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Estimation of Heat Losses from the Receivers for Solar Energy Collecting System of Korea Institute of Energy Research

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Heat losses from the receivers for a dish-type solar energy collecting system constructed at Korea Institute of Energy Research are analyzed. The Stine and McDonald's model is used to estimate the convection loss. The Net Radiation method and the Monte-Carlo method are used to calculate the radiation heat transfer rate from the inside surface of the receiver to the surroundings. Two different receivers are suggested here and the performances of the receivers are estimated and compared with each other based on the prediction of the amount of heat losses from the receivers. The effects of the receiver shape and the radiation properties of the surface on the thermal performance are investigated. The performance of Receiver Ⅰ is better than that of Receiver Ⅱ, and the amount of solar irradiation that is not captured by the receiver after being reflected by the concentrator becomes significant if the temperature of the working fluid is low.

Key Words : Parabolic Dish Collectors, Receiver Design, Heat Losses, Monte-Carlo Method

Nomenclature

- A_o : Outside surface area of a receiver (m^2)
- A_w : Heat transfer area inside a receiver (m^2)
- d : Aperture diameter of cavity (m)
- E_{bj} : Blackbody emissive power of zone j (W/m^2)
- F_{i-j} : View factor from cavity zone i to cavity zone j
- G_{bn} : Solar beam irradiance received for normal incidence on the reflector (W/m^2)
- Gr_L : Grashof number based on length L
- h : Convective heat transfer coefficient ($W/m^2 \cdot K$)
- I : Solar beam intensity ($W/m^2 \cdot sr$)
- k_i : Thermal conductivity ($W/m \cdot K$)
- L : Average internal dimension of cavity (m)
- l_i : Average thickness of the insulator of a receiver (m)
- Nu_L : Nusselt number based on length L
- T_a : Ambient temperature (K)
- T_w : Surface temperature inside a receiver (K)
- δ : Skewness angle
- ε_j : Surface radiation emissivity of zone j
- θ : Angle of concentrator/cavity axis with the horizontal surface (rad)
- ρ_r : Average reflectivity of the reflector
- σ : Scattering parameter (rad)
- Ω : Solid angle (sr)

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1. Introduction

A dish-type solar energy collecting system has been developed at Korea Institute of Energy Research (KIER) for the first time in Korea. The overview of the system is shown in Fig. 1. This system consists of fifteen parabolically concave circular mirrors that are mounted on the parabolic structure. The shapes of the mirrors are the exactly same and the diameters and the focal

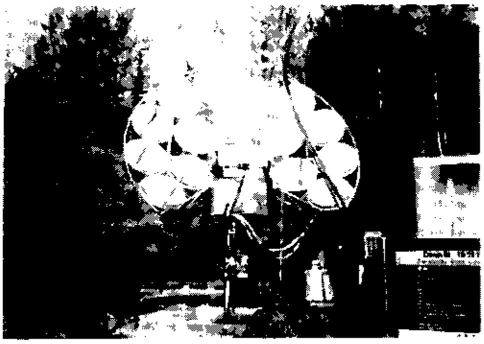


Fig. 1 Overview of the system considered.

lengths of these are 1 m and 3 m, respectively. Two different types of receivers are suggested as the first design for the systems as shown in Figs. 2 and 3. The outside dimensions are basically the same, but the inside dimensions of the receivers, which are important from the heat transfer point of view, are different.

In order to obtain the optimal design of a receiver, heat losses from a receiver should be thoroughly analyzed. Since convection and radiation losses are normally significant compared to conduction loss, convection and radiation heat transfer from a receiver to the surroundings are carefully investigated. There are several empirical correlations for estimating convection losses for different shapes of cavity-type receivers. LeQuere et al. (1981) examined the natural convection losses from two different sized cubical cavities which were similar in shape but different in size. Clausing (1981, 1983) described an analytical model for estimating convection losses from the open cubical cavity receiver. Koenig and Marvin (1981) gave an empirically-derived correlation for convection loss from the cylindrical cavity type receivers, including the effects of operating temperature and angle. An estimation of convection losses from a central cylindrical cavity receiver was also performed by Siebers and Kraabel (1984). Improving correlations proposed by Siebers and Kraabel, Stine and McDonald (1989) have suggested an empirical correlation, which has included the effects of size of a receiver aperture and the receiver tilt angle. However, the estimations by these correlations are not consistent so that we have to be careful when we choose one of these to apply to a particular receiver.

On the other hand, it is difficult to find a simple method for predicting radiation heat transfer from a receiver to the surroundings. Since the major concerns for most of the studies were the convection losses, the radiation losses were approximately predicted by a simple equation in those researches. If the operating temperature is low, however, the convection losses are relatively low compared with those for the receivers working at high temperatures and the radiation losses become significant instead. Especially, the solar energy concentrator of the KIER system is somewhat different from others. The dish-type solar energy collecting systems usually have a big single dish as a concentrator while the KIER system has fifteen circular mirrors instead of it. Hence, the characteristics of solar irradiation entering the receivers become different so that it is difficult to use the design information that has been reported for other systems.

Therefore, three modes of heat losses from the receivers suggested in the present study are estimated and the thermal performances of the receivers are compared to find an optimal design for the KIER system. Convection loss is predicted using the empirical correlation by Stine and McDonald (1989), and radiation heat loss is calculated by the Net Radiation (Modest, 1993; Dehghan and Behnia, 1996) and the Monte-Carlo methods (Yang et al., 1995). Based on the results, each mode of heat loss and the thermal performance of two receivers are compared at several different operating temperatures. In addition, the effects of absorptivity of the inner surface and the shape of the receivers are investigated.

2. System Geometries

Two different receivers are suggested for the system and each of them is shown in Figs. 2 and 3, respectively. Receiver Ⅰ has a conical shape and Receiver Ⅱ is the combination of a half circle and a short cylinder. The outside diameters and the heights of the receivers are 230 mm and 450 mm, respectively, and the aperture radii are 180 mm. The inner surface areas of Receiver Ⅰ and

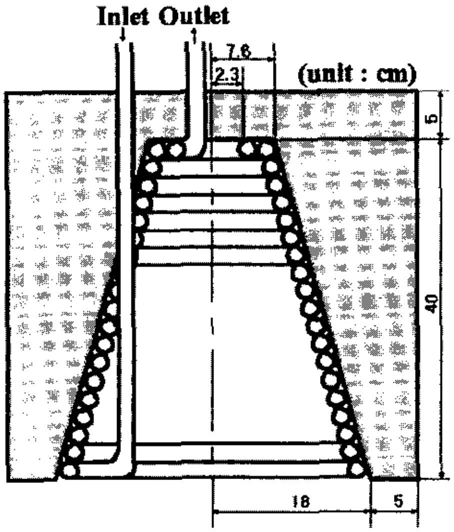


Fig. 2 Sketch of Receiver I

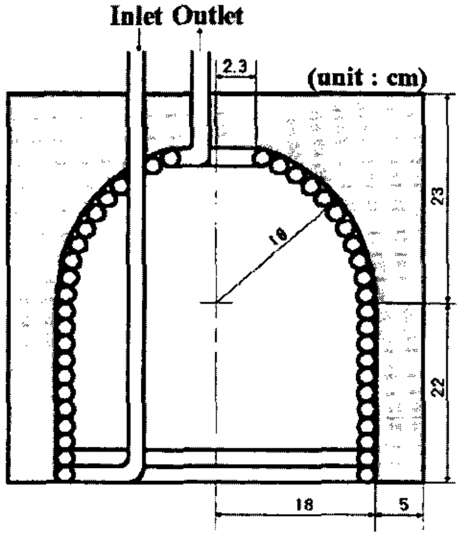


Fig. 3 Sketch of Receiver II

II are 0.35 m^2 and 0.45 m^2, respectively. Therefore, the inner surface area of Receiver II is about 25% larger than that of Receiver I. As can be seen in the figures, the inlet of the working fluid is located at the bottom of the receiver while the exit is at the top. The minimum thickness of insulation for both receivers is 50 mm. Insulation of Receiver I looks much thicker than that of Receiver II except for the aperture area so that it is easy to expect that the conduction loss from Receiver I is smaller than that from Receiver II. However, it has been known that the conduction heat transfer from a receiver is relatively small compared to other modes of heat transfer and the effects of insulation thickness on thermal performances of a receiver is not significant.

3. Estimation of Heat Losses

In order to evaluate the conductive heat transfer rate from the receivers, the following equation (Kaushika, 1993) is used.

$$Q_{cond}=\frac{1}{\frac{1}{A_o h}+\frac{l_i}{k_i\sqrt{A_o A_w}}}(T_w-T_a) \quad (1)$$

In this equation, the average convective heat transfer coefficient on the external surface of a receiver is needed to calculate the conduction loss. The empirical correlation for external flow around a cylinder proposed by Hilpert (1933) is used in this study. The shadow of a receiver on a concentrator decreases the amount of solar irradiation entering the receiver so that it is not always preferable to increase the insulation thickness of the receiver. Hence, the optimal thickness for the insulation wall exists for each system generally. However, we do not need to consider the shading effect for this study because there is no reflector at the center of the parabolic structure of the KIER system.

The convective heat transfer from an aperture of a receiver is difficult to be analyzed because there are many factors around a receiver affecting convective heat transfer. Convection loss varies with receiver angle, shape of the receiver, speed and direction of wind, working temperature, etc. Several empirical correlations have been suggested by several research groups. Each of these is developed for the particular shapes and operating conditions of a receiver so that it is difficult to find the correlation that can be generally applicable to many different types of receivers with reasonable accuracy. Therefore, the appropriate correlation for a particular receiver should be carefully selected in order to predict the convection loss accurately. Otherwise, painstaking experiments should be required.

In order to select the best model for the KIER

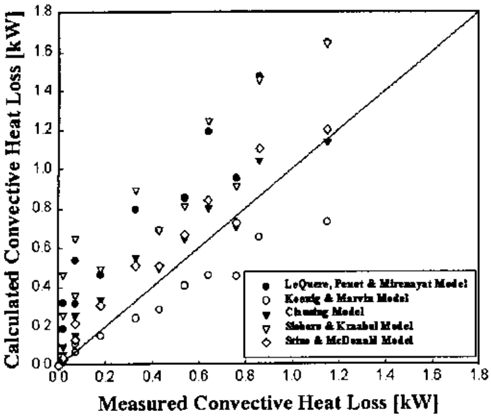


Fig. 4 Comparisons of measured and calculated convection losses from the STEP receiver using different empirical correlations.

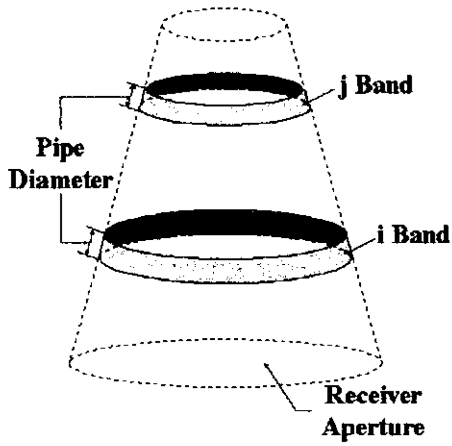


Fig. 5 Conceptual diagram for the Net Radiation Method

receivers, several models are applied to the receiver developed for the Solar Total Energy Project (STEP) (McDonald, 1995), which is very similar to that for the KIER system in its shape and the working temperature among other receivers existing in the world. The comparisons between the empirical values at working temperatures of 149℃ and 204℃ and the calculated values from each model are shown in Fig. 4. From the figure, it is clear that the Clausing's model and the Stine and McDonald's model can predict the convection loss from the STEP receiver accurately compared to other models. Here, the Stine and Mocdonald's model is used due to its explicit simplicity.

Improving the correlation by Siebers and Kraabel (1984), Stine and McDonald have suggested the following empirical correlation, which has included the effects of the size of a receiver aperture and the receiver angle.

$$Nu_L = 0.088\,Gr_L^{1/3}\left[\frac{T_w}{T_a}\right]^{0.18}(\cos\theta)^{2.47}\left[\frac{d}{L}\right]^{s} \quad (2)$$

where s is equal to $1.12-0.98\times d/L$. From this equation, the average convective heat transfer coefficient on the inner surface of a receiver can be obtained and the total losses by convection heat transfer can be calculated using the Newton's cooling law.

Radiation losses from a receiver can be classified into two categories. One is surface emission from the inner surface of the receiver to the surroundings, which is called emission loss. The other is reflected solar irradiation by the inner surface that escapes from the receiver. In other words, solar irradiation is reflected by the concentrator first. After being focused at the receiver, solar irradiation hits the inner surface of the receiver. Although it depends on the radiation property of the inner surface, most of the energy is absorbed by the surface. However, a certain amount of solar energy that enters the receiver escapes from the cavity by surface reflecting, unfortunately. This is called reflection loss in the present study. The Net Radiation (Modest, 1993; Dehghan and Behnia, 1996) and the Monte-Carlo Method (Yang et al., 1995) are used to evaluate the emission and reflection losses, respectively.

For the Net Radiation Analysis, the inner surface of the receiver is divided into a number of small bands as shown in Fig. 5. Assuming that the inner surface of a receiver is diffuse and gray, radiation heat exchange between each band can be expressed as the following equations from the energy balance on each surface of the bands.

$$\sum_{j=1}^{n}\left[\frac{\delta_{ij}}{\varepsilon_j}-\left(\frac{1}{\varepsilon_j}-1\right)F_{i-j}\right]q_i=\sum_{j=1}^{n}[\delta_{ij}-F_{i-j}]E_{bj} \quad (3)$$

Therefore, if the temperature, the emissivity and the view factors are known for each band in a receiver, it is easy to calculate the radiation heat transfer rate in each band using Eq. (3). It is supposed that the temperature distribution inside a receiver has a linear variation from the inlet to the exit. The view factors between the bands are obtained from the ready-made formulas for similar geometries and the reciprocity of the view factor (Modest, 1993). The radiation loss from the cavity means the heat transfer rate through the artificial surface that coincides with the aperture plane.

In order to estimate the amount of escaping solar energy reflected by the inner surface, the statistical ray tracing method, which is called the Monte-Carlo method, is used for this study. For the analysis, it is assumed that the solar ray is specularly reflected on the surface of the mirror, and then it is diffusely reflected on the inner surface of the receiver. Otherwise, it is absorbed into the inner surface of the receiver. The radiation properties are assumed to be independent on the wave number. The parabolic structure holding the mirrors is assumed to face the sun all the time during the operation. Because the receiver is made of a long circular tube, the inner surface is wavy and the shape of each wave is a half circle. However, the inner surface is assumed to be flat. If we consider the real wavy surface, it is too complicated to trace the photon bundle. In order to begin the ray tracing procedure, an artificial photon bundle is numerically generated toward the parabolic structure. The direction of the photon bundle generated is parallel to that of solar ray coming from the sun to the collector directly. Diffuse radiation, which is not important from the heat transfer point of view, is not considered here. Whenever the photon bundle hits the surface, it is statistically determined whether it is absorbed or reflected using the radiation properties of the surface. If the total number of photon bundle absorbed into the inner surface of the receiver is N_{abs} among N of photon bundles generated, the value of radiation loss by the inner surface reflection of the receiver is calculated as follows.

$$Q_{rad}=Q_{in}\frac{N-N_{abs}}{N} \tag{4}$$

where Q_{in} represents the total solar irradiation entering the receiver and is known by estimating the heat flux distribution in a receiver aperture. For calculation, about 250,000 of photon bundles for each mirror are generated to obtain statistical consistency. The deviation between the solutions for this situation is at most less than 0.5%.

Since the shape of the sun is close to a disk when seeing it from the earth, the sun subtends an angle of 0.533° and the solar ray has the shape of a cone. The intensity distribution of the cone ray is called sunshape and can be expressed as Gaussian function.

$$f=\frac{I}{G_{bn}}=\frac{R}{2\pi\sigma^2}\exp\left[-\frac{\delta^2}{2\sigma^2}\right] \text{ when } \delta\leq n\sigma \tag{5}$$
$$=0 \qquad \text{ when } \delta>n\sigma$$

where R is equal to $\rho_r/(1-exp(-n^2/2))$. Integrating Eq. (5) with respect to solid angle, the concentration ratio at any point can be found as follows (Jeter, 1986).

$$C_r=\frac{q}{G_{bn}}=\iint f\ \cos\phi\ d\Omega \tag{6}$$

Then, the heat flux distribution is obtained by calculating it with respect to proper points in a focal plane.

4. Results and Discussion

For the present study, it is assumed that the normal incident irradiation is 800 W/m², the velocity of wind is 3.5m/s, the thermal conductivity of the insulation material is 0.046W/m · K, and the ambient temperature is 25℃. In addition, the working temperature, which means the average of the inlet and outlet temperatures, ranges from 100 to 200℃. The inlet temperature is assumed to be equal to the ambient temperature, which is 25℃. This means that the exit temperature becomes 375℃ at most if the working temperature is 200℃. The reason why these operating conditions are selected here is that thermal fluids, which are commonly used as working fluids, usually change their phases over 400℃.

Figure 6 shows the distribution of the concentration ratio on a receiver aperture plane that is calculated with Eq. (6). The value of σ is 0.267°, which is calculated from the size of the sun and

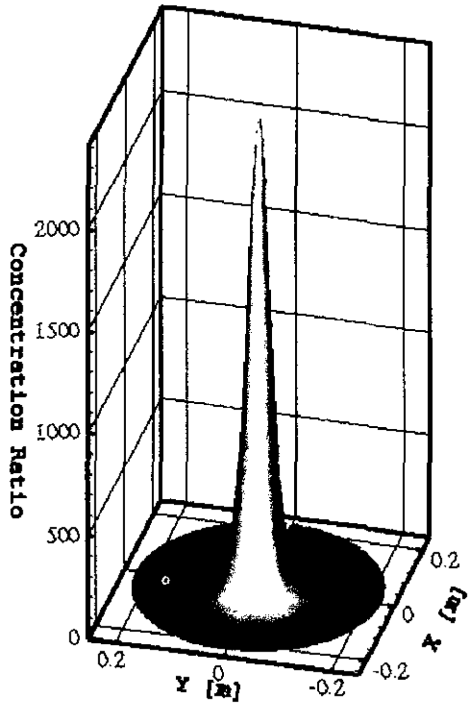
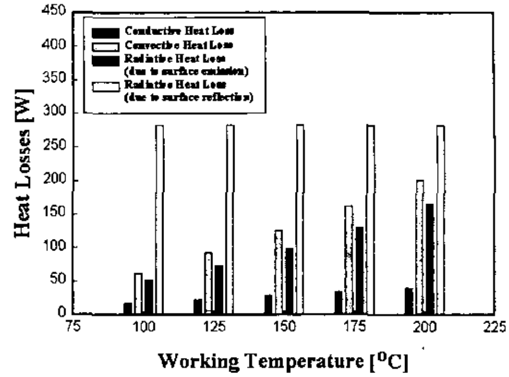
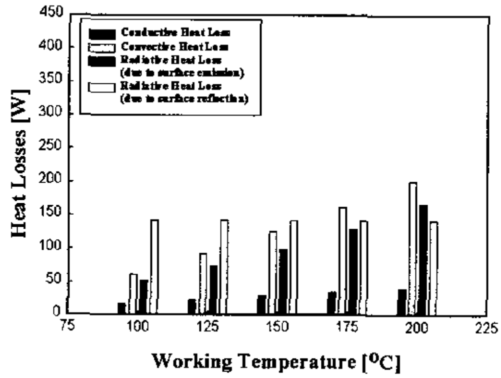


Fig. 6 Distribution of the local concentration ratio on the focal plane for a solar half angle of 0.267°



(a) Absorptivity of the inner surface=0.85

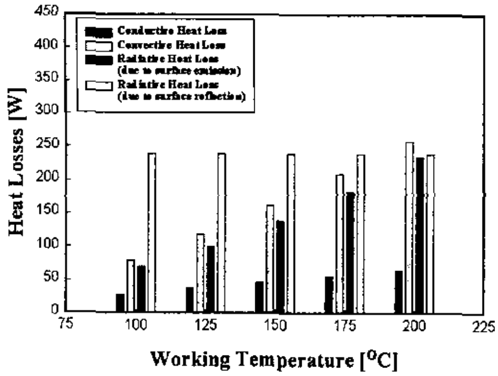


(b) Absorptivity of the inner surface=0.92

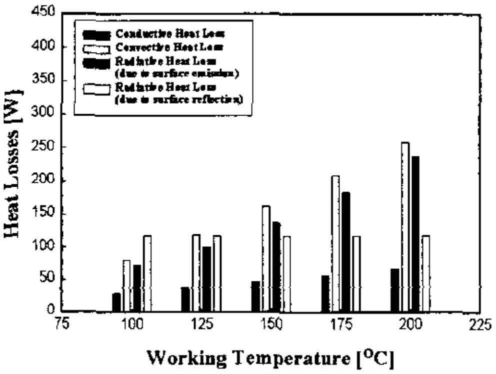
Fig. 7 Comparisons of heat losses from Receiver Ⅰ

the average distance between the earth and the sun. The highest concentration ratio of the system is lower than those in other systems that have a single reflector, generally. It is because fifteen separate reflectors cannot focus solar irradiation effectively due to the structure of the system. However, it can be found that most solar irradiation reflected by the mirrors enters a receiver aperture of 18 cm in radius and its amount is approximately 7.7 kW when the reflectivity of the mirrors is 0.85. In addition, the apparent reflectivity of the receivers toward the surroundings can be calculated by the Monte-Carlo Method. When the absorptivity of the inner surface of the receivers is 0.85, the apparent reflectivities of Receiver Ⅰ and Ⅱ converge to 0.0366, 0.0311, respectively. It is because the inner surface area of Receiver Ⅱ is greater than that of Receiver Ⅰ. However, the difference between two reflectivities is not significant at this situation. Since convection loss increases with the inner surface area, it is not always preferable to increase the inner surface area in order to increase the apparent absorptivity of a receiver.

Heat losses from Receiver Ⅰ and Ⅱ versus the working temperature are given in Figs. 7 and 8. As shown in the figures, the conduction loss is not significant compared to convection and radiation losses. When the absorptivities of the inner surfaces are 0.85 and 0.92, each mode of heat losses from Receiver Ⅰ is presented in Figs. 7(a) and (b), respectively. If the absorptivity of the inner surface of the receivers is 0.85, total loss of Receiver I increases from 0.41 kW to 0.69 kW as the working temperature increases from 100 to 200℃. Conduction, convection, and emission losses increase with the working temperature while reflection loss remains constant. If the absorptivity of the inner surface of the receiver is 0.92, total loss decreases to 0.27 and 0.55 kW at the working temperature of 100 and 200℃, respectively. Reflection loss decreases by about 50%



(a) Absorptivity of the inner surface=0.85



(b) Absorptivity of the inner surface=0.92

Fig. 8 Comparisons of heat losses from Receiver II

while other modes of heat loss do not change significantly. This means that the absorptivity of the inner surface of a receiver is one of the most important design parameters to increase the thermal performance of the receiver. Generally, radiation loss due to surface reflection is neglected because convection and emission losses are dominant if the working temperature is high. However, it cannot be negligible any more if the working temperature is relatively low, and it must be taken into account for accurate estimation.

Heat losses from Receiver II for the inner surface absorptivities of 0.85 and 0.92 are presented in Figs. 8(a) and 8(b), respectively. When the absorptivity of the inner surface of the receiver is 0.85, total loss ranges from 0.42 kW to 0.80 kW when the working temperature varies from 100 to 200℃. The difference between total losses from Receiver I and Receiver II is not significant

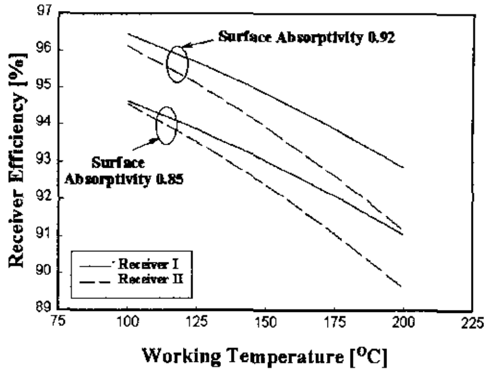


Fig. 9 Comparison of the thermal efficiencies of Receiver I and II

when the working temperature is 100℃. However, it increases with the working temperature. For the working temperature of 100℃, reflection loss for Receiver II is less than that for Receiver I, while convection and emission losses for Receiver II are greater than those for Receiver I. As the working temperature increases, the differences between convective losses and emission losses for Receiver I and II increase while reflection loss does not change. Therefore, the difference of total loss increases with the working temperature. If the absorptivity of the inner surface of Receiver II is 0.92, total loss decreases to 0.30 kW and 0.68 kW for the working temperature of 100 and 200℃. As mentioned for Receiver I, it is because of decrease of radiation loss due to surface reflection. From the results, it can be concluded that if the inner surface area increases convective and emission loss increase while reflection loss decreases. If the working temperature is high, convection and emission losses become dominant.

The thermal efficiencies of the receivers are shown in Fig. 9 as the summary of calculations and it is found that the efficiencies of Receiver II are lower than those of Receiver I. It is basically because the inner surface area of Receiver I is about 25% smaller than that of Receiver II with the same aperture size. The best efficiency is obtained with Receiver I when the absorptivity of the inner surface is 0.92. The efficiencies of Receiver II for both absorptivities of 0.85 and 0.92 decrease more rapidly with the working temperature than those of Receiver I do. The maxi-

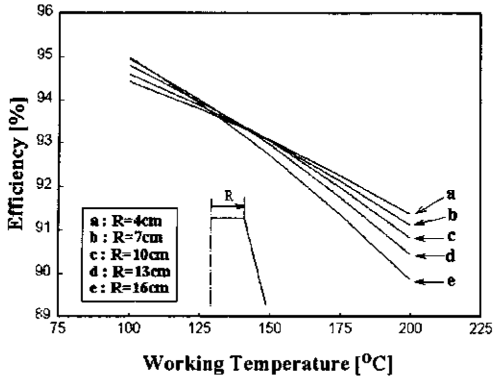


Fig. 10 The effect of the shape of a receiver on thermal efficiency

mum difference between the efficiencies for Receiver Ⅰ and Ⅱ for the present study is 1.5% at the working temperature of 200℃.

In Fig. 10, the effect of the receiver shape on the thermal efficiency is graphically shown. Fixing the aperture size, the radius of the top end is changed. The thermal efficiencies from 100℃ to 150℃ of the working temperature are about the same. On the other hand, the differences increase with the working temperature. When the radius of the top end is 4 cm, the decreasing rate of the efficiency versus the working temperature is the lowest. Hence, we can obtain the best thermal performance from this geometry if the working temperature is 200℃.

5. Conclusions

The estimation of heat losses from the receivers suggested for the dish-type solar energy collecting system at Korea Institute of Energy Reseach has been accomplished. Since the working temperature of the system is low and the system has fifteen mirrors as a concentrator unlike other common systems, several interesting results are obtained as follows:

(1) Generally, the thermal performance of Receiver Ⅰ is better than that of Receiver Ⅱ for this operating conditions. The thermal efficiencies decrease with increasing the working temperature.

(2) The thermal performance is very sensitive to the absorptivity of the inner surface of a receiver. Reflection loss decreases by 50% as the absorptivity changes from 0.85 to 0.92. On the other hand, emission loss increases slightly with the absorptivity.

(3) If the working temperature is high, reflection loss is negligible compared with other modes of heat losses. However, reflection loss should be taken into account for the optimal design of a receiver if the working temperature is low. It is because convection and emission losses decrease significantly and have the almost same order of magnitude with reflection loss.

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