621.472:536.33

Paper No. 199-10

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

Takeshi Kunitomo*, Kazuo Aizawa**, Kenji Marumoto**

and Hideki Tamehiro**

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 Bannerot⁽¹⁾ discussed the optimum solar collector operation for maximizing the cycle output, obtaining valuable conclusions. However, in this case, the heat transfer characteristics was 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 : absorptivity
CR : concentration ratio
D : Diameter, m
F : view factor
 $F_{\lambda_1 \sim \lambda_2}(T)$: emissive power of the blackbody in the spectral region of $\lambda_1 \sim \lambda_2$ at T
g : gravitational acceleration
h : heat transfer coefficient, $W/m^2 \cdot K$
 $\dot{m}$: flow rate of heat transfer medium, kg/s
 T_{fb} : bulk temperature, K
 T_{go} : glass temperature, K
 T_e : temperature of environment, K (≈ 293.2 K in calculation)
 T_i : temperature of an absorber tube, K
 T_{fm}, T_{fi} : outlet and inlet temperatures of a fluid, K
L : effective length of an absorber tube, m
M : molecular weight, kg/mol
Nu : Nusselt number
Pr : Prandtl number
p : pressure
 Q_i : solar energy absorbed by an absorber tube, W
 Q_f : heat transfer to a fluid, W
 Q_{ir} : heat loss by radiation from an absorber tube, W
 Q_{ic} : heat loss by convection and conduction or free molecule conduction from an absorber tube, W
 Q_{il} : heat loss from an absorber tube, W
 Q_{oc} : heat loss by convection from a glass tube to the environment, W
 C_p : specific heat of a fluid, J/kg·K

* Professor,
Department of Engineering Science,
School of Mechanical Engineering,
Kyoto University, Kyoto, Japan

** Graduate student

Q_{or} : heat loss by radiation from a glass tube to the environment, W
 Q_s : solar beam irradiation, W
 R : reflectivity
 $\bar{R}$: universal gas constant
 Ra : Rayleigh number
 Re : Reynolds number
 U : thermal conductance, W/m $\cdot$ K
 Greek symbols
 α : accommodation coefficient
 δt : thermal boundary layer thickness in a tube, m
 β : volume expansion coefficient, K $^{-1}$
 η : collector efficiency
 η_c : Carnot efficiency
 η_e : exergy efficiency
 η_m : maximum heat engine efficiency
 κ : ratio of specific heats
 η^* : final conversion efficiency (power output efficiency)
 λ : thermal conductivity, W/m $\cdot$ K
 λ_o : free molecule conductivity, W/m $\cdot$ K
 λ_g : thermal conductivity of a glass tube, W/m $\cdot$ K
 μ_b, μ_w : viscosities at bulk and wall temperatures, Ns/m 2
 ν : kinematic viscosity, m 2 /s
 ξ : $2\delta t/D_{i1}$
 σ : Stefan Boltzmann constant, 5.67×10^{-8} W/m $^2 \cdot$ K 4
 Subscripts
 cd : conduction cv : convection
 Di : based on Di Do : based on Do
 f : fluid i : absorber tube
 o : glass tube t : transmitted
 1 : inner or $\sim 2.5 \mu m$
 2 : outer or $2.5 \mu m$

2. Theoty of Heat Transfer and Thermal Performance

2.1 Heat balance calculation

2.1.1 Relation 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:

$$\frac{T_i - T_{fm}}{T_i - T_{fi}} = \frac{4}{4+\xi} \left(1 + \frac{1}{4}\xi + \frac{4}{5}\xi^2 + \frac{1}{5}\xi^3 + \frac{4}{35}\xi^4 - \frac{1}{28}\xi^5 \right), \quad (1)$$

where $\xi = 2\delta t/D_{i1}$ is determined by

$$-\ln \left(1 + \frac{\xi}{4} \right) + \frac{1}{4}\xi - \frac{1}{32}\xi^2 + \frac{3}{352}\xi^3 - \frac{25}{11264}\xi^4 - \frac{7}{28160}\xi^5 + \frac{5}{33792}\xi^6 = \frac{105}{1408} \frac{1}{RePr(D_{i1}/L)}. \quad (2)$$

T_i is the wall temperature of the absorber tube, T_{fi} is the inlet temperature of the fluid, D_{i1} is the inner diameter of the absorber tube and δt 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.

$$\frac{T_i - T_{fm}}{T_i - T_{fi}} = \frac{102}{175} \exp(0.3981 - 16.47 \times \frac{1}{Re \cdot Pr \cdot (D_{i1}/L)}). \quad (3)$$

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

$$Nu = 0.023 Re^{0.8} Pr^{1/3} (\mu_b/\mu_w)^{0.14} \quad (4)$$

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

$$Q_f = \dot{m}(T_{fm} - T_{fo})C_p. \quad (5)$$

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 cylinder is calculated by

$$Nu_{Di} = [(Nu_{Dicd})^{15} + (Nu_{Dicv})^{15}]^{1/15} \quad (6)$$

where

$$Nu_{Dicd} = 2/\ln(D_{o1}/D_{i2})$$

$$Nu_{Dicv} = 1 / \left(\frac{1}{Nu'_{Dicv}} + \frac{1}{Nu'_{Docv}} \right)$$

$$Nu'_{Dicv} = 2 \{ \ln[1 + 2 \{ (0.518 Ra_{Di}^{1/4} \{ 1 + \frac{0.559}{Pr} \}^{3/5} \}^{-5/12} \}^{15} + [0.1 Ra_{Di}^{1/3}]^{15} \}^{-1/15} \}^{-1},$$

$$Nu'_{Docv} = 2 \{ -\ln[1 - 2 \{ \left(\frac{2}{1 - e^{-0.25}} \right)^{5/3} + (0.587 Gr_{Do}^{1/4})^{5/3} \}^{3/5} \}^{15} + [0.1 Ra_{Do}^{1/3}]^{15} \}^{-1/15} \}^{-1}$$

and

$$G = \left[\left(1 + \frac{0.6}{Pr^{0.7}} \right)^{-5} + (0.4 + 2.6 Pr^{0.7})^{-5} \right]^{-1/5}. \quad (7)$$

D_{o1} and D_{i2} are the inner diameter of the glass tube and the outer diameter of the absorber tube, respectively. Ra_{Di} and Ra_{Do} are defined by the following equations,

$$Ra_{Di} = D_{i2}^3 g \beta (T_i - T_{fb}) Pr / \nu^2$$

$$Ra_{Do} = D_{o1}^3 g \beta (T_{fb} - T_{o1}) Pr / \nu^2 \quad (8)$$

where g is the gravitational acceleration, β is the volume expansion coefficient and ν is the kinematic viscosity. The bulk temperature T_{fb} is obtained by iteration considering the heat balance between inner and outer cylinders as

$$\frac{T_{fb} - T_{o1}}{T_i - T_{fb}} = \frac{Nu'_{Dicv}}{Nu'_{Docv}} \quad (9)$$

The thermophysical properties to calculate Ra_{Di} and Ra_{Do} are evaluated at the mean temperature of T_i and T_{fb} , and that of T_{o1} and T_{fb} , 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

$$Q_{ic} = \pi L \cdot Nu_{Di} \cdot \lambda (T_i - T_{fb}) \quad (10)$$

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):

$$Q_{ic} = \pi D_{i2} L \alpha_r \Lambda_o p \left(\frac{273.2}{T_{o1}} \right)^{1/2} (T_i - T_{o1}), \quad (11)$$

where

$$\alpha_r = \alpha / \{1 + (1 - \alpha) \frac{D_{o1}}{D_{i2}}\}$$

and
$$\Lambda_o = \frac{1}{2} \left(\frac{\kappa + 1}{\kappa - 1} \right) \left(\frac{\tilde{R}}{2\pi M 273.2} \right)^{1/2} \quad (12)$$

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

$$p(\text{Pa}) \leq \frac{6.27 \times 10^{-3} T_{fb}^2}{(D_{o1} - D_{i2})(112 + T_{fb})} \quad (13)$$

(c) Convective heat transfer outside the semitransparent glass tube
The following relation (7),

$$Nu = \frac{hD_{o2}}{\lambda} = 0.373 Re^{1/2} + 0.057 Re^{2/3} \quad (14)$$

is adopted and the thermophysical properties are evaluated at the mean temperature of the glass surface temperature T_{o2} 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

$$\frac{1}{U} = \frac{1}{2\pi\lambda_g} \ln \frac{D_{o2}}{D_{o1}} + \frac{1}{h\pi D_{o2}} \quad (15)$$

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

$$Q_{oc} = U(T_{o1} - T_e)L \quad (16)$$

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_{ir} from the absorber surface can be evaluated by the following equation.

$$Q_{ir} = \pi D_{i2} L \left\{ \frac{A_{i1}(1-R_{o1})}{1-R_{o1}(1-A_{i1})} F_{0 \rightarrow 2.5}(T_i) + \frac{A_{i2}(1-R_{o2})}{1-R_{o2}(1-A_{i2})} F_{2.5 \rightarrow \infty}(T_i) \right\} - \pi D_{o1} L F_{o1} \left\{ \frac{A_{i1}A_{o1}}{1-R_{i1}R_{o1}} F_{0 \rightarrow 2.5}(T_o) + \frac{A_{i2}A_{o2}}{1-R_{i2}R_{o2}} F_{2.5 \rightarrow \infty}(T_o) \right\} \quad (17)$$

where $F_{\lambda_1 \rightarrow \lambda_2}(T)$ is the emissive power of blackbody in the spectral region $\lambda_1 \rightarrow \lambda_2$ (μm) at T , and F_{oi} 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

$$Q_{ot} = \pi D_{i2} L \left\{ \frac{A_{i1}(1-R_{o1})}{1-R_{i1}R_{o1}} F_{0 \rightarrow 2.5}(T_i) + \frac{A_{i2}(1-R_{o2}-A_{o2})}{1-R_{i2}R_{o2}} F_{2.5 \rightarrow \infty}(T_i) \right\} \quad (18)$$

Table 1. The absorptivities and reflectivities adopted in this calculation.

	absorber tube		glass tube	
wavelength	< 2.5 μm	$\geq 2.5 \mu\text{m}$	< 2.5 μm	$\geq 2.5 \mu\text{m}$
case 1	$A_{i1}=0.85$	$A_{i2}=0.15$	$A_{o1}=0.00$	$A_{o2}=0.91$
case 2	$A_{i1}=0.95$	$A_{i2}=0.10$	$R_{o1}=0.13$	$R_{o2}=0.09$

The radiative energy emitted from the glass tube to the environment is calculated by

$$Q_{or} = \pi \{ (D_{o1} + D_{o2}) / 2 \} L A_{o2} F_{2.5\infty}(T_o) \quad (19)$$

$$T_o = (T_{o1} + T_{o2}) / 2$$

2.2 Calculation procedure

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

$$Q_{oL} = Q_{or} + Q_{ot} + Q_{oc} \quad (20)$$

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

$$Q_{iL} = Q_{ic} + Q_{ir} \quad (21)$$

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

$$Q_i = \frac{Q_{sm} A_{i1} (1 - R_{o1})}{1 - (1 - A_{i1}) R_{o1}} \cdot 0.96 \quad (22)$$

and

$$Q_{oa} = Q_s A_{o2} \cdot 0.04 \quad (23)$$

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

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

$$Q_i = Q_f + Q_{iL} \quad (24)$$

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

$$Q_{iL} = Q_{oL} - Q_{oa} \quad (25)$$

First, the inlet temperature T_{fi} of fluid is given. The temperature T_i 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_i and T_{o1} , Q_{iL} and Q_{oL} 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_i 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_{fm} , the collector efficiency

$$\eta = Q_f / Q_s \quad (26)$$

the exergy efficiency

$$\eta_e = \eta_c \cdot \eta \quad (27)$$

and the final conversion efficiency

$$\eta^* = \eta_m \cdot \eta \quad (28)$$

where η_c is the Carnot efficiency

$$\eta_c = 1 - (T_e / T_{fm}) \quad (29)$$

and η_m 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:

$$\eta_m = 1 - \left(\frac{T_L}{T_H} \right)^{1/2} (= 1 - \left(\frac{T_e}{T_{fm}} \right)^{1/2}) \quad (30)$$

This agreed well with the thermal efficiencies of practical power stations⁽⁹⁾. T_H and T_L 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, η_c of eq.(29) and η_m of eq.(30) may be replaced with

$$\eta_c = 1 - (T_e / T_{fi}) \quad (31)$$

and

$$\eta_m = 1 - (T_e / T_{fi})^{1/2} \quad (32)$$

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_{fm} , η , η_e and η^* are shown for different absorber tube diameters at a constant aperture ($A_p = 0.36m$), 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 η_e and η^* are obtained for the smaller diameter, i.e. the larger concentration ratio. Although the collector efficiency η has a similar asymptotic value at a very large mass flow rate region, the condition which gives the maximum η_e and η^* is found in the region of medium η . Two nearly similar maxima of η_e or η^* 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 ($A_{i1} = 0.85$, $A_{i2} = 0.15$) with the ideal case of Fig.3 ($A_{i1} = 0.95$, $A_{i2} = 0.1$). Generally, the maximum values of η_e and η^* for the latter case show an increase of 20~25%

Fig.2 The result of T_{fm} , η , η_e and η^* for different tube diameters at a constant aperture ($A_p=0.36$) in the case of $A_{i1}=0.85$, $A_{i2}=0.15$ and $T_{fi}=293.2$ K.

Fig.3 The result of T_{fm} , η , η_e and η^* for different tube diameters at a constant aperture ($A_p=0.36$) in the case of $A_{i1}=0.95$, $A_{i2}=0.10$ and $T_{fi}=293.2$ 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 ($A_{i1}=0.85$, $A_{i2}=0.15$) are shown in the following.

In Figs.4 and 5, the effect of the concentration ratio on the values of T_m , η_e and η^* 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

Fig.4 The effect of the concentration ratio on the values of T_{fm} , η_e and η^* at atmospheric pressure in the annulus in the case of $A_{i1}=0.85$, $A_{i2}=0.15$ and $T_{fi}=293.2$ K.

Fig.5 The effect of the concentration ratio on the values of T_{fm} , η_e and η^* at a pressure of free molecule conduction region in the case of $A_{i1}=0.85$, $A_{i2}=0.15$ and $T_{fi}=293.2$ K. (The meanings of the lines are the same as those in Fig.4)

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 T_{fm} , η_e and η^* in the case of $A_{11}=0.85$ and $A_{12}=0.15$.

Fig.7 The effect of the inlet temperature on the values of η_e and η^* in the case of $A_{11}=0.85$ and $A_{12}=0.15$ for the latent 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 η_e and η^* . In Fig.8, taking the inlet temperature of fluid as the abscissa, η_e and η^* 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 η_e and η^* 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 $\eta_{e\max}$ and $\eta^*_{\max}$ 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

Fig.8 The effect of the inlet temperature on the values of η_e and η^* in the case of $A_{11}=0.85$ and $A_{12}=0.15$ for the constant flow rate.

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_{fm} 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 μ 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 1.013×10^6 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 ± 1 degree 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 practi-

Fig.9 The maximum values of η_e and η^* , and the value of T_{fm} for small D_{12} .

Fig.10 The maximum values of η_e and η^* , 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)

cal 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 η at the largest flow region is due to the fact that even a small experimental error in temperature measurement causes a large error in η 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.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 μ m emery paper), c(the surface coated with black paint), e(the selective absorber surface), g(glass tube)

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

version efficiency at a comparatively small concentration ratio promptly approach the asymptotic values for the case of a very large concentration ratio.

References

1. J.R.Howell and R.B.Bannerot, "Optimum solar collector operation for maximizing cycle work output", Solar Energy, Vol.19, 1977, p.149.
2. For example, H.C.Hottel et al., "The performance of flat plate solar-heat collectors", Trans. ASME, Vol.64, 1942, p.91.
3. Y.Katto, Introduction to Heat Transfer, Yokendo, Tokyo, 1976, p.90.
4. E.N.Sieder and G.E.Tate, "Heat transfer and pressure drop of liquids in tubes", Ind. Eng. Chem., Vol.28, 1936, p.1429.
5. T.H.Kuehn and R.J.Goldstein, "Correlating equations for natural convection heat transfer between horizontal circular cylinders", Int. J. Heat and Mass Transfer, Vol.19, 1976, p.1127.
6. S.Dushman, Scientific Foundations of Vacuum Technique, John Wiley & Sons, New York, 1962, p.48.
7. T.Igarashi, M.Hirata, "Study on heat transfer in detached flow", Trans. JSME, Vol.39, No.332, 1973, p.1890.
8. F.L.Curzon and B.Ahlborn, "Efficiency of a Carnot Engine at Maximum Power Outlet", AJP, Vol.43, 1975, p.22.
9. T.Saito, "Thermodynamic problems on the utilization of energy", Kikai no Kenkyu, Yokendo, Vol.30, 1978, p.1021.
10. J.A.Duffie, W.A.Beckman, Solar Energy Thermal Process, John Wiley & Sons, New York, 1974.

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 con-