [12,16], namely the collector exergetic efficiency in the following form:

$$\eta_b = (c_p \dot{m}_c)\left[(T_{out} - T_{in}) - T_a \ln(T_{out}/T_{in})\right] / \left[(A\ q_s)(1 - T_a/T_s)\right] \tag{3}$$

Considering the effectiveness η_{ef} and the exergetic efficiency η_b of a collector and maximizing them with respect to the mass flow rate (or the fluid velocity), we can find its optimum operating range, and subsequently draw inferences concerning the general trends in collector construction.

The above process should be based on a comprehensive analysis of the problems of heat transfer in a solar collector. This may be difficult for two reasons: 1) there is no universal description of heat transfer (on account of a large number of solar collector constructions), 2) in the heat transfer analysis of selected solar collectors there are a great many variables.

In view of these facts we decided to restrict our investigations to two essential types of absorbers: a parallel tube absorber (Fig. 2) and a parallel plate absorber (Fig. 3). Moreover, several variables were fixed (e.g. $T_a, T_{in}, (\tau\alpha)_{ef}$, et al.), while some of the variables varied within fixed ranges (B, L, q_s, u).

In real systems the flow of the fluid through the collector loop is usually forced by means of a pump, which consumes an additional amount of electrical energy. For this reason the operation of the collector should be optimized with respect to the energy used by the pump (e.g. as has been done by Winn and Hull [5]). In the present work these questions have been omitted, that is we have neglected the electrical power consumed by the pump.

3. HEAT TRANSFER ANALYSIS

The model of a parallel tube absorber is a tube (channel) with symmetrically placed fins [1,2,17] (Fig. 2). Calculations for this type of absorber are usually carried out by means of an analysis proposed by Hottel and Whillier [1], where the heat is

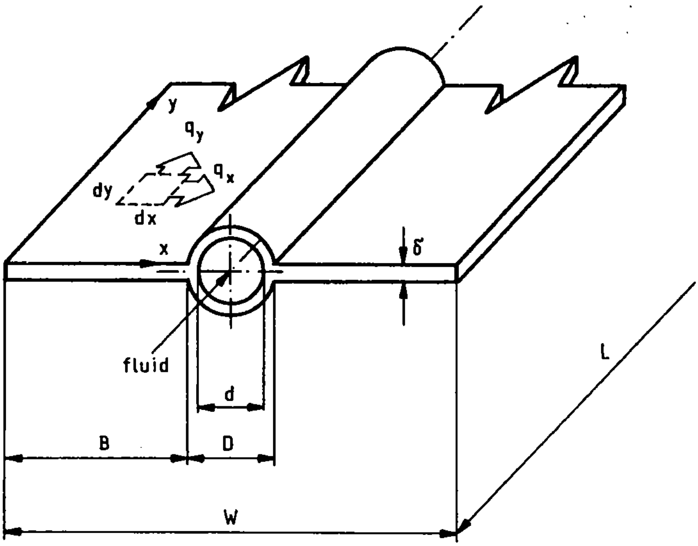

Fig.2. Elemental section of parallel tube absorber

assumed to be conducted only perpendicularly to the flow of the fluid. Investigations [18,19] show that the Hottel and Whillier relationships overestimate the values of the outlet fluid temperature and the thermal efficiency. This is why in the present paper heat conduction in the direction of the fluid flow is also taken into account. The heat conduction equation for fin $(0 \le x \le B,\ 0 \le y \le L)$ – Fig. 2 – can be written in the form

$$\nabla^2 T + \left[q_a - U_L\left(T - T_a\right)\right] / \delta\lambda = 0 \tag{4}$$

with the boundary conditions

$$\begin{array}{llll} \partial T/\partial x = 0 & & \text{for } x = 0 \text{ and } 0 \le y \le L \\ \partial T/\partial y = 0 & & \text{for } y = 0 \text{ and } 0 \le x \le B \\ & \text{and} & \text{for } y = L \text{ and } 0 \le x \le B \\ T(x,y) = T_B(y) & & \text{for } x = B \text{ and } 0 \le y \le L \end{array} \qquad (5)$$

The heat conduction equation for the tube wall (assuming that the wall is at a constant temperature in the whole transverse section) can be written in the form

$$F\lambda \frac{d^2 T_B}{dy^2} + D\left[q_a - U_L(T_B - T_a)\right] - \alpha_f \pi d(T_B - T_f) + 2\delta\lambda \left.\frac{\partial T}{\partial x}\right|_{x=B} = 0 \qquad (6)$$

(where $F = \pi(D^2 - d^2)/4$) with the boundary conditions

$$dT_B/dy = 0 \quad \text{for} \quad y = 0 \quad \text{and} \quad y = L \qquad (7)$$

The temperature of the fluid is determined by thermal balance for the fluid element. Thus

$$\dot{m}_t c_p (dT_f/dy) - \alpha_f \pi d(T_B - T_f) = 0 \qquad (8)$$

with the boundary condition

$$T_f(y = 0) = T_{in} \qquad (9)$$

For the parallel plate absorber (Fig. 3) analogous equations have the following forms:

$$\frac{d^2 T}{dy^2} + \left[q_a - U_L(T - T_a) - \alpha_f(T - T_f)\right]/\delta\lambda = 0 \qquad (10)$$

with the boundary conditions

$$dT/dy = 0 \quad \text{for} \quad y = 0 \quad \text{and} \quad y = L \qquad (11)$$

and

$$\dot{m}_c c_p (dT_f/dy) - \alpha_f S(T - T_f) = 0 \qquad (12)$$

with boundary condition (9).

For a parallel tube absorber the heat transfer coefficient (the arithmetic-mean one[1)]) inside the tube was calculated with the use of the Micheev correlation [20]

$$\mathrm{Nu}_f = 0{,}15\ \mathrm{Re}_f^{0{,}33}\,\mathrm{Pr}_f^{0{,}43}\,\mathrm{Gr}_f^{0{,}1}\,(\mathrm{Pr}_f/\mathrm{Pr}_w)^{0{,}25}\,\varepsilon_L \qquad (13)$$

(indices „f" and „w" relate to the temperatures of the fluid and the wall respectively).

1) For detailed analysis concerning different kinds of mean temperatures and resulting Nusselt numbers see Jacob [21].

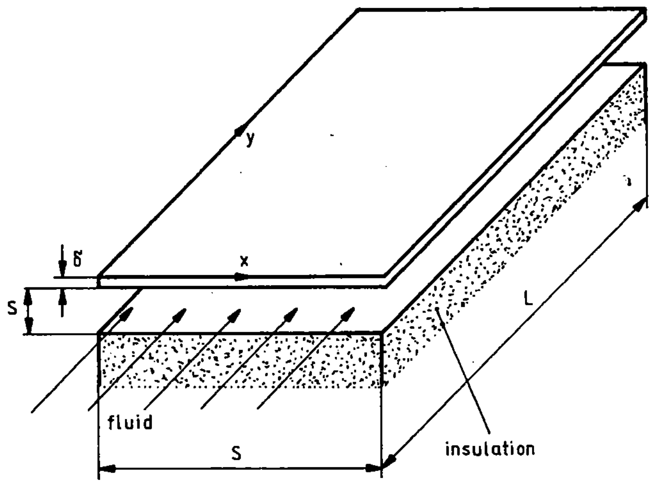

Fig. 3. Parallel plate absorber

For the parallel plate absorber the heat transfer coefficient inside the channel was calculated according to correlation [1][1)]

$$\overline{\text{Nu}} = 4{,}9 + \left[0{,}0606\left(\text{Re Pr } d_h/L\right)^{1{,}2}\right] \times \times \left[1 + 0{,}0909\left(\text{Re Pr } d_h/L\right)^{0{,}7} \text{Pr}^{0{,}17}\right]^{-1} \tag{14}$$

For both absorber types the collector overall loss coefficient was caculated without including edge losses, that is $U_L = U_t + U_b$ (where U_b was approximated as $U_b = \lambda_i/\delta_i$). For determining the top loss coefficient the following equation was used [1]:

$$U_t = \left[N/\left\{(C/T_{\text{ab}})\left[(T_{\text{ab}} - T_a)/(N+f)\right]^e\right\} + 1/\alpha_w\right] + \sigma(T_{\text{ab}} + T_a)\left(T_{\text{ab}}^2 + T_a^2\right) \times \times \left\{(\varepsilon + 0{,}00591\ N\ \alpha_w)^{-1} + \left[(2N + f - 1 + 0{,}133\ \varepsilon)/\varepsilon_g\right] - N\right\}^{-1} \tag{15}$$

where: $C = 520(1 - 0{,}000051\ \beta^2)$ for $0° \leq \beta \leq 70°$

(for $70° \leq \beta \leq 90°$, use $\beta = 70°$),

$e = 0{,}43\,(1 - 100/T_{\text{ab}})$,

$f = (1 + 0{,}089\ \alpha_w - 0{,}1166\ \alpha_w \varepsilon)(1 + 0{,}07866\ N)$.

[1)] See also Jacob's solution [21].

The effective transmittance-absorptance product was obtained from the relation [1]

$$(\tau\alpha)_{ef} = 1{,}02\ \tau\ \alpha \tag{16}$$

Besides the model of forced convection in the channel, for describing heat transfer in a parallel plate absorber use was also made of the model of unsteady-state heating of the infinite plate [22] – heat conduction in the fluid in the direction perpendicular to the surface of the absorber was assumed to play the decisive role.

The occurrence of free convection is a separate problem. The following two cases can be distinguished: 1) the collector operates without a pump (e.g. thermosyphon circulation; this kind of collector operation was not considered), 2) the fluid flow rate through the collector is controlled by means of the pump, whereas inside the absorber free convection overlaps forced convection (producing the so-called mixed convection). In a parallel tube absorber this phenomenon is taken into account in the Micheev formula (the Grashof number is raised to the power 0,1)[1)]. Considerations regarding mixed convection in a parallel plate absorber (carried out by the authors) show the necessity of experiments.

4. NUMERICAL CALCULATIONS

The following quantities were treated in the analysis as constants and given the values:

T_a	$= 20°C$,	α_w	$= W\cdot m^{-2}\cdot K^{-1}$,	T_{in}	$= 15°C$,
ρ	$= 1000\ kg\cdot m^{-3}$,	c_p	$= 4200\ J\cdot kg^{-1}\cdot K^{-1}$,	α	$= 0{,}95$,
ε	$= 0{,}95$,	λ	$= 150\ W\cdot m^{-1}\cdot K^{-1}$,	δ	$= 0{,}002$ m,
τ	$= 0{,}87$,	ε_g	$= 0{,}88$,	δ_i	$= 0{,}05$ m,
λ_i	$= 0{,}028\ W\cdot m^{-1}\cdot K^{-1}$,	β	$= 45°$,	N	$= 1$

for a parallel tube absorber: $D = 0{,}012$ m, $d = 0{,}010$ m,
for a parallel plate absorber: $s = 0{,}01$ m.

The quantities which varied in the course of analysis were the following:

- solar radiation intensity (q_s): 300; 600; 900 $W\cdot m^{-2}$;
- fin length (B): 0,00; 0,01; 0,04; 0,08; 0,14 m;
- absorber length (L): 0,5; 1,0; 1,5; 2,0 m;
- fluid flow velocity (u): usually 20÷30 values from the range $10^{-6}\div10^{-1}\ m\cdot s^{-1}$.

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1) This is why the Micheev correlation seems to be one of the best for the problems of this kind.

Calculations were made for more than 1000 different sets of data in all.

The values of the coefficients U_L and α_f were calculated iteratively. Each iteration required the numerical solution of the system of equations (4), (6), (8) with boundary conditions (5), (7), (9) or the system of equations (10), (12) with boundary conditions (11), (9) (by the finite difference method).

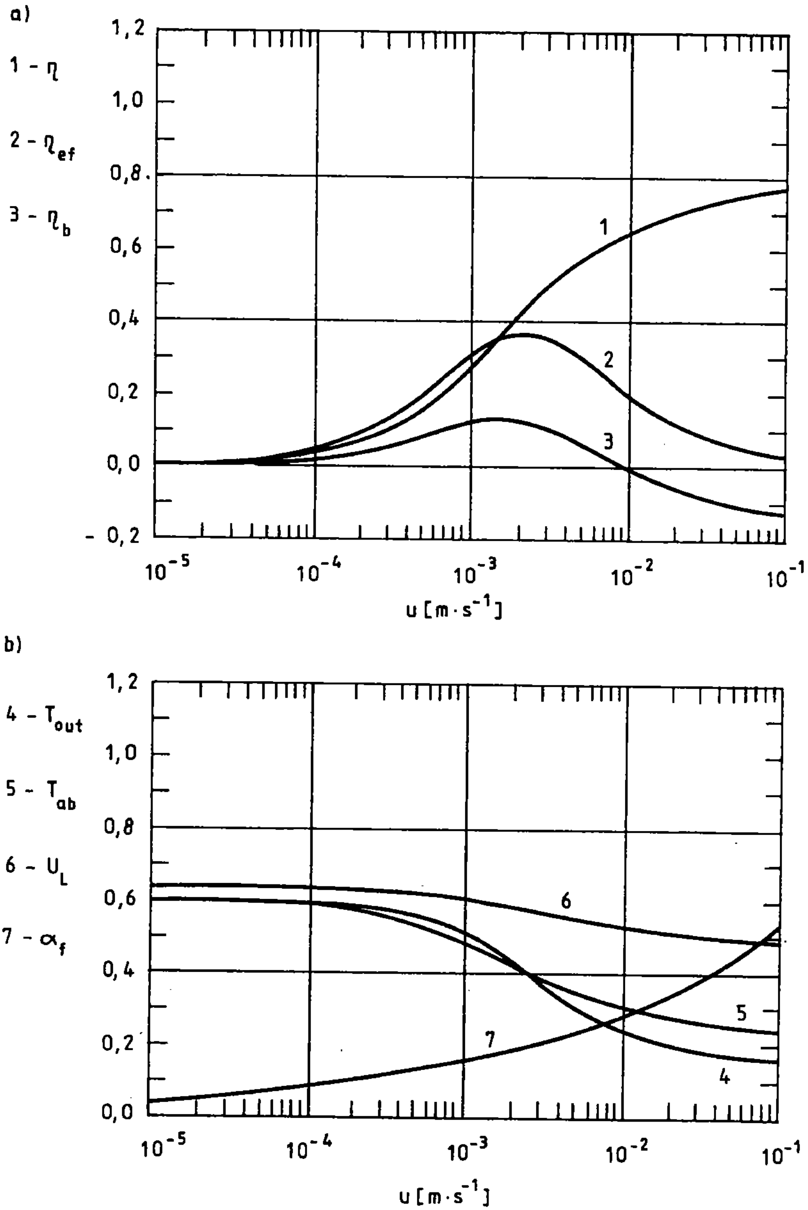

Fig.4. η [−], $\eta_{ef} \times 10$ [−], $\eta_b \times 10$ [−], $T_{out} \times 10^{-2}$ [°C] $T_{ab} \times 10^{-2}$ [°C] $U_L \times 10^{-1}$ [$W \cdot m^{-2} \cdot K^{-1}$], $\alpha_f \times 10^{-3}$ [$W \cdot m^{-2} \cdot K^{-1}$] vs fluid velocity u (for B = 0,14 m; L = 0,5 m; q_s = 300 $W \cdot m^{-2}$)

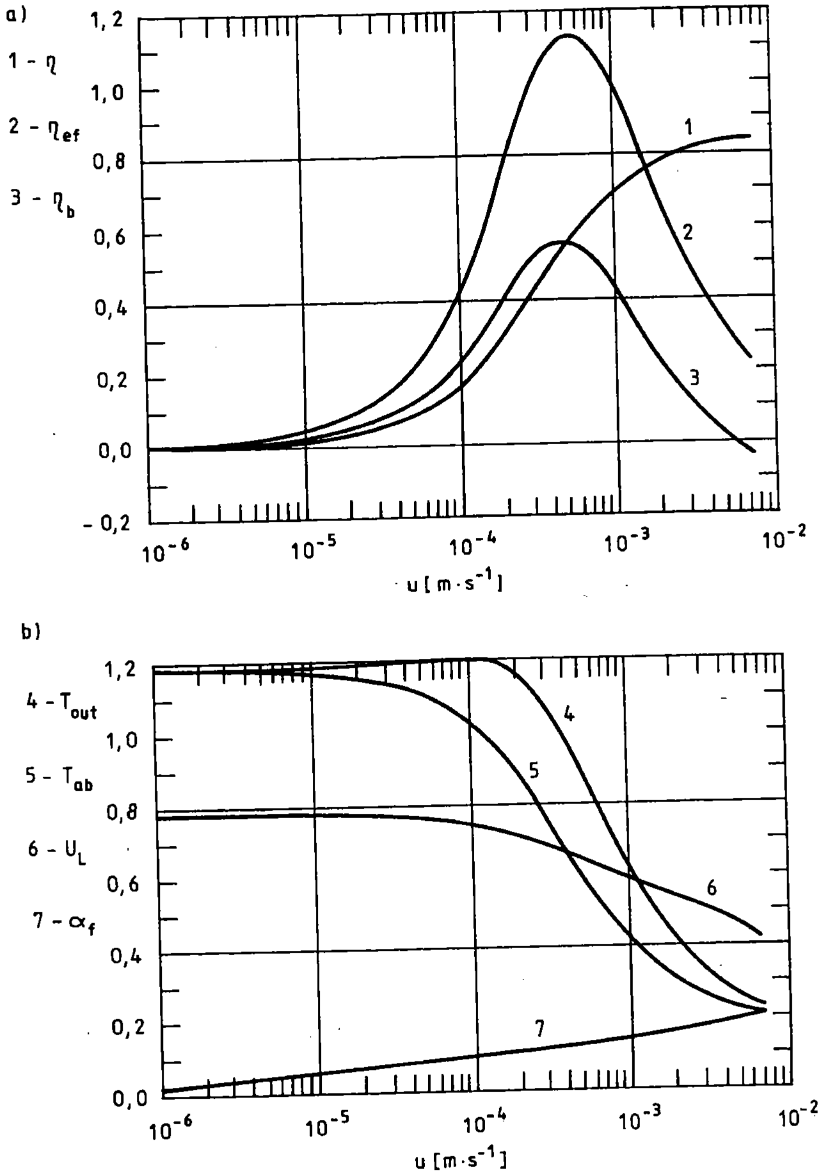

Fig.5. η [–], $\eta_{ef} \times 10$ [–], $\eta_b \times 10$ [–], $T_{out} \times 10^{-2}$[°C] $T_{ab} \times 10^{-2}$[°C] $\cdot U_L \times 10^{-1}$[W·m^{-2}·K^{-1}], $\alpha_f \times 10^{-3}$[W·m^{-2}·K^{-1}] vs fluid velocity u (for B = 0 m; L = 2,0 m; q_s = 900 W·m^{-2})

Some calculations were carried out, in spite of the fact that their technical or physical meaning may be questionable (on account of very low velocities – Fig. 4 and 5). However, such calculations were, in authors' opinion, necessary to demonstrate the variation of the quantities under consideration in the range as wide as possible. For this reason some relationships must have been to some extent extrapolated (e.g. Micheev correlation).

5. RESULTS

On the basis of numerical results it was possible to obtain the graphs of seven quantities involved in the operation of the collector (η, η_{ef}, η_b, T_{out}, T_{ab}, U_L, α_f) *vs* fluid velocity u (examples in Fig. 4 and Fig. 5).

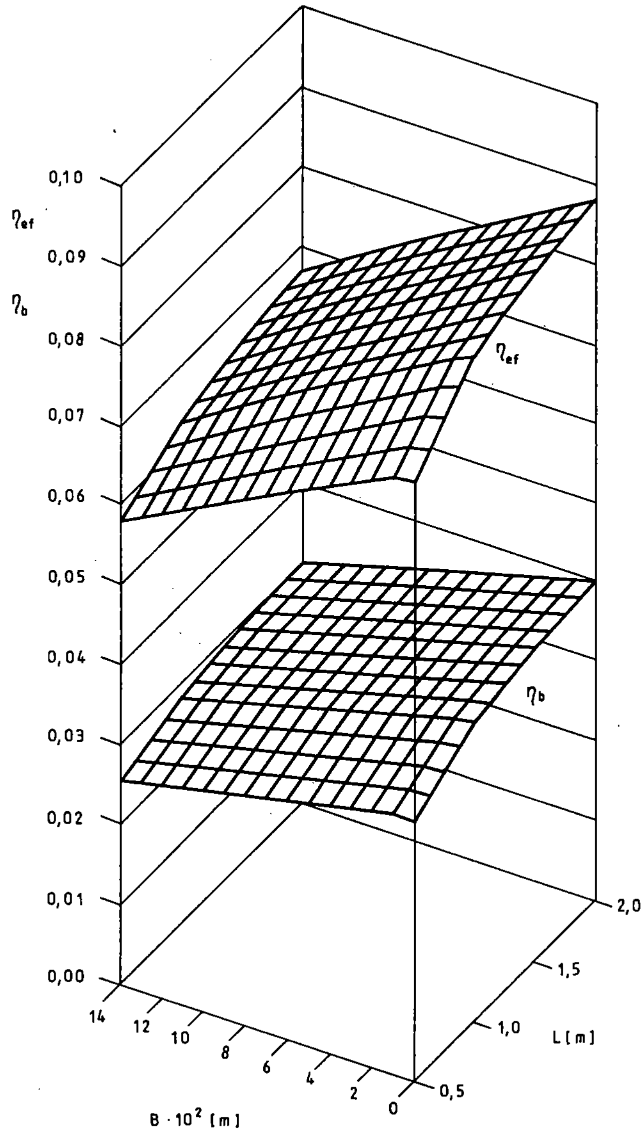

Fig.6. Maximum values of η_{ef} and η_b as functions of dimensions B and L for q_s = 600 W·m^{-2}

Fig. 6 shows the maximum values of the effectiveness η_{ef} and the exergetic efficiency η_b of the collector *vs* the dimensions B and L with q_s as the parameter (obtained from 20 graphs similar to these shown in Fig. 4 and 5).

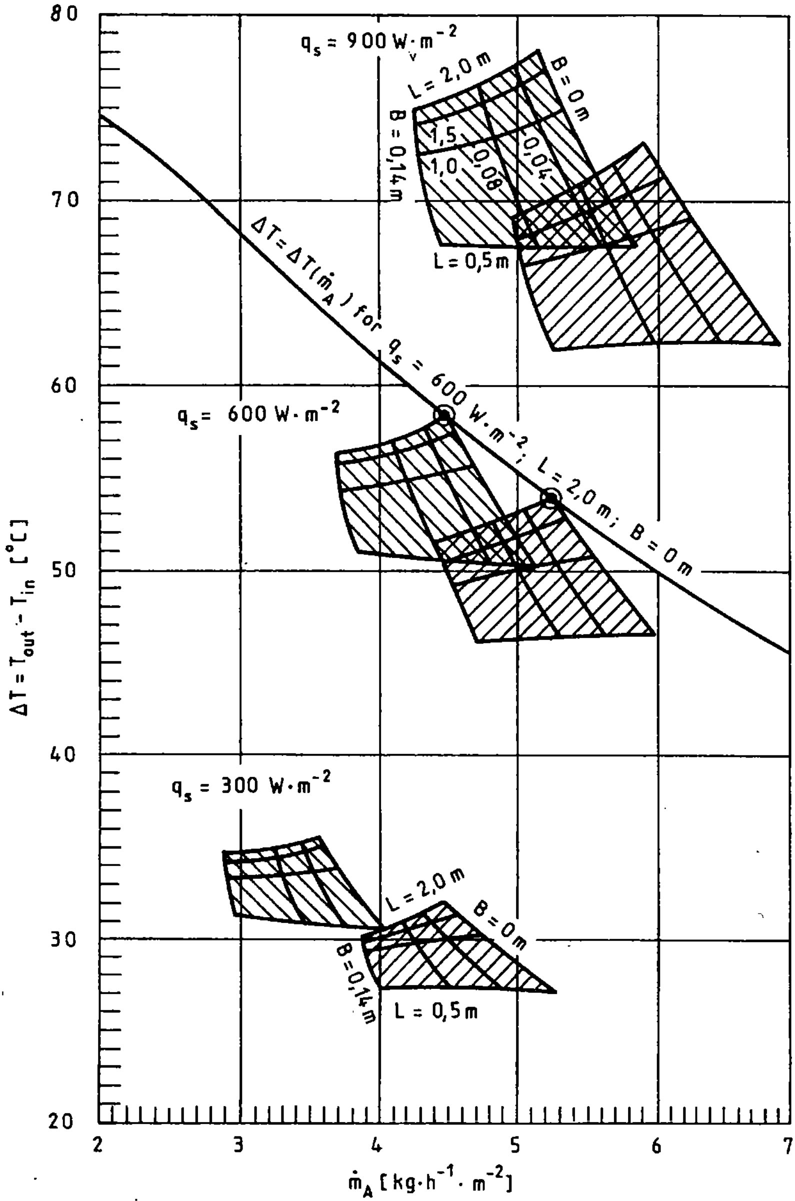

Fig.7. Ranges of $\dot{m}_A$ and ΔT corresponding to the maximum values of η_{ef} [cross-hatched] and η_b [hatched] (for $0{,}0 \le B \le 0{,}14$ m; $0{,}5 \le L \le 2{,}0$ m and q_s = 300, 600, 900 W·m^{-2}). Lines B = const and L = const are marked. Line $\Delta T = \Delta T(\dot{m}_A)$ with marked points of maximum η_{ef} and η_b is drawn

Fig. 7 shows the ranges of fluid mass flow rate per collector unit area ($\dot{m}_A = \dot{m}_c/A$) and the fluid temperature rises ($\Delta T = T_{out} - T_{in}$) corresponding to the maximum values of η_{ef} and η_b. Inside the hatched regions the lines of the constant B and L are marked. For the exemplary dimensions (B = 0 m; L = 2,0 m) and irradiance

(q_s = 600 W·m^{-2}) the line $\Delta T = \Delta T(\dot{m}_A)$ has been drawn. To illustrate better how the hatched regions were obtained, on the line $\Delta T = \Delta T(\dot{m}_A)$ the points of maximum η_{ef} and η_b are marked (compare with Fig. 4 and 5).

Calculations for the parallel plate absorber have shown that this type of absorber is the limit case of a parallel tube absorber (where $B \rightarrow 0$) – which is shown in Fig. 8.

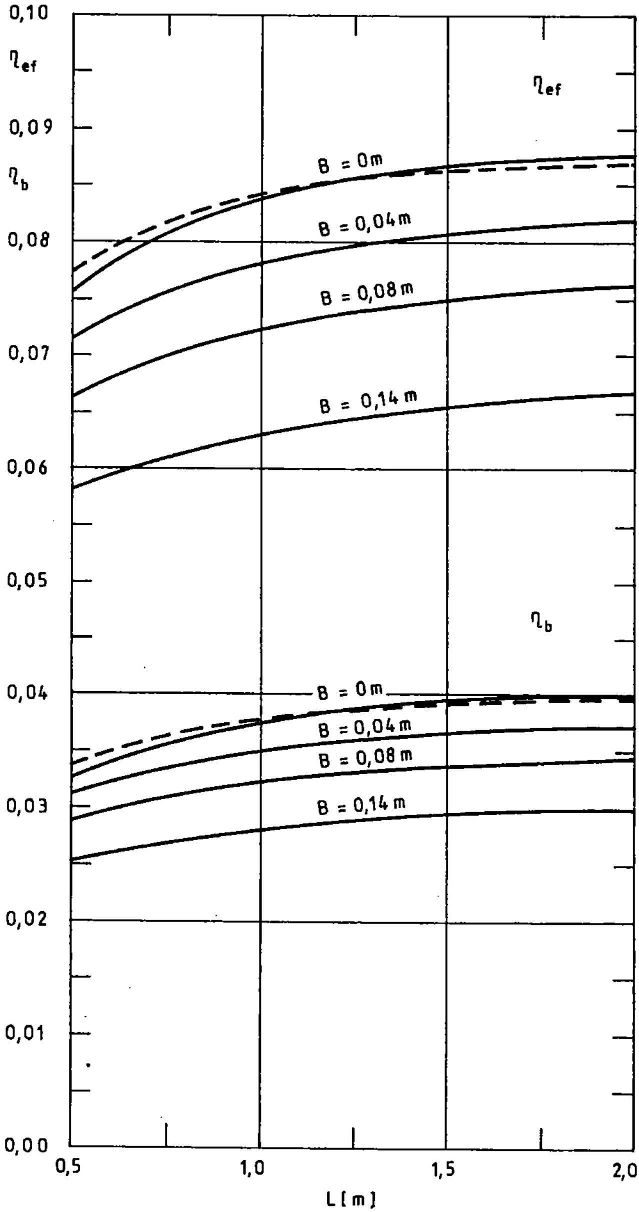

Fig.8. Maximum values of η_{ef} and η_b vs L for parallel tube absorber (———) and parallel plate absorber (– – – –); q_s = 600 W·m^{-2}

6. CONCLUSIONS

It has proved possible to determine the quantities describing the operation of a flat-plate liquid heating solar collector in a more comprehensive manner than could be done by means of collector thermal efficiency. These quantities, unlike the thermal efficiency of the collector, reach the local maxima for fluid mass flow rate per collector unit area of a few $kg \cdot h^{-1} \cdot m^{-2}$ (which corresponds with [6,7,8,9]) and relatively high temperature rises (e.g. $\Delta T \cong 60°C$ for $q_s = 600\ W \cdot m^{-2}$). In view of the operating conditions, the above-mentioned collectors might be called „low mass flow rate collectors" (LMFR-collectors)[1].

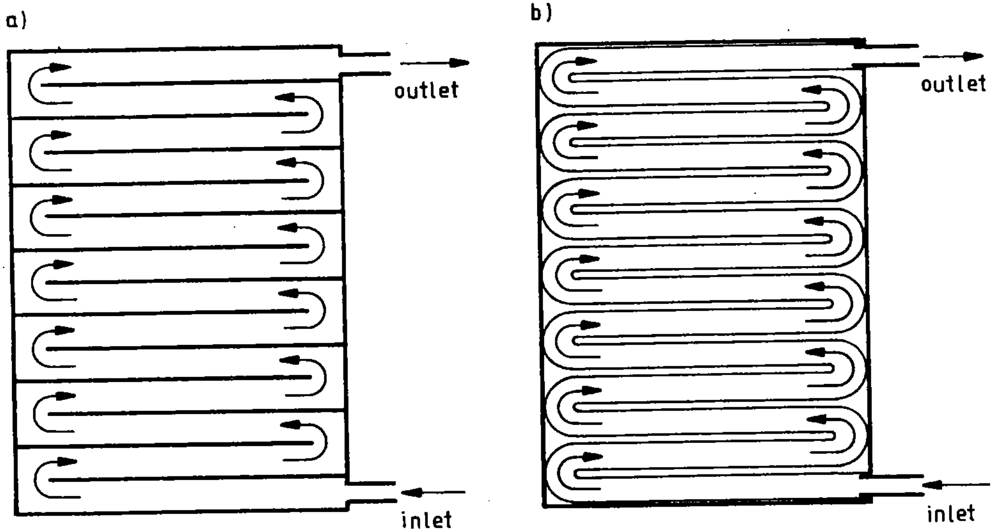

Fig.9. Proposal of LMFR-collector constructions: a) parallel plate absorber with „dividing walls", b) absorber made of a tube laid in the form of „compact serpentine" (preferred design)

The attained maximum values of collector effectiveness η_{ef} and exergetic efficiency η_b show that the most efficient collectors are those relatively long with the distance between tubes as small as possible (in the limit case – collectors with parallel plate absorbers) – that is collectors in which the absorber is in contact with the fluid on the area as large as possible (Fig. 6 and 8). From the technical point of view, the ensuring of equal flow rates in each of the tubes (or a constant intensity of flow in the whole cross-section of a gap) is very difficult [10]. Hence, it is evi-

[1] The term „low flow" the autors borrowed from the investigations carried out in the Thermal Insulation Laboratory of the Technical University of Denmark.

dent that a somewhat different collector construction should be considered, in which it would be possible to obtain uniform fluid flow and at the same time considerable absorber-fluid contact (Fig. 9). Fig. 9a shows a parallel plate absorber with „dividing walls", permitting a serpentine flow of the fluid, whereas in Fig. 9b the absorber is made of one tube laid in the form of a „compact serpentine". In an absorber of this type it would be possible to obtain an apparent slow fluid flow from the „cold" ($T_f \cong T_{in}$) edge to the „hot" ($T_f \cong T_{out}$) edge of the collector, although the real flow in the tube (channel) would be rather fast (which would additionally intensify the absorber-fluid heat transfer). Preliminary calculations showed that for the above constructions the amount of energy used by the pump would not increase significantly.

On the basis of the analysis results it is also evident that in LMFR-collectors the temperature rises are relatively high, while the irradiances have „average" values, which might be of great importance for such countries as Great Britain, France, Germany, Denmark, Poland, etc.

Moreover, it is evident (Fig. 7) that the effective use of LMFR-collectors would require controlling the mass flow rate in accordance with the irradiance (e.g. $\dot{m}_A = \text{const}(q_s)$ or $T_{out} = \text{const}(q_s)$).

It should be mentioned that the calculations were made for selectivity $\alpha/\varepsilon = 1$. For selective surfaces the „map" in Fig. 7 would have another but similar form. For $\alpha/\varepsilon > 1$ the outlet fluid temperature might exceed 100°C – in this case the fluid mass flow rate should be increased (automatically increasing the collector thermal efficiency) to ensure $T_{out} < 100°C$.

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MAŁE WYDATKI MASOWE W PŁASKICH CIECZOWYCH KOLEKTORACH SŁONECZNYCH

Streszczenie

Analizowano zagadnienia wymiany ciepła w płaskich cieczowych kolektorach słonecznych. Użyto pewnych udoskonalonych kryteriów sprawności. Wykazano, że niskie wydatki masowe (NWM) czynnika roboczego są korzystnym zakresem pracy dla kolektorów słonecznych. Jako udoskonaloną konstrukcję kolektora typu NWM zaproponowano szczególnego rodzaju absorber serpentynowy.

НЕБОЛЬШИЕ МАССОВЫЕ ПОТОКИ ЖИДКОСТИ В СОЛНЕЧНЫХ КОЛЛЕКТОРАХ

Краткое содержание

В работе анализируются проблемы теплопереноса в плоских жидкостных солнечных коллекторах.

Представлены некоторые усовершенствованные критерии термических коэффициентов полезного действия коллекторов. Показано, что небольшие массовые потоки жидкости (НМПЖ) равнозначны с благоприятными условиями работы солнечных коллекторов. Предложена усовершенствованная конструкция коллектора типа НМПЖ с особенным серпантиновым абсорбером.