On the Study of Convection Loss from Open Cavity Receivers in Solar Paraboloidal Dish Applications

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Abstract

This study undertakes the numerical investigation of natural convection loss from cavity receivers employed in solar paraboloidal dishes. Three different receiver geometries have been considered. One of these is the experimental model receiver for validating the numerical results. The other two are essentially the ones currently used in ANU 20 m^2 and 400 m^2 dishes. According to the preceding work (Paitoonsurikarn & Lovegrove, 2002), a good agreement between experimental and numerical results of model receiver was obtained. Furthermore, the numerical results of all receivers were qualitatively comparable to the predictions by other available correlations hitherto. In the present paper, a new simplified correlation is proposed.

The combined free-forced convection study, i.e. that includes the effect of wind speed and direction on convection loss, has also been undertaken. However, at this stage, the study is limited only in the case of wind parallel and normal to the aperture plane. Some of the recently obtained results are presented.

INTRODUCTION

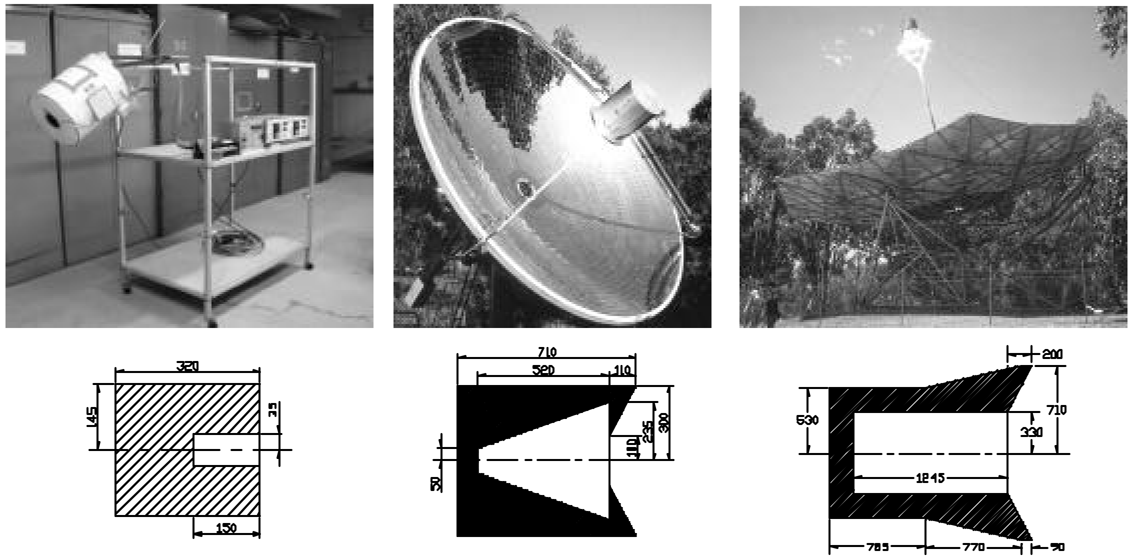
In order to assess and subsequently improve the thermal performance of open-cavity receivers employed in paraboloidal dish solar concentrators (e.g. Lovegrove *et al*, 2003), its associated heat losses need to be determined with sufficient accuracy. In general, the conduction and radiation modes of heat losses can be determined relatively easily by the standard methods described in the literature. On the other hand, the determination of convection loss is more complex.

The present study undertakes the further investigation of natural convection loss from open-cavity receivers following from the preceding work (Paitoonsurikarn & Lovegrove, 2002). At this stage, the natural convection loss from three different geometries has been considered. One of these is an experimental model receiver for validating the numerical results. The other two are the ones currently used in ANU 20 m^2 and 400 m^2 dishes. Some additional parameters to the convective loss, such as aperture to cavity diameters ratio are taken into account in this study.

In addition, a study of combined free-forced convection, that includes the effect of wind speed and direction on convection loss, has also been undertaken. However, at this stage, the study is limited only to the cases of wind parallel and normal to the aperture plane. Some of the recently obtained results are also presented.

NUMERICAL PROCEDURE

Most of the detail for numerical procedure is similar to that described in the previous paper. In the present work, the CFD Software Package, Fluent 6.0 (Fluent Inc., 2002) is employed in the 3D simulation of both natural and combined convections through the aperture of the cavity receiver.



(a) Model Receiver (b) 20 m^2 Dish Receiver (c) 400 m^2 Dish Receiver

Figure 1 Three Receiver Systems and Cross Sectional Diagram of All Three Receiver Models.

There are three receiver geometries considered as illustrated in Figure 1 together with their actual arrangements. Figure 1 a) shows the electrically-heated model receiver used in the counterpart experimental work by (Taumoefolau & Lovegrove, 2002), whilst the other two are the solar receivers currently employed in the 20 m^2 and 400 m^2 dishes at ANU. Problem formulation and boundary condition used for each type of convection study are described below.

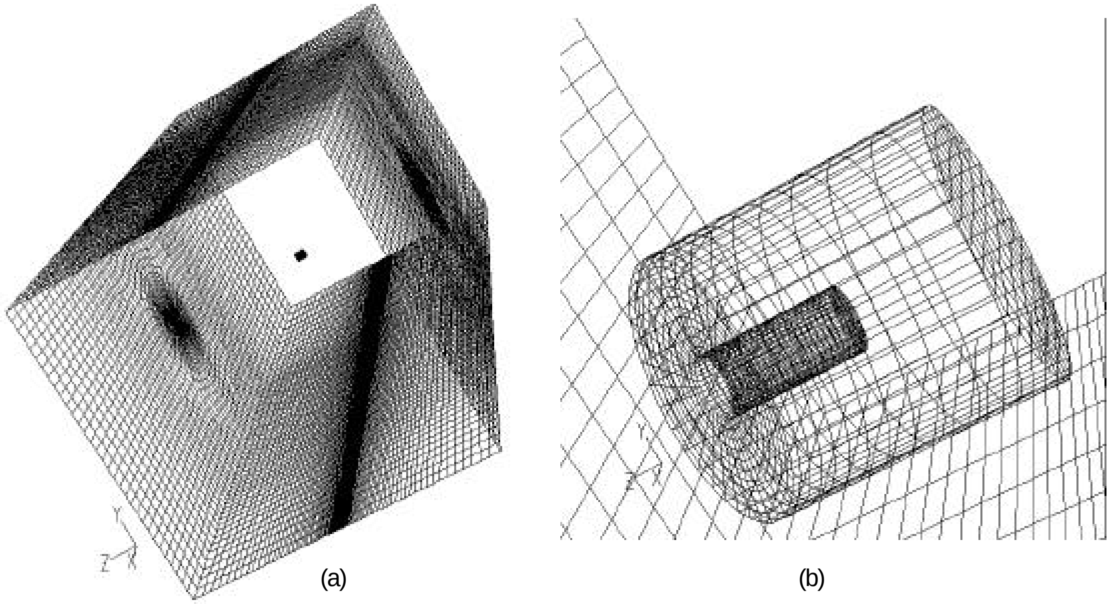
Natural Convection

In the CFD analysis, the receiver is assumed to be placed in a sufficiently large enclosure with walls at ambient temperature. Due to the symmetrical flow geometry with respect to the middle vertical plane, the computational extent comprises only one half of the physical domain. The size of the enclosure was determined in a preliminary study such that it showed negligible effect on fluid and heat flows in the vicinity of the receiver. It was found that the diameter and height of the enclosure should be approximately twenty times the diameter of the receiver to achieve this. More detail on the approach used can be found in the previous paper (Paitoonsurikarn & Lovegrove, 2002).

For boundary conditions, the cylindrical enclosure wall was set to ambient temperature of 27^{o}C. The receiver's cavity and outer walls were assumed to be isothermal and adiabatic, respectively. The cavity wall temperature for each receiver was set as follows:

- For the model receiver, the average experimental values of cavity wall temperature data of 445^{o}C and 408^{o}C were used for the cylindrical section and circular end plate, respectively.
- For the 20 m^2 dish receiver, the typical experimental data of cavity wall temperature data were used, which were 637^{o}C, 655^{o}C and 576^{o}C for the annular section surrounding the aperture, frustum section and circular back section, respectively.
- For the 400 m^2 dish receiver, the temperature of the whole cavity wall was arbitrarily set at 627^{o}C.

The main parameter of interest in the previous work was the cavity inclination which was simulated by directing the gravity vector to the desired direction. The effect of reduced aperture area for model receiver has now also been investigated by introducing an annulus obstacle of zero thickness at the aperture plane. Aperture diameters of 70.0, 59.5, 50.0, 42.0, and 35.0 mm were considered in the present study corresponding to the ratio of aperture to cavity diameters of 1.00, 0.85, 0.71, 0.6 and 0.5, respectively.



(a) (b)

Figure 2 Typical computational grid for combined convection study of model receiver: (a) Entire domain and (b) Close-up view of receiver.

Combined Convection

Two cases of combined convection, i.e. wind parallel and normal to the aperture plane, have been considered. A typical flow configuration for the former is depicted in Figure 2. The receiver is modelled as if it is located in the very centre of a wind tunnel of square cross section. The airflow is in the negative x-direction and parallel to the aperture plane. The width and height of the wind tunnel is chosen to be about twenty times the diameter of the receiver in accordance with the size of the simulation region used for the natural convection case.

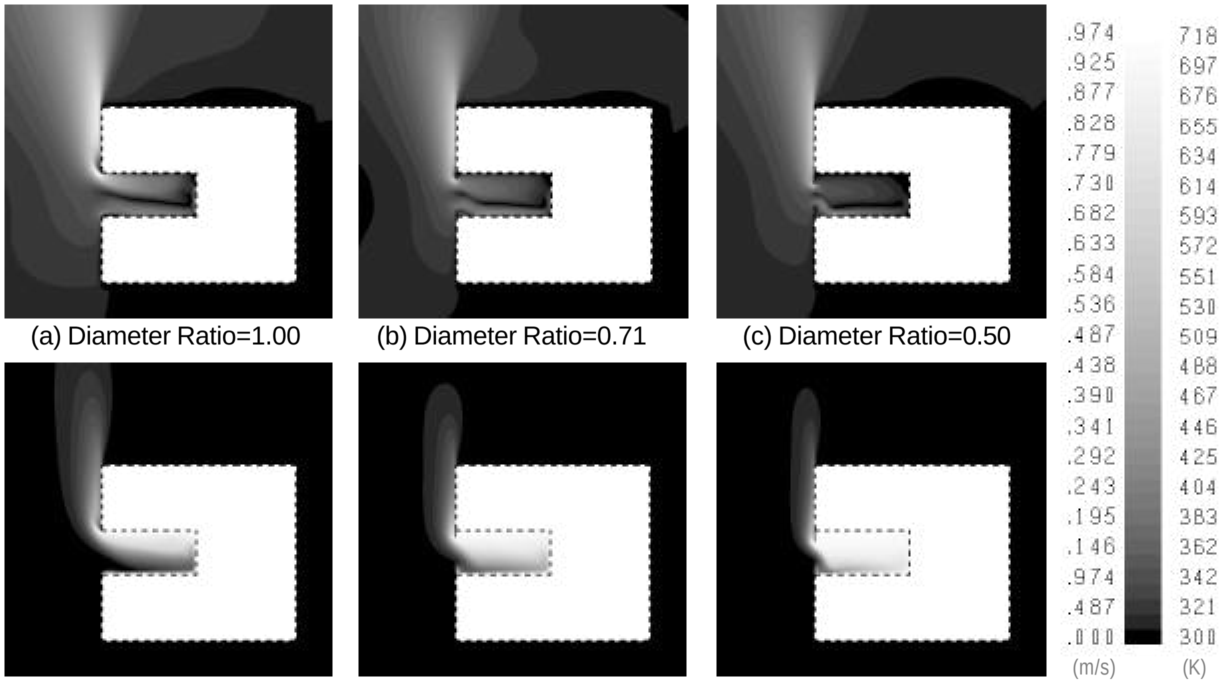
For boundary conditions, the temperatures of the incoming airflow and the tunnel wall were set at an ambient temperature of 27^{o}C. The temperatures of cavity and receiver outer walls for each receiver are the same as those used in the natural convection study.

Simulation with the wind speed up to 20 m/s has been carried out.

For both convection studies, the flow and heat transfer simulation is based on the simultaneous solution of the system of equations describing the conservation of mass, momentum, energy and turbulent transport property. The Spalart-Allmaras one-equation turbulent model recommended by Spalart (2000) is employed.

The steady-state governing equations are solved in Fluent using a coupled solver, which means that temperature and flow fields are coupled with each other and are solved simultaneously. All temperature-dependent properties of air are evaluated at local temperature by using the least-square fit equations derived from thermodynamic data compilations taken from Holman (1997).

RESULTS AND DISCUSSION



(d) Diameter Ratio=1.00 (e) Diameter Ratio=0.71 (f) Diameter Ratio=0.50

Figure 3 Contour Plot of Velocity Magnitude ((a)-(c)) and Temperature ((d)-(f)) at Symmetrical Plane for Model Receiver with Different Aperture to Cavity Diameters Ratio.

Natural Convection

The effect of aperture to cavity diameters ratio is illustrated in Figure 3 for a case of model receiver in the horizontal position. The contours of velocity magnitude in Figures 3 a)-c) clearly show that the obstacle at the aperture does attenuate the amount of air convection in and out of the cavity. Also, the temperature contours in Figures 3 d)-f) indicate that larger amount of hot air is trapped in the cavity as the diameter ratio is reduced. It is found that heat loss is not exactly proportional to the reduced aperture area, which can be realized by the fact that it could be approximated from a product of exiting hot plume's occupying area and velocity. While the former decreases conformably with decreasing aperture area, the latter, which is found to be 0.94, 0.68, and 0.64 m/s for the corresponding diameter ratio of 1.00, 0.71, and 0.50, decreases about 28% for the first decrement but only 4% for the second one. Hence, the reduction of heat loss will be slightly slower than the reduction of aperture area. This argument has been confirmed by experimental evidence (Taumoefolau, 2003).

As discussed in the previous work, the correlations proposed so far by many researchers, i.e. (Clausing, 1981), (Clausing *et al*, 1987), (Koenig & Marvin, 1981), (Stine & McDonald, 1989), and (Leibfried & Ortjohann, 1995), do not have a universal range of applicability and also sometimes show large differences in heat loss predictions for the similar receiver geometry. To address these shortcomings a new correlation has been developed.

Correlations for prediction of heat transfer coefficients are expressed in terms of Nusselt number (Nu), a non-dimensionalised heat transfer coefficient defined by:

$$Nu = \frac{hL_S}{k}, \quad (1)$$

where h is the convective heat transfer coefficient, k is the conductivity of the fluid, and L_s is the characteristic length of the flow situation.

Natural convection situations show a strong dependence on the Rayleigh number (Ra) which is a non dimensional ratio of buoyancy and diffusion effects defined by:

$$Ra = \frac{g\beta?TL_s^3}{\upsilon\alpha}, \tag{2}$$

where g is the gravitational constant, ΔT is the relevant temperature difference, β, υ,and α are the thermal expansion coefficient, kinematic viscosity, and thermal diffusivity of the fluid respectively.
Commonly used correlations for predicting the heat transfer from isothermal vertical flat plates are, for laminar free convection (Eckert, 1950):

$$Nu=0.508\left(\frac{Pr}{0.952+Pr}\right)^{1/4} Ra^{1/4}, \tag{3}$$

and turbulent free convection (Churchill & Chu, 1975):

$$Nu=0.15\left(\frac{Pr^{9/16}}{0.671+Pr^{9/16}}\right)^{16/27} Ra^{1/3}, \tag{4}$$

where Pr is the dimensionless group representing the ratio of momentum and thermal diffusivities of fluid and defined by:

$$Pr = \frac{\upsilon}{\alpha}. \tag{5}$$

Both suggest that the correlation of the form $Nu=CRa^n f(Pr)$ is appropriate to the system of free convection flow of interest. This is also confirmed by the correlation developed by McAdams (1954) for predicting the free convection from upper surface of heated horizontal flat plate:

$$Nu=0.54Ra^{1/4}. \tag{6}$$

Because the cavity can be considered as the combination of vertical and horizontal surfaces of various orientations, the correlation to predict the free convection loss from the cavity should have the similar form as these. Based on this idea, an improved correlation has been developed. The major feature has been to adopt a characteristic length scale of the cavity that is a strong function of angle of inclination. The assumption made was that the true characteristic length L_s should depend on average cavity diameter D_{cav}, aperture diameter D_{ap}, cavity length L, and sine and cosine of the inclination angle ϕ. Fitting to all the CFD simulation results, has yielded the following simple result:

$$Nu=0.00324Ra^{0.447}, \tag{7}$$

with

$$L_s = \left(4.79\cos^{4.43}(\phi)-0.37\sin^{0.719}(\phi)\right)D_{cav} + \left(1.06\cos^{3.24}(\phi)-0.0462\sin^{0.286}(\phi)\right)D_{ap}$$

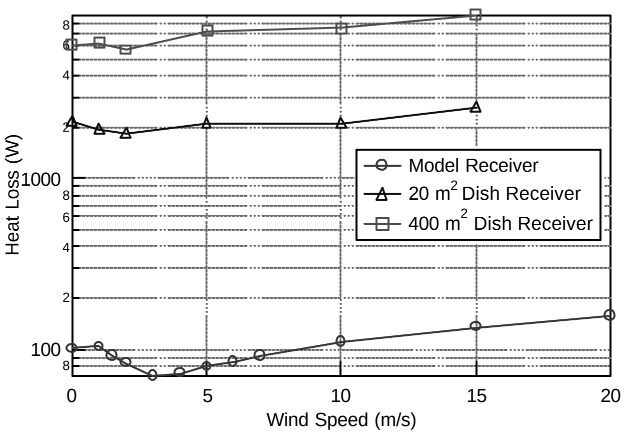


Figure 5 The Relationship between Wind Speed and Heat Loss for All Three Receivers with Wind Parallel to the Aperture Plane.

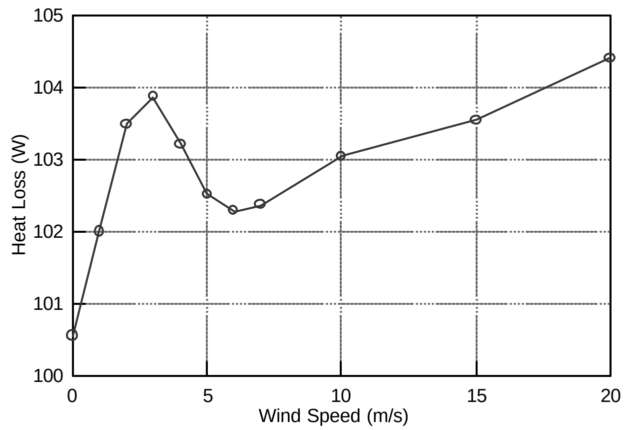


Figure 6 The Relationship between Wind Speed and Heat Loss for Model Receiver with Wind Normal to the Aperture Plane.

$$+\left(7.07\cos^{5.31}(\phi)+0.221\sin^{2.43}(\phi)\right)L\ . \qquad (8)$$

All properties are evaluated at the "film temperature", the average of wall and ambient temperatures. The prediction by present correlation together with the numerical data from all three receivers is presented in Figure 4. However, a prediction error of up to about 50 % is found. Further refinement of the correlation will be carried out when more numerical data are obtained.

Combined Convection

Figures 5 and 6 summarize the most recent results of the combined convection study. In Figure 5, the relationship of wind speed and convection heat loss for all three receivers with wind parallel to the aperture plane is shown. Surprisingly, it is found that heat loss, is actually reduced below the natural convection value by wind speeds up to about 7 m/s. This is confirmed by the experimental observations of Taumoefolau (2003). According to Figure 5, the magnitude of wind speed that results in the minimum heat loss tends to decrease with increasing receiver dimension. The explanation must presumably be related to the fact that the low wind speeds are preventing or sweeping away thermal stratification in the air and so suppressing natural convection. At higher wind speeds, the wind will scour hot air from the cavity and so increase losses as expected.

Figure 6 shows the similar result for a case of model receiver with wind normal to the aperture plane. Initially, the sharp increase of heat loss is found up to its local maximum at wind speed of about 3 m/s. After that heat loss decreases to its local minimum at wind speed of about 6 m/s, where it starts to unboundedly increase with increasing wind speed. Modelling of the cases of two actual receivers

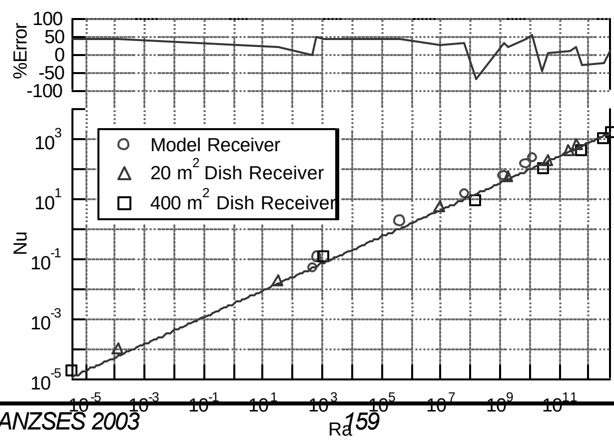


Figure 4 Comparison of Prediction by Present Correlation and Numerical Results of Three Receivers.

needs to be carried out to investigate the effect of size in this case. A possible explanation for this behaviour is that at very low wind speeds, the wind from the front must enhance natural convection in some manner. Above 3 m/s it must be suppressing natural convection and then for high speeds, once again increasing losses by scouring the cavity.

From the result of wind parallel to the aperture plane, though the relationship found qualitatively complies with the experiment by Taumoefolau (2003), there is a vast difference in magnitude between both results. To be more specific, experimentally-obtained heat loss increases more sharply with increasing wind speed after it passes the minimum value. This might be because in the experiment there is the considerable fluctuation in the wind current from the fan and so the direction of wind might not perfectly parallel to the aperture plane. The numerical simulation with wind direction slightly inclined into the cavity will be undertaken to test this hypothesis.

CONCLUSIONS

A correlation model for predicting the natural convection loss from open-cavity solar receivers has been developed. Initial indications are that by incorporating and angle dependant length scale, the correlation is more accurate and general than those previously published. CFD analysis is continuing to allow for further verification and refinement. In particular, the series simulation of receiver with same cavi ty geometry but varying aperture to cavity diameter ratio is a priority.

CFD analysis of combined free-forced convection has been carried out. The relationship between wind speed and heat loss in the case of the experimental model receiver with wind parallel and normal to the aperture plane has been examined. Interesting trends showing that some wind speeds actually reduce heat loss over the natural convection level have been found. This has also been found in associated experimental work, confirming that it is a real phenomenon. Large quantitative differences between the numerical and experimental results remain however. The modification of the numerical simulation to resolve this is under investigation.

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