Theoretical and Experimental Study on Heat Transfer and Thermal Performance of Concentration Solar Collector of Horizontal Coaxial Cylinders

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Heat transfer and thermal performance of the concentrating solar collector of horizontal coaxial cylinders for a solar thermal power source are studied theoretically and experimentally. The design parameters considered in the theory are the concentration ratio, the diameters of a semi-transparent glass tube and a selective absorber tube, the effective length and the optical property of an absorber tube, the storage temperature, the pressure in the annulus, and the inlet temperature and the flow rate of a heat transfer medium. The thermal performance of the absorber system is evaluated by the outlet temperature of the fluid, the collector efficiency, the exergy efficiency and the final conversion efficiency. The experimental results confirm the calculation method of heat transfer and thermal performance.

1. Introduction

There are several plans for the design of a concentrating system for thermal use of solar energy and for a solar thermal electric power system. Such systems have to be designed considering not only the collector efficiency but also the final conversion efficiency, including the thermodynamic efficiency of a heat engine connected to the system. From this point of view, Howell and Bannert(1) discussed the optimum solar collector operation for maximizing the cycle output, obtaining valuable conclusions. However, in this case, the heat transfer characteristics were not discussed exactly. Although an appropriate treatment of heat transfer can be found for the case of a flat plate collector in many earlier papers(2), one can scarcely find an exact treatment of heat transfer for concentrating systems. In this paper, the concentrating solar collector of horizontal coaxial cylinders is taken as a representative design for a solar thermal power source. Its heat transfer characteristics, collector efficiency, exergy efficiency, final conversion efficiency (power output efficiency) and outlet temperature of a fluid are studied theoretically and experimentally. The design parameters considered in the theory are the concentration ratio, the diameters of a semi-transparent glass tube and a selective absorber tube, the effective length and the optical property of an absorber tube, the storage temperature, the pressure in the annulus, and the inlet temperature and the flow rate of a heat transfer medium. The outlet temperature of the heat transfer medium and the amount of solar beam radiation absorbed are determined from the heat balance calculation of radiative, convective and conductive heat transfer for various combinations of the parameters. In this investigation only the sensitive heat absorption is considered, and air is adopted as a heat transfer medium. An experimental set up for a practical solar energy absorber is designed and operated to confirm the calculation procedure and the calculated results.

Nomenclature

\[ A : \text{absorptivity} \]
\[ C_p : \text{specific heat} \]
\[ CR : \text{concentration ratio} \]
\[ D : \text{Diameter, m} \]
\[ F : \text{view factor} \]
\[ F_{\lambda} : \text{emissive power of the blackbody in the spectral region of } \lambda \]
\[ g : \text{gravitational acceleration} \]
\[ h : \text{heat transfer coefficient, } W/m^2\cdot K \]
\[ m : \text{flow rate of heat transfer medium, kg/s} \]
\[ T_{fb} : \text{bulk temperature, K} \]
\[ T_{go} : \text{glass temperature, K} \]
\[ T_e : \text{temperature of environment, K} \]
\[ (\approx 293.2 \text{ K in calculation}) \]
\[ T_f : \text{temperature of an absorber tube, K} \]
\[ T_{in}, T_{out} : \text{outlet and inlet temperatures of a fluid, K} \]
\[ L : \text{effective length of an absorber tube, m} \]
\[ M : \text{molecular weight, kg/mol} \]
\[ Nu : \text{Nusselt number} \]
\[ Pr : \text{Prandtl number} \]
\[ p : \text{pressure} \]
\[ Q_s : \text{total energy absorbed by an absorber tube, W} \]
\[ Q_{ht} : \text{heat transfer to a fluid, W} \]
\[ Q_{hr} : \text{heat loss by radiation from an absorber tube, W} \]
\[ Q_{hc} : \text{heat loss by convection and conduction or free molecule conduction from an absorber tube, W} \]
\[ Q_{ff} : \text{heat loss from an absorber tube, W} \]
\[ Q_{tc} : \text{heat loss by convection from a glass tube to the environment, W} \]

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** Graduate student
$Q_0$: heat loss by radiation from a glass tube to the environment, $W$
$Q_s$: solar beam irradiation, $W$
$R$: reflectivity
$R$: universal gas constant
$Re$: Rayleigh number
$Re$: Reynolds number
$U$: thermal conductance, W/m·K
Greek symbols
$\alpha$: accommodation coefficient
$\delta$: thermal boundary layer thickness in a tube, m
$\beta$: volume expansion coefficient, K$^{-1}$
$\eta_c$: Carnot efficiency
$\eta_e$: exergy efficiency
$\eta_m$: maximum heat engine efficiency
$K$: ratio of specific heats
$\eta_k$: final conversion efficiency (power output efficiency)
$\lambda$: thermal conductivity, W/m·K
$\lambda_f$: free molecule conductivity, W/m·K
$\lambda_g$: thermal conductivity of a glass tube, W/m·K
$\nu_b$, $\nu_w$: viscosities at bulk and wall temperatures, m$^2$/s
$\nu$: kinematic viscosity, m$^2$/s
$\zeta$: $2\delta t/D_{11}$
$\sigma$: Stefan Boltzmann constant, 5.67×10$^{-8}$ W/m$^2$·K$^{-4}$

Subscripts
cd: conduction
cv: convection
Di: based on Di
Do: based on Do
f: fluid
i: inside the glass tube
o: outside the glass tube
1: inner or $\sim$ 2.5 $\mu$m
2: outer or 2.5 $\mu$m

2. Theory of Heat Transfer and Thermal Performance

2.1 Heat balance calculation

2.1.1 Heat balance for convective and conductive heat transfer

A schematic diagram of the system considered in this investigation is shown in Fig.1.

(a) Convective heat transfer inside the absorber tube

In the case of a laminar flow(3), for the tube length $L$ shorter than the entrance length, the outlet temperature $T_{fm}$ of the fluid is calculated by the following equation:

$$\frac{T_{11} - T_{fm}}{T_{11} - T_{fi}} = \frac{4\eta_c(1 + \frac{1}{4^5} + \frac{1}{5^5} + \frac{1}{6^5} + \frac{1}{7^5})}{28^5},$$

where $\xi = 2\delta t/D_{11}$ is determined by

$$\frac{1}{28160} \frac{1}{5^3} \frac{1}{3522} \frac{1}{252} \frac{1}{1225} \frac{1}{12625} \frac{1}{7} \frac{3197}{33792} \frac{105}{1408} Re Pr (D_{11}/L),$$

(2)

$T_1$ is the wall temperature of the absorber tube, $T_{fi}$ is the inlet temperature of the fluid, $D_{11}$ is the inner diameter of the absorber tube and $\delta t$ is the thermal boundary layer thickness. For the tube length longer than the entrance length, the outlet temperature is calculated by

Fig.1 A schematic diagram of the collector treated.

$$\frac{T_{11} - T_{fm}}{T_{11} - T_{fi}} = \frac{102}{175} \exp(0.3981 - 16.47) \times \frac{1}{Re Pr (D_{11}/L)^{0.14}}.$$  

(3)

In the case of a turbulent flow, the mean heat transfer coefficient is calculated by the relation(4).

$$Nu = 0.023 Re^{0.8} Pr^{1/3} (D_{11}/\mu_l)^{0.14},$$

(4)

and the effect of the entrance region is neglected. The amount of heat transferred to the fluid, $Q_f$, in the absorber tube is calculated by

$$Q_f = \dot{m} (T_{fm} - T_{fo}) C_p.$$  

(5)

The thermophysical properties of the fluid are evaluated at the arithmetic mean temperature of inlet and outlet bulk temperatures.

(b) Convective and conductive heat transfer in the annulus

The correlating equations proposed by Kuehn and Goldstein(5) for heat transfer from a horizontal cylinder to a cylindrical enclosure by conduction and natural convection of laminar and turbulent flows are applied in this investigation. The Nusselt number $Nu_{Di}$ based on the inner diameter of the cylinder is calculated by

$$Nu_{Di} = \left( Nu_{Dcv}^{15} + Nu_{Dcv}^{15} \right)^{1/15},$$

(6)

where

$$Nu_{Dcv} = \left( \frac{1}{Nu_{cv}} + \frac{1}{Nu_{ocv}} \right)^{1/15},$$

$$Nu_{Dcv} = 2 \ln(1+2\left[0.518 Ra_{D1}^{1/4}(1 + 0.558 s/s) \right]^{155})$$

$$+ [0.1 Ra_{D1}^{15} (s/5)^{15}]^{-1},$$

$$Nu_{ocv} = 2 \ln(1+2\left[(1 + \frac{2}{1+0.22})^{5/3} + 0.587 Ra_{D1}^{15} (s/5)^{15} \right])^{-1} + [0.1 Ra_{D1}^{15} (s/5)^{15}]^{-1},$$

and

$$G = [(1 + 0.6 \frac{1}{Pr_{D1}})^{-5} + (0.4 + 2.6 \frac{1}{Pr_{D1}})^{-5}]^{-1/5},$$

(7)
$D_{12}$ and $D_{12}$ are the inner diameter of the glass tube and the outer diameter of the absorber tube, respectively. $R_{D1}$ and $R_{D0}$ are defined by the following equations,

$$R_{D1} = \frac{D_{12}^3}{4\pi} \frac{g \theta(T_1 - T_{1b})}{Pr/\nu^2}$$

$$R_{D0} = \frac{D_{12}^3}{4\pi} \frac{g \theta(T_{1b} - T_{01})}{Pr/\nu^2}$$  \hspace{1cm} (8)

where $g$ is the gravitational acceleration, $\beta$ is the volume expansion coefficient and $\nu$ is the kinematic viscosity. The bulk temperature $T_{1b}$ is obtained by superposing the heat balance between inner and outer cylinders as

$$T_{1b} - T_{01} = \frac{Nu'_{D1} D_{12}}{Nu_{D0} D_{12}}$$  \hspace{1cm} (9)

The thermophysical properties to calculate $R_{D1}$ and $R_{D0}$ are evaluated at the mean temperature of $T_1$ and $T_{1b}$, and that of $T_{01}$ and $T_{1b}$, respectively. The amount of heat $Q_{ic}$ transferred from the inner tube to the outer glass tube by convection and conduction in the annulus is calculated by

$$Q_{ic} = \pi L \frac{Nu'_{D1} D_{12}}{Nu_{D0} D_{12}} \lambda(T_1 - T_{1b})$$  \hspace{1cm} (10)

In the region of free molecule conduction at an extremely low pressure in the annulus, the amount of heat $Q_{ic}$ transferred by conduction is expressed by the following equation(6):

$$Q_{ic} = \pi \Delta L \frac{D_{12}^3}{4\pi} \frac{\alpha R_1}{1+\alpha R_1} \frac{p(273.2)^{1/2}(T_1 - T_{01})}{T_{01}}$$  \hspace{1cm} (11)

where

$$\alpha = \frac{1}{1+1.9 \alpha R_1}$$

and

$$\Delta_0 = \frac{1}{2} \left( \frac{R}{273.2} \right)^{1/2}$$  \hspace{1cm} (12)

The accommodation coefficient $\alpha$ is set equal to 0.9. The pressure in the annulus in the free molecule conduction region should satisfy the following condition.

$$p(\text{Pa}) \leq \frac{R_1}{(D_{12} - D_{12})(112 + D_{12})}$$  \hspace{1cm} (13)

(c) Convective heat transfer outside the semitransparent glass tube

The following relation(7),

$$Nu = \frac{hD_2}{\lambda} = 0.375 Re^{1/2} + 0.057 Re^{2/3}$$  \hspace{1cm} (14)

is adopted and the thermophysical properties are evaluated at the mean temperature of the glass surface temperature $T_{02}$ and the environmental temperature $T_e$. The total thermal conductance $U$ from the inner surface of the glass tube to the environment is expressed as

$$\frac{1}{U} = \frac{1}{2\pi R_1} \frac{D_{12}}{\pi D_{12}} \frac{p(273.2)^{1/2}}{T_{01}}$$

and the amount of heat $Q_{oc}$ transferred by convection is calculated by

$$Q_{oc} = \frac{U(T_{01} - T_e) L}{L}$$  \hspace{1cm} (16)

In the above calculations, the temperature difference at the inner and outer surfaces of the absorber tube is neglected since the material of high thermal conductivity is usually used for the tube.

2.1.2 Relations for radiative heat transfer

The radiative heat transfer in the system with both a selective absorber surface and an outer semitransparent glass tube is easily formulated. The radiative properties of selective absorber surface are assumed to change abruptly at the cut-off wavelength of 2.5 μm. The outer glass tube can usually be assumed to change its radiative properties at the same wavelength. The absorptivities and reflectivities adopted in this investigation are shown in Table 1. The thickness of glass tube is set at 2 mm. The net radiative heat loss $Q_{lr}$ from the absorber surface can be evaluated by the following equation.

$$Q_{lr} = \pi D_1 L \frac{A_{11}(1-R_{11})}{1-R_{11}(1-A_{11})} F_0 \nu 2.5(T_1)$$

$$+ \frac{A_{12}(1-R_{12})}{1-R_{12}(1-A_{12})} F_0 \nu 2.5(T_0)$$

$$- \pi D_1 L \frac{A_{11} A_{12}}{1-R_{11}(1-A_{12})} F_0 \nu 2.5(T_1)$$

$$+ \frac{A_{12} A_{20}}{1-R_{12}(1-A_{20})} F_0 \nu 2.5(T_0)$$  \hspace{1cm} (17)

where $F_1 \nu \lambda_2(T)$ is the emissive power of blackbody in the spectral region $\lambda_1 \nu \lambda_2(\mu m)$ at $T_1$, and $F_0$ is the view factor from the glass tube surface to the absorber tube surface. The radiative energy transmitted from the annulus to the environment is calculated by

$$Q_{ot} = \pi D_1 L \frac{A_{11}(1-R_{11})}{1-R_{11}(1-A_{11})} F_0 \nu 2.5(T_1)$$

$$+ \frac{A_{12}(1-R_{20}-A_{20})}{1-R_{12}(1-A_{20})} F_0 \nu 2.5(T_1)$$  \hspace{1cm} (18)

Table 1 The absorptivities and reflectivities adopted in this calculation.

<table>
<thead>
<tr>
<th>wavelength</th>
<th>absorber tube</th>
<th>glass tube</th>
</tr>
</thead>
<tbody>
<tr>
<td>&lt; 2.5 μm</td>
<td>$A_{11}=0.85$</td>
<td>$A_{12}=0.95$</td>
</tr>
<tr>
<td>&gt; 2.5 μm</td>
<td>$A_{01}=0.00$</td>
<td>$A_{02}=0.91$</td>
</tr>
<tr>
<td>case 1</td>
<td></td>
<td></td>
</tr>
<tr>
<td>case 2</td>
<td>$A_{11}=0.95$</td>
<td>$A_{12}=0.10$</td>
</tr>
<tr>
<td></td>
<td>$A_{01}=0.13$</td>
<td>$A_{02}=0.09$</td>
</tr>
</tbody>
</table>
The radiative energy emitted from the glass tube to the environment is calculated by

$$Q_{or} = \frac{\pi(D_{o1} + D_{o2})/2}{L} A_{o2} F_2 S_{\nu o}(T_o)^2, \quad T_o = \frac{T_{o1} + T_{o2}}{2}$$ (19)

2.2 Calculation procedure

The heat loss $Q_{ol}$ which passes through the glass tube to the environment is obtained by

$$Q_{ol} = Q_{or} + Q_{ot} + Q_{oc}.$$ (20)

The total heat loss $Q_{l1}$ from the absorber surface is obtained by

$$Q_{l1} = Q_{lc} + Q_{lx}.$$ (21)

The solar energy absorbed by the selective absorber surface $Q_{s}$ and that by the glass tube $Q_{oa}$ are approximately calculated by

$$Q_s = \frac{R_{M} A_{T1} - (1 - R_{o1})}{1 - (R_{T1})} 0.96$$ (22)

and

$$Q_{oa} = Q_{o} A_{o2} 0.04.$$ (23)

respectively, where $Q_s$ is the solar beam irradiation to the receiver mirror, and $R_{o1}$ is the overall reflectivity of the mirror system, including the shape factor. In the calculation $R_{o1}$ is assumed to be 0.85.

Thermal equilibrium condition of the system at constant solar beam irradiation is studied numerically, considering that $Q_s$ is equal to the sum of $Q_f$ and $Q_{l1}$.

$$Q_s = Q_f + Q_{l1}.$$ (24)

For thermal equilibrium about the glass tube, the following equation must be fulfilled.

$$Q_{l1} = Q_{ol} = Q_{oa}$$ (25)

First, the inlet temperature $T_{f1}$ of fluid is given. The temperature $T_1$ of absorber tube is assumed, and $Q_f$ can be calculated by eq. (5). Next, the inner surface temperature $T_{o1}$ of glass tube is assumed. From the temperatures $T_1$ and $T_{o1}$, $Q_{l1}$ and $Q_{oa}$ are calculated. These two quantities must fulfill the heat balance condition of eq. (25). Otherwise, a different $T_{o1}$ is assumed until it is determined finally by iteration. At the next stage, the heat balance condition of eq. (24) is checked. If the condition is not attained, a new $T_1$ is adopted, and the iteration is carried out until the thermal equilibrium is attained. In the above calculation, both the temperatures of absorber and the glass tube are assumed to be constant along the tube axis.

The thermal performance of the absorber system is evaluated by the outlet temperature of fluid $T_{fc}$, the collector efficiency

$$\eta = \frac{Q_{l1}}{Q_s},$$ (26)

the exergy efficiency

$$\eta_e = \frac{\eta_c \eta}{\eta_c},$$ (27)

and the final conversion efficiency

$$\eta_c^* = \eta_c \eta,$$ (28)

where $\eta_c$ is the Carnot efficiency

$$\eta_c = 1 - \frac{T_e}{T_f},$$ (29)

and $\eta_c^*$ is the maximum heat engine efficiency. Curzen and Ahlborn(8) evaluated the maximum heat engine efficiency, considering the heat transfer in the isothermal processes of the Carnot cycle, and introduced the following expression:

$$\eta_m = 1 - \left( \frac{T_l}{T_h} \right)^{1/2} \left( 1 - \left( \frac{T_l}{T_f} \right)^{1/2} \right).$$ (30)

This agreed well with the thermal efficiencies of practical power stations(9). $T_h$ and $T_f$ are the heat source and the heat sink temperatures of the Carnot cycle, respectively.

In the case of using an energy storage unit in the system, $\eta_f$ of eq.(29) and $\eta_m$ of eq.(30) may be replaced with

$$\eta_c = 1 - \frac{T_e}{T_{f1}}$$ (31)

and

$$\eta_m = 1 - \left( \frac{T_e}{T_{f1}} \right)^{1/2},$$ (32)

respectively.

2.3 Calculated results

In this investigation, only the results for the case of the standard solar beam radiation of 0.837 kW/m² are shown. One can find a good review of the local condition of the solar beam radiation on earth by Duffie and Beckman(10). A different amount of solar beam radiation from the standard is simply replaced with a different aperture of the mirror for constant diameter of absorber tube.

In Figs.2 and 3, the results of $T_{fc}$, $\eta$, $\eta_{o1}$ and $\eta^*$ are shown for different absorber tube diameters at a constant aperture ($\lambda_{o1}=0.36$m), for atmospheric pressure and for extremely low pressure of free molecule conduction region in the annulus. The ratio of mass flow rate to tube length is taken as abscissa. The larger values of $\eta_{o1}$ and $\eta^*$ are obtained for the smaller diameter, i.e. the larger concentration ratio. Although the collector efficiency $\eta$ has a similar asymptotic value at a very large mass flow rate region, the condition which gives the maximum $\eta_{o1}$ and $\eta^*$ is found in the region of medium $\eta$. Two nearly similar maxima of $\eta_{o1}$ or $\eta^*$ are found in one curve, corresponding to a laminar flow region and to the beginning of a turbulent flow region, respectively. Correspondingly, two different values of the ratio of flow rate to tube length can be adopted for the temperature of the higher temperature energy source of a heat engine. The effect of the optical properties of absorber surface is easily determined by comparison of the practical case of Fig.2 ($A_{T1}=0.85$, $A_{T2}=0.15$) with the ideal case of Fig.3($A_{T1}=0.95$, $A_{T2}=0.1$). Generally, the maximum values of $\eta_{o1}$ and $\eta^*$ for the latter case show an increase of 20-25%
Fig. 2 The result of $T_{fm}$, $n$, $n_e$, and $n^*$ for different tube diameters at a constant aperture ($A_1=0.36$) in the case of $A_{11}=0.85$, $A_{12}=0.15$ and $T_{fl}=293.2$ K.

Fig. 3 The result of $T_{fm}$, $n$, $n_e$, and $n^*$ for different tube diameters at a constant aperture ($A_1=0.36$) in the case of $A_{11}=0.95$, $A_{12}=0.10$ and $T_{fl}=293.2$ K. (The meanings of the notations and the lines are the same as those in Fig. 2)

over those for the former case. This difference is preserved in all cases of the following calculation. Accordingly, only the results for the practical case of $(A_{11}=0.85, A_{12}=0.15)$ are shown in the following.

In Figs. 4 and 5, the effect of the concentration ratio on the values of $T_m$, $n_e$ and $n^*$ is shown at atmospheric pressure and at a pressure of free molecule conduction region, respectively. From these figures, it is expected that the optimum combination of the ratio of mass flow rate to tube length and the concentration ratio exists for each absorber tube diameter.

Next, the effect of inlet temperature is examined. In the case of latent heat energy storage, the melting temperature or the operational temperature is assumed as the temperature of the higher temperature energy source of heat engine, and also as the inlet temperature of collector system. In Figs. 6 and 7, examples of these calculations are shown for a particular concentration ratio and tube diameter. We can
Fig. 6 The effect of the inlet temperature on the values of $\eta_0$ and $\eta^*$ in the case of $A_1=0.85$ and $A_2=0.15$ for the constant heat energy storage. (The meanings of the notations and the lines are the same as those in Fig. 6)

choose the optimum value of $T_{fi}$ to give the maximum $\eta_0$ and $\eta^*$. In Fig. 8, taking the inlet temperature of fluid as the abscissa, $\eta_0$ and $\eta^*$ are given for a constant concentration ratio and a constant absorber tube diameter, and we can easily find an optimum value of $T_{fi}$ for each case. In Figs. 9 and 10, the maximum values of $\eta_0$ and $\eta^*$ are given for smaller and larger absorber tube diameters, respectively, taking the concentration ratio as the abscissa. In all cases for the combinations of the pressure, the inlet temperature and the absorber tube diameter, the values of $\eta_{0\text{ max}}$ and $\eta^*_{\text{max}}$ at a comparatively small concentration ratio promptly approach the asymptotic values for a very large concentration ratio. So we do not have to adopt a very high concentration ratio which may hardly be attained from the view point of machining. Especially in the case of the usual system including a thermal storage unit, it is enough to adopt the value of 10 as the concentration ratio for the pressure of free molecule conduction region.

The sudden change of $T_{fi}$ found in the figures is due to the vanishing of one of the two maxima shown in Figs. 2 and 3, namely to the fact that the practical flow condition is sometimes limited to only a laminar flow or only a turbulent flow.

3. Experimental Study

3.1 Experimental setup

Experimental study on a solar collector of horizontal coaxial cylinders was carried out to substantiate the calculation method of heat transfer. A schematic diagram of the experimental setup is shown in Fig. 11. A cylindrical parabolic mirror of reflectivity 0.85 and of shape factor 0.63 was used. The outer tube of the coaxial cylinders is made of pyrex glass, having an outer diameter of 40 mm and an inner diameter of 36 mm. The inner tube, i.e. the absorber tube, is made of 18-8 stainless steel, having an outer diameter of 12 mm and an inner diameter of 8 mm. Its effective length of absorption is 618 mm, and both ends are thermally insulated by a cobalt glass tube. Three types of surfaces of the absorber tube are used; (a) a surface polished by 5 $\mu$m emery paper, (b) a surface coated with black paint, and (c) a selective absorber surface coated with oxide film. The optical properties of the glass and the absorber surfaces are shown in Fig. 12. The pressure in the annulus is maintained at 1.013x10^4 Pa for the usual conduction and convection region, and at about 0.6 Pa for the free molecule conduction region. The absorber system and the solar beam radiometer could follow the solar beam radiation with an accuracy of 11 degrees for solar altitude and azimuth angle. The temperatures of the heat transfer medium at inlet and outlet posi-
tions, and also those of the absorber tube and the glass tube at several points were measured by CA thermocouples. The flow rate of heat transfer medium was measured by the float flowmeter and by the capillary tube manometer. All the experiments were conducted under the stationary condition of solar beam irradiation.

3.2 Experimental results

The experimental results obtained for the outlet temperature of heat transfer medium, and the collector efficiency are shown in Figs.13(a)-(f). The numbers put to the data points indicate the equivalent concentration ratio with which the practical solar beam irradiation measured at each condition is replaced since the aperture of the cylindrical parabolic mirror is kept constant for all the experiments. The solid lines in the figures are the calculated values for the concentration ratios indicated by the method described in the preceding section. In the case of the polished absorber surface of Fig.13(a), the surface was partly oxidized after a short period operation, and different values of the optical properties (curve a' in Fig.12) were adopted for the calculation. In all the cases of the experiments, good agreement between the calculation and the experiment could be obtained, excluding several points in the low flow rate region of Fig.13(f). This discrepancy is due to the inferior selectivity caused by a long time operation at atmospheric pressure condition. Another discrepancy in temperature at the lowest flow rate region is due to the heat loss by conduction from the end of absorber tube although it is thermally insulated by the connection of a cobalt glass tube of low thermal conductivity. The discrepancy in $\eta$ at the largest flow region is due to the fact that even a small experimental error in temperature measurement causes a large error in $\eta$ in this region. The authors believe these experimental results confirm the calculation method of thermal performance of a solar energy collector of horizontal coaxial cylinders.

4. Conclusions

The conclusions which one can draw

Fig.9 The maximum values of $\eta_e$ and $\eta^*$, and the value of $T_{fm}$ for small $D_{12}$.

Fig.10 The maximum values of $\eta_e$ and $\eta^*$, and the value of $T_{fm}$ for large $D_{12}$. (The meanings of the notations and the lines are the same as those in Fig.9)

Fig.11 A schematic diagram of the experimental setup.

Fig.12 The optical properties of the glass and the absorber surfaces. a(the surface polished by 5 µm emery paper), c(the surface coated with black paint), e(the selective absorber surface), g(glass tube)
from the foregoing analysis and experiment are as follows.

(1) For the operational conditions of a concentrating system of solar thermal electric power, the heat transfer characteristics must be considered exactly. The operational conditions which give maximum exergy efficiency and maximum final conversion efficiency are found in the region of the medium collector efficiency.

(2) An optimum combination of the ratio of mass flow rate to the effective tube length and the concentration ratio exists for each absorber tube diameter.

(3) For a constant concentration ratio and a constant absorber tube diameter, one can find an optimum value of the inlet temperature of a heat transfer medium.

(4) One does not have to adopt a very high concentration ratio since the maximum values of exergy efficiency and final conversion efficiency at a comparatively small concentration ratio promptly approach the asymptotic values for the case of a very large concentration ratio.

Fig. 13 The experimental results for the outlet temperature of the heat transfer medium and the collector efficiency.

References


