621.472:536.24

Paper No. 211-13

Transient Characteristics of Flat-plate Solar Collector*

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A flat-plate solar collector model with uniform capacity distribution was proposed, and its basic equations were derived, to discuss transient characteristics for periodic solar-intensity variation.

The finite difference calculations based on these equations showed better agreement with the experimental results than the results calculated by a previous method.

Then, for the general treatment of collectors covering a wide purpose, the basic equations were normalized, and eleven non-dimensional parameters affecting the outlet water temperature variation were clarified. The influences of the parameters were discussed and summarized.

Key words: Thermal Engineering, Thermal Engineering Equipment, Thermal Conduction, Solar Energy, Flat Plate Collector, Transient Characteristics

1. Introduction

As one of the characteristic problems in solar energy utilization, the fluctuation of solar intensity is pointed out. Particularly under the cloudy, rainy and snowy weather condition like in Japan, the short period fluctuation of solar intensity due to cloud movement appears obviously.

This research is concerned with the evaluation of transient performance of flat plate solar collectors for air conditioning and hot water supply, which are subjected to comparatively short period solar intensity fluctuation, and aims to get basic information about the influences of physical parameters composing the collector on the heat transfer of the total solar energy utilization system.

Since a work by Hottel and Woertz⁽¹⁾, so many researches not only about a flat plate solar collector, but also about a large-scale solar collector for electric power generation have been performed.^{(2)~(4)} Almost all of those works in the past were based on the assumptions of steady state, quasi-steady state or uniform temperature distribution (the lumped capacitance model).

Klein et al.⁽⁵⁾ improved the zero-capacitance model in the past, where the heat capacity of a collector was neglected, and proposed two lumped capacitance models named "one-node" and "multi-node" models, where the heat capacity of a collector is

assumed to be concentrated upon one or a few nodal points. They performed the calculation using these models. To discuss the accuracy of the multi-node model, which is more precise than the one-node model, in the case of periodic solar intensity fluctuation considered in this paper, the calculated results using the multi-node model are shown in Fig. 1, compared with the experimental results by authors for the period of 600 seconds (10 minutes). In Fig. 1, symbols $\blacktriangle$, $\triangle$ and $\square$ denote the measured temperature at the center of the collector plate, near the exit of the collector plate and at the cen-

Fig. 1 Comparison between experimental results and analytical results by Multi node model

* Received 16th February, 1981.

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ter of the glass cover, respectively. The solid line shows the calculated results. The experimental apparatus and the method are shown later. It is understood from the figure that the calculated results disagree seriously with the measured results both for the collector plate and the glass cover. This is partly due to their lumped capacitance assumption, that the collector capacity concentrates upon a few points. Therefore, Klein's model cannot predict the periodic response of a collector sufficiently, for comparatively short period intensity fluctuation considered in this paper. Hahne et al.⁽⁶⁾ showed the heat balance equation obtained from a relatively exact physical model considering the capacity distribution, to clarify the steady or transient performances of a flat plate collector subjected to a stepwise insolation. But they didn't discuss sufficiently about the response for periodic intensity fluctuation.

In this paper, the following were performed, to clarify the periodic characteristics of the outlet water temperature variation in representative flat plate solar collectors due to the solar intensity fluctuation.

(1). A realistic physical model with distributed heat capacity was proposed. Basic equations of the physical model were derived, considering the energy balance.

(2). Those equations were normalized to discuss the periodic characteristics generally, and non-dimensional parameters affecting the characteristics were shown.

(3). Experimental results were compared with the calculated results by the above equations (item(1)), to confirm the accuracy of the physical model.

(4). The influences of the parameters (item(2)) on the outlet water temperature variation were discussed.

Nomenclatures

C : heat capacity of the collector per unit area, $J/(m^2K)$
 c : specific heat of a fluid, $J/(kgK)$
 d : plate thickness, m
 h : heat transfer coefficient, $W/(m^2K)$
 I : solar intensity, W/m^2
 k : thermal conductivity, $W/(mK)$
 L : the distance which the fluid moves in a half period, m
 l : collector length (Ref. Fig. 2), m
 m : volumetric water flow rate, m^3/s
 T : a half period, s
 t : time, s
 t' : $\equiv t - t_s$

Fig. 2 Physical model

v : fluid mean velocity, m/s
 W : collector width, m
 x : an axis over the collector surface (Ref. Fig. 2), m
 α : absorptivity
 θ : temperature, $^{\circ}C$
 δ : temperature rise without heat loss, $^{\circ}C$
 κ : thermal diffusivity, m^2/s
 ρ : fluid density, kg/m^3
 σ : Stefan-Boltzmann's constant ($\div 5.67 \times 10^{-8}$), $W/(m^2K^4)$
 τ : transmissivity
 (subscripts)
 a : ambient value
 f : value of the fluid layer
 fp : value between the fluid and the collector plate
 g : value of the cover plate
 i : value of the fluid at the collector entrance
 max : the maximum value
 min : the minimum value
 o : value of the fluid at the collector exit
 p : value of the collector plate
 s : the value corresponding to the maximum fluid outlet temperature for $l/L = 0.2$.
 t : total value including the collector plate, the cover plate and the fluid layer
 $\sim$: the equivalent value considering both convection and radiation
 1 : value between the air layer and the cover or the collector plate
 2 : value between the cover and the ambient

2. Physical model

The solar collector was simplified for analysis, and the physical model shown in Fig. 2 was proposed, which consisted of a transparent cover, an air layer, a collector plate and a fluid layer. Differential equations for the physical model were derived from heat balance at places along the fluid flow direction (x -direction). The physical model is comparatively exact, considering heat transfer between plates and ambient air, the air layer and the fluid layer, heat conduction within the cover plate, the collector plate, and heat capacities of the cover plate, the collector plate and the fluid layer. The assumptions for the model are as follows.

(1). Temperature distributions in the cover plate, the collector plate and the fluid layer are one-dimensional and vary only in the flow direction (x -direction).

(2). The fluid temperature at a place along the flow direction is represented by the bulk temperature at the place.

(3). Heat transfer to the fluid is quasi-steady.

(4). The bottom surface of the fluid layer is adiabatic.

(5). Heat capacity of the air layer is negligible.

The convection and radiative heat transfers coexist within the air layer.

The propriety of assumption(1) is shown in Chapter 4.

From the above assumptions, the follow-

ing basic equations for a collector are obtained.

$$C_p \frac{\partial \theta_p}{\partial t} = \tau_g \alpha_p I - \bar{h}_1 (\theta_p - \theta_g) + d_p k_p \frac{\partial^2 \theta_p}{\partial x^2} - h_{fp} (\theta_p - \theta_f) \quad (1)$$

$$C_f \frac{\partial \theta_f}{\partial t} = h_{fp} (\theta_p - \theta_f) - \frac{mc\rho}{W} \frac{\partial \theta_f}{\partial x} \quad (2)$$

$$C_g \frac{\partial \theta_g}{\partial t} = \alpha_g I + \bar{h}_1 (\theta_p - \theta_g) + d_g k_g \frac{\partial^2 \theta_g}{\partial x^2} - \bar{h}_2 (\theta_g - \theta_a) \quad (3)$$

Here, equivalent heat transfer coefficients $\bar{h}_1$ and $\bar{h}_2$ are defined at cover plate surfaces and at the upper surface of the collector plate, considering radiation.

$$\bar{h}_1 = \left\{ \frac{\sigma(\theta_p^4 - \theta_g^4)}{(1/\varepsilon_p) + (1/\varepsilon_g) - 1} + h_1(\theta_p - \theta_g) \right\} / (\theta_p - \theta_g) \quad (4)$$

$$\bar{h}_2 = \varepsilon_g \sigma(\theta_g^4 - \theta_a^4) / (\theta_g - \theta_a) + h_2 \quad (5)$$

Boundary conditions at collector outlet and inlet are:

$$x=0, x=l : \partial \theta_p / \partial x = 0, \partial \theta_g / \partial x = 0 \quad (6)$$

$$x=0 : \theta_f(t, 0) = \theta_i \quad (7)$$

Next, Eqs. (1)~(7) were normalized, using non-dimensional length x^* , non-dimensional time t^* and non-dimensional temperatures θ_p^* , θ_f^* , θ_g^* and θ_a^* shown below.

$$x^* = x/L \quad (8)$$

$$t^* = t/T \quad (9)$$

$$\theta_p^* = (\theta_p - \theta_i) / \bar{\theta} \quad (10)$$

$$\theta_f^* = (\theta_f - \theta_i) / \bar{\theta} \quad (11)$$

$$\theta_g^* = (\theta_g - \theta_i) / \bar{\theta} \quad (12)$$

$$\theta_a^* = (\theta_a - \theta_i) / \bar{\theta} \quad (13)$$

Where, T denotes a half period and L denotes the distance which the fluid moves in a half period. That is, L is given by the following relation, using the mean fluid velocity v .

$$L = vT \quad (14)$$

Fig. 3 Experimental apparatus

$\bar{\theta}$ means the temperature rise for the imaginary condition where a stationary fluid layer, the cover plate and the collector plate receive the solar energy of the intensity I for time T without heat loss.

$$\bar{\theta} = \frac{TI_{\max}}{C_g + C_p + C_f} = \frac{TI_{\max}}{C_t} \quad (15)$$

Then, the normalized basic equations and boundary conditions become Eqs. (16)~(18) and Eqs. (19)~(20).

$$\begin{aligned} \frac{\partial \theta_p^*}{\partial t^*} = & \xi_9 \left(1 + \frac{1}{\xi_1} + \frac{1}{\xi_2} \right) \left(\frac{I}{I_{\max}} \right) \\ & - \frac{\xi_3 \xi_7}{\xi_2} (\theta_p^* - \theta_g^*) - \frac{\xi_5 \xi_6 \xi_7}{\xi_2} (\theta_p^* - \theta_f^*) \\ & + \frac{\xi_6 \xi_7 (\xi_8)^2}{\xi_2} \frac{\partial^2 \theta_p^*}{\partial (x^*)^2} \quad (16) \end{aligned}$$

$$\frac{\partial \theta_f^*}{\partial t^*} = \frac{\xi_1 \xi_5 \xi_6 \xi_7}{\xi_2} (\theta_p^* - \theta_f^*) - \frac{\partial \theta_f^*}{\partial x^*} \quad (17)$$

$$\begin{aligned} \frac{\partial \theta_g^*}{\partial t^*} = & \xi_{10} \left(1 + \xi_2 + \frac{\xi_2}{\xi_1} \right) \left(\frac{I}{I_{\max}} \right) \\ & + \xi_3 \xi_7 (\theta_p^* - \theta_g^*) - \xi_4 \xi_7 (\theta_g^* - \theta_a^*) \\ & + \xi_7 \xi_8 \frac{\partial^2 \theta_g^*}{\partial (x^*)^2} \quad (18) \end{aligned}$$

$$x^*=0, x^*=\xi_8 : \partial \theta_p^* / \partial x^* = \partial \theta_g^* / \partial x^* = 0 \quad (19)$$

$$x^*=0 : \theta_f^* = 0 \quad (20)$$

$\xi_1 \sim \xi_{10}$ are the following non-dimensional parameters.

$$\xi_1 = C_p / C_f, \quad \xi_2 = C_p / C_g, \quad \xi_3 = \bar{h}_1 l^2 / (d_g k_g)$$

$$\xi_4 = \bar{h}_2 l^2 / (d_g k_g), \quad \xi_5 = h_{fp} l^2 / (d_p k_p)$$

$$\xi_6 = d_p k_p / (d_g k_g), \quad \xi_7 = \kappa_g T / l^2$$

$$\xi_8 = l / L, \quad \xi_9 = \alpha_p \tau_g, \quad \xi_{10} = \alpha_g$$

When non-dimensional parameters $\xi_1 \sim \xi_{10}$ and θ_i^* are given, time-dependencies of non-dimensional temperature distributions of the cover, the collector plate and the fluid layer, corresponding to the initial temperature and the solar intensity variation, can be calculated by Eqs. (16)~(20). For

Fig. 4 Details of the collector

Table 1 The standard value and the variable range for each physical quantity

physical quantity	standard value	variable range
C_p kJ/m ² ·K	3.12 *	0.624~6.24
C_f "	5.42 *	2.71~10.8
C_g "	5.53 *	2.77~11.1
C_i "	14.1 *	7.05~28.2
$d_p k_p$ W/K	1.30×10^{-2} *	$6.50 \times 10^{-5} \sim 2.60 \times 10^{-1}$
$d_g k_g$ "	2.23×10^{-3} *	$2.23 \times 10^{-4} \sim 4.46 \times 10^{-3}$
$\tilde{h}_1$ W/m ² ·K	6.85 **	3.43~13.7
$\tilde{h}_2$ "	9.57 **	$9.57 \sim 9.57 \times 10^3$
h_{fp} "	2.31×10^3 **	$1.16 \times 10^3 \sim 4.62 \times 10^3$
l m	1.98 *	0.990~3.96
L "	9.85	$6.57 \times 10^{-1} \sim 1.48 \times 10^2$
T s	9.00×10^3	$3.00 \times 10^2 \sim 2.70 \times 10^3$

* value decided by the specifications

** value estimated from the standard working conditions

calculations of periodic steady states, shown later in Chapter 5, the initial condition is not necessary as a matter of course.

3. Experimental apparatus and procedures

Experiments were performed indoors to keep the surrounding conditions constant. The whole apparatus is shown in Fig.3. The apparatus consists of a flat plate collector placed horizontally, infrared lamps to simulate the solar radiation, a constant temperature tank with a temperature control unit and a pump. Fig.4 shows the construction and the details of a collector on the market (blue panel by Yazaki So-gyo) used in this experiment. This is one of the usual flat plate collectors, consisting of a cover glass, a collector plate with the selective surface, fluid ducts for a heat transfer medium and insulators. Specifications of the collector and the maximum variation ranges for each physical quantities in usual flat plate collectors are shown in Table 1.

Experiments were done using water as the heat transfer medium, and keeping the inlet water at almost constant temperature, $70 \pm 0.5^\circ\text{C}$. Radiative heat flux from the infrared lamps is adjustable to six intensity steps ($I/I_{\max} = 1/6, 1/3, 1/2, 2/3, 5/6, 1$), varying the number of lighting lamps (the spectrum distribution is unchanged), so that the normal solar intensity variation, $138 \sim 826 \text{ W/m}^2$, is available on the cover glass surface. Lamps were arranged, such that uniform radiative heat flux could be obtained. The intensity distribution over the collector surface ($2\text{m} \times 1\text{m}$) was confirmed to be within about $\pm 10\%$, using a pyranometer (MS-42type, by Eiko Seiki). Temperature of the collector plate, the glass cover and the surrounding air were measured by Cu-Co thermocouples of 0.1mm diameter. And inlet and outlet water temperature were measured by ones of 1.5mm diameter. Thermocouples were spot welded along the center line of the collector plate surface at places of 0.21m, 0.97m, 1.74m distances from the entrance header. Thermocouples were also put on the cover glass with transparent ad-

hesive at almost the same places with the collector plate.

The flow rate was measured by a rotor meter, calibrated in the flow rate and temperature ranges $3.33 \times 10^{-6} \sim 3.33 \times 10^{-5} \text{ m}^3/\text{s}$ and $20 \sim 90^\circ\text{C}$.

The experimental procedure is as follows.

(1). The flow rate and the inlet water temperature (70°C) are set at constant values.

(2). The collector is preheated by a constant radiative heat flux for more than $1.2 \times 10^3 \sim 1.8 \times 10^3$ seconds, to get the steady initial collector condition.

(3). Lighting-on and off are repeated

Fig. 5 Transverse temperature distribution of the collector plate

Fig. 6 Temperature response of the collector

every T seconds (a half period), and the temperature variations of the glass cover, the collector plate and the fluid are recorded.

Experiments simulating the real sunshine fluctuation were also performed, in the above (2) and (3) processes.

4. Analytical and experimental results

4.1 Discussion on the transverse temperature distribution of the collector plate

The temperature variations at concave and convex points over the collector plate near the fluid entrance were measured to discuss the transverse temperature distribution. The results are shown in Fig. 5. It is clear from the results, that the collector plate temperature is uniform in transverse direction. Therefore, the propriety of the assumption (1) in Chapter 2 is confirmed.

4.2 Comparison between analyses and experiments

Analytical results by the above-mentioned basic equations are shown in Fig. 6, compared with the corresponding experimental results. Calculations were performed numerically, converting Eqs. (16) ~ (18)

(a) Measured global solar radiation fluctuation and its band approximation (Aug. 13, 1979, Tokyo)

(b) Outlet water temperature variation of the collector

Fig. 7 Comparison between simulating experiment and calculation

into explicit type simultaneous finite difference equations. In experiments, the collector was preheated to steady state, and then the rectangular-wise heating was done, repeating the cycle of lighting-on and off every 300 seconds (the period is 600 seconds). The equivalent heat transfer coefficients, Eqs. (4), (5), used in calculations, were approximated, for simplicity, by mean values (constant values) for steady lighting-on and lighting-off cases, because the temperature variation was small. The value of $I/I_{\max}$ in Eqs. (16), (18) becomes 0 and 1 for lighting-off and lighting-on cases, respectively.

Figures 7(a) shows the measured global solar radiation fluctuation. Figure 7(b) shows the calculated and measured temperature responses θ_0 of the outlet water, for the case of Fig. 7(a). Radiative heat flux from the infrared lamps was adjusted using six intensities between 138 and 826 W/m², to simulate the real solar radiation fluctuation. The approximated flux is shown by the solid line in Fig. 7(a). Calculated results agree well with the experimental results in Figs 6 and 7(b). Therefore, it is concluded that the collector plate temperature distribution and the outlet water temperature variation can be predicted accurately, using the basic equations of the physical model proposed in this paper.

Table 2 The standard value and the variable range for each non-dimensional parameter

non-dimensional parameter	standard value	variable range
$\xi_1 \equiv C_p/C_f$	0.5	$5.0 \times 10^{-2} \sim 2.0$
$\xi_2 \equiv C_p/C_g$	0.5	$5.0 \times 10^{-2} \sim 2.0$
$\xi_3 \equiv \tilde{h}_1 l^2 / (d_g k_g)$	1.0×10^4	$5.0 \times 10^2 \sim 8.0 \times 10^5$
$\xi_4 \equiv \tilde{h}_2 l^2 / (d_g k_g)$	2.0×10^4	$2.0 \times 10^3 \sim 8.0 \times 10^7$
$\xi_5 \equiv h_{fp} l^2 / (d_p k_p)$	5.0×10^4	$3.0 \times 10^3 \sim 1.0 \times 10^8$
$\xi_6 \equiv d_p k_p / (d_g k_g)$	5.0	$1.0 \times 10^{-2} \sim 1.0 \times 10^3$
$\xi_7 \equiv \kappa_g T / l^2$	1.0×10^{-4}	$4.0 \times 10^{-7} \sim 5.0 \times 10^{-3}$
$\xi_8 \equiv l/L$	0.2	$5.0 \times 10^{-3} \sim 5.0$
$\xi_9 \equiv \alpha_p \tau_g$	0.78	$0.33 \sim 0.88$
$\xi_{10} \equiv \alpha_g$	0.16	$0.04 \sim 0.64$
$\theta_a^* \equiv C_f(\theta_a - \theta_i) / (I_{\max} T)$	0.0	$-1.5 \sim 2.0 \times 10^{-1}$

Table 3 Relation between the variation of a non-dimensional parameter and the variations of physical quantities composing the parameter

	the variation of composing physical quantities (times)								
	$\tilde{h}_1$	$\tilde{h}_2$	h_{fp}	C_p	C_g	C_f	$d_p k_p$	$d_g k_g$	l
ξ_1	$\frac{m}{n}$	$\frac{m}{n}$	$\frac{m}{n}$	$\frac{m}{n}$	$\frac{m}{n}$	$\frac{1}{n}$	$\frac{m}{n}$	$\frac{m}{n}$	1
ξ_2	$\frac{1}{\theta}$	$\frac{1}{\theta}$	$\frac{1}{\theta}$	$\frac{m}{\theta}$	$\frac{1}{\theta}$	$\frac{m}{\theta}$	$\frac{1}{\theta}$	$\frac{1}{\theta}$	1
ξ_3	m	1	1	1	1	1	1	1	1
ξ_4	1	m	1	1	1	1	1	1	1
ξ_5	1	1	m	1	1	1	1	1	1
ξ_6	1	1	m	1	1	1	m	1	1
ξ_7	m	m	m	1	1	1	m	m	1
ξ_8	1	1	1	1	1	1	m^2	m^2	m
θ_a^*	m	m	m	m	m	m	m	m	1

where, $\begin{cases} n \equiv (mC_p + mC_g + C_f)/C_i \\ \theta \equiv (mC_p + C_g + mC_f)/C_i \end{cases}$

5. Influences of the physical quantities on the periodic performances

In this chapter, the sizes and the physical properties (materials) of the collector elements, and the operating conditions (which are called physical quantities in the subsequent discussions) are chosen more generally and widely to discuss their influences on the collector performances by numerical calculations, using finite difference equations of the non-dimensional basic equations (16)~(20), because the accuracy of the basic equations shown in chapter 2 are confirmed.

As is clear from Eqs. (16)~(18), non-dimensional parameters, composing the basic equations, are ten parameters $\xi_1 \sim \xi_{10}$ which are shown in the left column of Table 2. In the calculation, the standard values of each non-dimensional parameters were first decided, based on the specifications (Ref. the column; standard value in Table 1) of a representative flat plate collector (Blue panel by Yazaki Sogyo). Next, the maximum and the minimum values of these parameters were estimated roughly, considering the sizes, the materials and the flow rate of the usual collectors (Ref. the column; range of the value in Table 1). Then, five values were decided for each parameter within the range, including the maximum and the minimum values. Concerning the collector plate, for example, an aluminum plate or a plastic plate was imagined for the maximum or the minimum thermal conductivity estimations. The standard values and the ranges thus decided are shown in Table 2. To clarify the physical meaning of each non-dimensional parameter, the cases, where

one of the non-dimensional parameters was multiplied by m , keeping the other non-dimensional parameters and the working or ambient conditions, $T, L, I, \theta_a - \theta_i$, constant, were discussed, and the corresponding variations of the composing physical quantities are shown in Table 3. For example, it corresponds to vary h_{fp} and $d_p k_p$ by m times that the parameter ξ_6 is varied by m times, keeping the other non-dimensional parameters constant.

The non-dimensional outlet fluid temperature θ_0^* was calculated, varying one or a few of the non-dimensional parameters. The results are shown in Figs 8~12 and Fig. 14. The abscissa $(t')^*$ is a non-dimensional time defined by the following equations[†].

$$\{t' \equiv t - t_s, \dots\dots\dots (21)$$

$$\{(t')^* \equiv t'/T \dots\dots\dots (22)$$

where, t_s denotes the time when the outlet water temperature becomes maximum for $l/L = 0.2$. In each figure, parameters, except for the chosen parameter, are all fixed to their standard values^{††}. Through Figs. 8 ~ 12 and Fig. 14, the broken line shows the result for standard condition shown in Table 2. The results are all for the periodic steady state, where $I/I_{\max}$ varies periodically to 1 or 0.

In Fig. 8(a), the heat capacity ratio of the collector plate and the fluid $\xi_1 = C_p/C_f$ is varied. It is understood from the figure that the outlet water temperature becomes higher and, at the same time, the temperature variation becomes larger, by increasing this parameter. Figure 8(b) shows the results where the heat capacity ratio $\xi_2 = C_p/C_g$ is varied. As is clear from the figure, the trend of outlet water temperature variation is similar to the case of Fig. 8(a), although the parameter variation is inverted. Therefore, it is concluded that the influences of non-dimensional parameters ξ_1 and ξ_2 are almost the same.

In Figs 9(a)(b), parameters $\xi_3 = \tilde{h}_1 l^2 / (d_g k_g)$ and $\xi_4 = \tilde{h}_2 l^2 / (d_g k_g)$ are varied, respectively. As shown in Table 3, it means that heat transfer coefficient $\tilde{h}_1$ between the glass cover and the collector plate, and heat transfer coefficient $\tilde{h}_2$ outside the glass cover are varied in proportion to the parameters. It is understood from these figures that the influence of heat transfer coefficient is small. For short period case, however, the influence of $\tilde{h}_1$ slightly appears. The above results support the approximation in Chapter 2, that $\tilde{h}_1$ and $\tilde{h}_2$ are constants.

In Fig. 10(a), the result, where $\xi_5 =$

(a) Influence of $\xi_1 = C_p/C_f$

(b) Influence of $\xi_2 = C_p/C_g$

Fig. 8

[†] t_s , thus decided, appears periodically every $2T$ times. But the calculated results are the same for any t_s , because the temperature distribution considered is periodic steady state.

^{††} Concerning parameters ξ_9 and ξ_{10} , five coupled values were considered as shown later in Fig. 10(c), because these parameters cannot vary independently by their physical meanings.

(a) Influence of $\xi_3 = \bar{h}_1 l^2 / (d_g k_g)$ (b) Influence of $\xi_4 = \bar{h}_2 l^2 / (d_g k_g)$

Fig. 9

(a) Influence of $\xi_5 = h_{fp} l^2 / (d_p k_p)$ (b) Influence of $\xi_6 = d_p k_p / (d_g k_g)$ (c) Influence of $\xi_9 = \alpha_p \tau_g$, $\xi_{10} = \alpha_g$

Fig. 10

$h_{fp} l^2 / (d_g k_g)$ is chosen as the parameter, is shown. From table 3, this means to vary fluid heat transfer coefficient h_{fp} in proportion to the variation of ξ_5 . When the parameter is less than unity, the trend that the outlet water temperature variation decreases, is obvious. For the parameter greater than unity, however, three curves almost agree with each other, and the influence of the parameter are scarcely recognized. In Fig. 10(b), the parameter is $\xi_6 = d_p k_p / (d_g k_g)$, which denotes the ratio of conductive properties. It is clear from Table 3, that $d_p k_p$ is also varied in proportion to the variation of ξ_6 in addition to the case of Fig. 10(a). Comparing Figs 10(a) (b), it is concluded that the influence of $d_p k_p$ is negligible, and that both parameters ξ_5 and ξ_6 show the similar influences. In Fig. 10(c), dependencies on $\xi_9 = \alpha_p \tau_g$ and $\xi_{10} = \alpha_g$ are discussed. When the absorptivity of the cover plate is large, a similar tendency with Figs 10(a)(b) is seen, but, in the usual absorptivity range, the influence on the outlet water temperature variation is negligible.

Figure 11 is the result of choosing $\xi_7 = \kappa_g T / l^2$ as the parameter. From Table 3, it is understood that all heat transfer coefficients $\bar{h}_1, \bar{h}_2, h_{fp}$ and conductive properties $d_p k_p, d_g k_g$ are varied in proportion to the variation of ξ_7 . When the parameter is small, the thermal response of the collector becomes very slow, and the temperature variation is small. For a large parameter value, on the other hand, the temperature level falls seriously, and at the same time, the heat loss becomes large, and the temperature rise is prevented. It is interesting to note that the temperature variation becomes maximum, for medium parameter values around unity.

Fig. 11 Influence of $\xi_7 = \kappa_g T / l^2$ Fig. 12 Influence of $\xi_8 = l / L$

Figure 12 shows the case, where the ratio $\xi_8 \equiv l/L$ of the collector length l and the fluid movement L for a half period is varied, which means the variations of the values l^2 , $dpkp$, $dgkg$ in proportion to the parameter. As parameter ξ_8 increases, the collector length l becomes relatively longer, and the fluid temperature rises considerably. It is characteristic that the temperature variation phase shifts with ξ_8 . Figure 13 shows the dependency of the amplitude $\{(\theta_0^*)_{\max} - (\theta_0^*)_{\min}\}/2$ of the non-dimensional outlet water temperature variation on ξ_8 . As ξ_8 increases, the amplitude varies like a wave at first, and gradually converges to an almost constant value in the range of $\xi_8 > 2$. It is also understood that the amplitude of the outlet water temperature variation shows maximum around $\xi_8 \approx 0.7$, when the other parameters are fixed to their standard values.

Fig. 13 Relation between the amplitude of θ_0 and l/L

Fig. 14 Influence of $\theta_a^* = C_t(\theta_a - \theta_i)/(T I_{\max})$

Table 4 Dependency of outlet water temperature variation on each non-dimensional parameter

influence of the parameter	non-dimensional parameter										
	ξ_1	ξ_2	ξ_3	ξ_4	ξ_5	ξ_6	ξ_7	ξ_8	ξ_9	ξ_{10}	θ_a^*
small			○	○					○	○	
appears only for small parameter value					○	○			○		○
increases the amplitude of variation	○	○							○		
shifts the phase									○		
the maximum amplitude of variation exists							○				
does not change the wave shape of variation											○
varies the temperature range									○		

Figure 14 is the case of choosing $\theta_a^* \equiv C_t(\theta_a - \theta_i)/(IT)$ as the parameter, where all heat transfer coefficients, heat capacities and conductive properties are varied in proportion to the parameter^{†††}. In Fig. 14, the mode of temperature response is unchanged. It is also characteristic that the amount of response is also unchanged for parameters more than zero.

From the above, the dependencies of outlet water temperature variation on parameters $\xi_1 \sim \xi_{10}$ and θ_a^* are summarized in Table 4.

6. Conclusion

The following conclusions were obtained, discussing the periodic characteristics of flat plate solar collectors due to the solar intensity fluctuation.

(1). A physical model to analyze the periodic characteristics, and its basic equations (1) ~ (3) were proposed. The model was proved to be appropriate by comparing with the experimental results.

(2). To consider the general and various flat plate solar collectors, the above equations were normalized into non-dimensional equations (16) ~ (18), and non-dimensional parameters $\xi_1 \sim \xi_{10}$, θ_a^* which concerns the periodic characteristics were clarified.

(3). By the numerical calculations, varying the above non-dimensional parameters, the dependencies of collector performance on those parameters, $\xi_1 \sim \xi_{10}$, θ_a^* were discussed. The results were summarized in Table 4.

Based on the above conclusions, authors intend to get a general empirical formula to predict the periodic characteristics of flat plate collectors.

Authors wish to express their appreciation to Mr. Makoto Ishida, Mr. Yasushi Yamana and Mr. Masashi Hasegawa, who were students of Tokyo Institute of Technology, for their great efforts to help this work.

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^{†††} From Table 2, there is a range where this parameter has a negative value. For such cases, it is considered that the temperature difference $(\theta_a - \theta_i)$ between the ambient and inlet water temperature becomes negative.