A New Correlation for Predicting the Free Convection Loss from Solar Dish Concentrating Receivers

S. Paitoonsurikarn and K. Lovegrove
Centre for Sustainable Energy Systems,
Department of Engineering,
Australian National University,
Canberra ACT 0200,
AUSTRALIA
E-mail: sawat.paitoonsurikarn@anu.edu.au

Abstract

The study of free convection loss from an open-cavity receiver used in solar dish application has been undertaken by many researchers. Various correlations for free convection prediction have been proposed, e.g., by Stine & McDonald (1989), and Leibfried & Ortjohann (1995). Nonetheless, it was found that each correlation has a limited range of applicability, which is inherently based on the particular cavity geometry and experimental condition used in each of those work (Paitoonsurikarn & Lovegrove, 2002).

In this paper, a new correlation based on the numerical simulation results of three different cavity geometries is presented, one of which is a small-scale experimental model receiver, while the other two are the full-scale receivers currently used in the ANU 20 m^2 and 400 m^2 dishes at the ANU. The correlation is an improved version of one proposed in two previous ANZSES papers (Paitoonsurikarn & Lovegrove, 2003; Paitoonsurikarn et al, 2004), based on an ensemble cavity length scale. The correlation is derived from a larger database but yet further simplified in form.

Despite a wide variety of cavity geometries and operating conditions, the proposed correlation can predict approximately 50% of the data within ±20% and 90% of the data within ±50%. This is better than any of the other correlations published to date. The new correlation is also simpler to use than the most accurate of those previously published. Considering its prediction performance separately on each receiver, it was found that the correlation better correlates the data of two full-scale receivers. It correlates as high as 70% of the data within ±20%, and up to 100% coverage within ±50%. This further encourages its use in the design of actual receiver applications.

Keywords: Dish cavity receiver, Free convection, Correlation.

1. INTRODUCTION

An open-cavity type receiver is widely used in solar paraboloidal dish applications, eg. as the superheated stream generator in the ANU 400 m^2 "Big Dish" system for electrical power production (eg. Lovegrove *et al*, 2003). In these applications, the receiver is a key component in determining the overall system performance. This gives a high priority to an evaluation of its associated thermal losses, which are generally categorised into three modes, i.e. the conduction loss through the insulation wall, the radiation loss and the convection loss through the cavity aperture.

Conduction and radiation losses can be determined relatively easily by standardised methods given in a heat transfer literature. On the other hand, the determination of the convection loss is rather difficult, and heavily relies on a semi-empirical model because of its inherently complex nature.

Various correlations for free convection prediction have been proposed in the previous works, e.g., ones by Stine & McDonald (1989), and Leibfried & Ortjohann (1995). However, as suggested in Paitoonsurikarn & Lovegrove (2002) it was found that each correlation has a limited range of applicability, which is inherently based on a particular cavity geometry and operating conditions used in the experiment in each of those works.

Accordingly, in this study numerical simulations with the use of a computational fluid dynamic software package, Fluent 6.0 (Fluent Inc., 2001), have been undertaken to quantify the free convection loss from different cavity geometries ranging from small- to large-scale. The primary objective is to establish a new empirical correlation that is universally applicable. A correlation was first proposed in Paitoonsurikarn & Lovegrove (2003), which is based on 21 data points derived from numerical simulations of three different cavity geometries. It was developed on the concept of using an ensemble cavity length scale to take into account the effects of cavity geometry and inclination. Subsequently, the correlation was further modified to take into account the effect of different cavity wall temperatures and included 51 data points in Paitoonsurikarn *et al* (2004). In the same paper, another correlation based on the work of Leibfried & Ortjohann (1995) was also proposed.

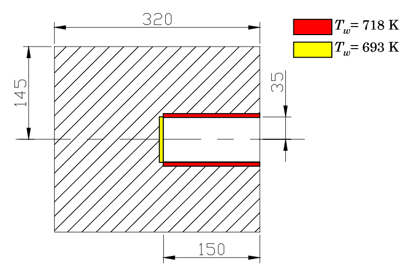
In this paper, as the continuation of the work described above, more numerical results were derived to cover effects of various parameters, e.g. cavity size, of all three cavity receivers, and subsequently used to refine the previously-proposed correlation.

Comparisons between the numerical result and the experiment are also presented in order to further prove the validity of the simulation method.

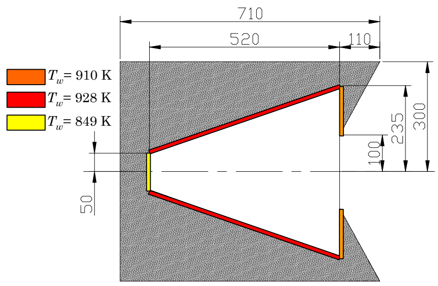
2. CAVITY GEOMETRIES

Four cavity geometries were considered as shown in Figure 1. Their relevant dimensions and cavity wall temperatures used in the base case simulation are indicated in the figure. Notably, the first three shown in Figures 1a-c are the same as those used in the correlation development in the previous papers, i.e. the experimental model receiver and two receivers currently used in the ANU 20 m^2 and 400 m^2 dishes at the Solar Thermal Facilities, the ANU. The other receiver shown in Figure 1d is the one used in the experimental work by McDonald (1995), which was included in the study for a further validation of the numerical results.

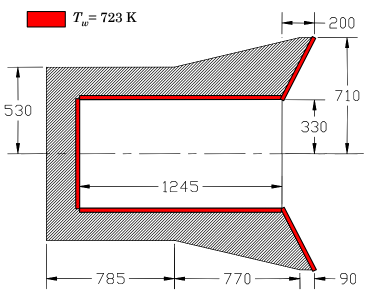
For each receiver, a series of simulations was undertaken to quantify the wall heat loss at varying receiver inclinations ϕ, ranging from 0^o (cavity aperture facing horizontally) to 90^o (cavity aperture facing vertically downward). Details of numerical procedure and construction of computational grids can be found in the previous ANZSES papers (eg. Paitoonsurikarn & Lovegrove, 2003).



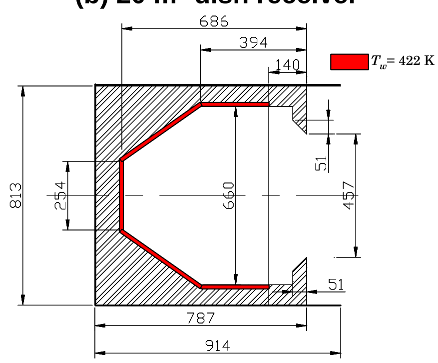
(a) Model receiver



(b) 20 m^2 dish receiver



(c) 400 m^2 dish receiver



(d) McDonald receiver

Figure 1 Configurations of receivers used in the study.

3. COMPARISON WITH EXPERIMENTS

Two main comparisons with experiments were made for the base cases of the model receiver and the McDonald receiver. Additional comparisons for the model receiver with different aperture areas and cavity wall temperatures were also undertaken as shown in the following subsection.

3.1. Model Receiver

Figure 2 shows comparisons for the base case and the case of different aperture areas. The exposure ratio R_{exp} in the figure is defined as the ratio of the aperture diameter to the cavity diameter. Clearly, both simulation and experiment show the good agreement to each other. It is also indicated in the figure that the heat loss significantly decreases with decreasing exposure ratio. For example, at zero inclination, the heat loss is reduced by almost 70%, when the exposure ratio decreases from 1.0 to 0.50.

It should be noted that the numerical results tend to overestimate the experimental data for the case of R_{exp}=0.50. This is expected to be due to the fact that in the experiment the annulus attached to the front of the cavity to reduce the aperture area had a finite thickness, i.e., ~5 mm, while in the simulation the thickness of the annulus was assumed zero (i.e. the annulus was modeled as the zero thickness wall). The convective flow is presumably reduced in the former case due to the thicker obstacle at the aperture plane, and the reduction should be more pronounced for the smaller exposure ratio. This is possibly the reason why the overestimation of the numerical result is not discernable for the case of R_{exp}=0.75, but somewhat obvious for the case of R_{exp}=0.50.

Figure 3 shows the comparison for the case of different cavity wall temperatures. Note that Taumoefolau *et al* (2004) carried out the experiments at the nominal cavity wall temperatures of 823 K and 923 K, the results of which are compared with the numerical results with the cavity wall temperatures of 800 K and 900 K, respectively. It evidently shows that both are overall in the very good agreement despite the slight difference of the wall temperatures used. However, at wall temperature of 900 K and the inclination of 90^{o}, the numerical result shows a tendency to underestimate the experimental result. This is presumably because the temperature used in the simulation was uniform throughout the cavity interior unlike that of the experiment, in which the end wall of the cavity was slightly cooler than its side wall. This temperature pattern can be realised from the wall temperature boundary condition used in the base case, which was essentially derived from the experimental data at nominal temperature of 723 K (cf. Figure 1a).

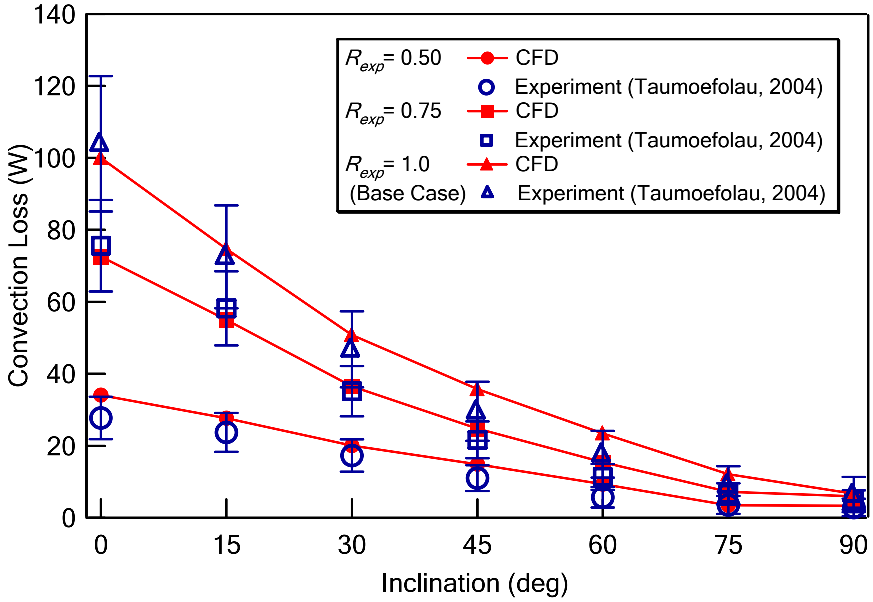


Figure 2 Comparison for the case of the model receiver with varying aperture areas.

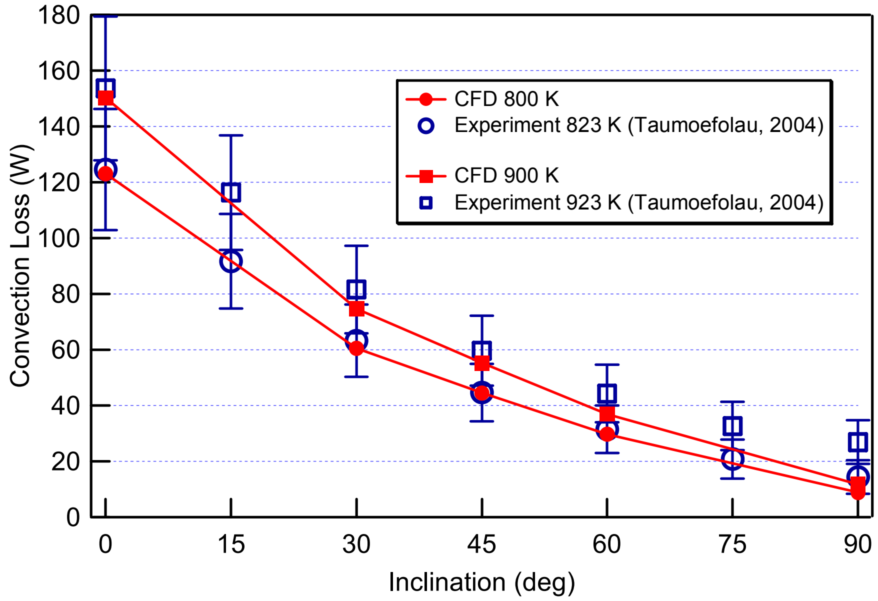


Figure 3 Comparison for the case of the model receiver with varying cavity wall temperature.

The cooler end wall promotes circulation inside the cavity at high inclination, resulting in higher convection loss than it would be if the end and side wall temperatures are similar. In the experiment, this increase of heat loss is more pronounced at higher nominal temperature, as the difference in the end and side wall temperatures becomes higher. There the numerical result is expected to underestimate the experimental value.

3.2. McDonald Receiver

Figure 4 shows the comparison of the convection loss from the numerical simulation and the experiment by McDonald (1995). It shows that the numerical results are in an acceptable agreement with the experimental data for the low inclination, i.e., $0^{o} \leq \phi \leq 30^{o}$, while they considerably overestimate the experimental data for the higher inclination. This disagreement is believed to result from the fact that the heat transfer surface of the Stine & McDonald receiver was coiled metal tube. As a result, the actual surface is corrugated, which is different from the simple smooth surface used in the numerical model.

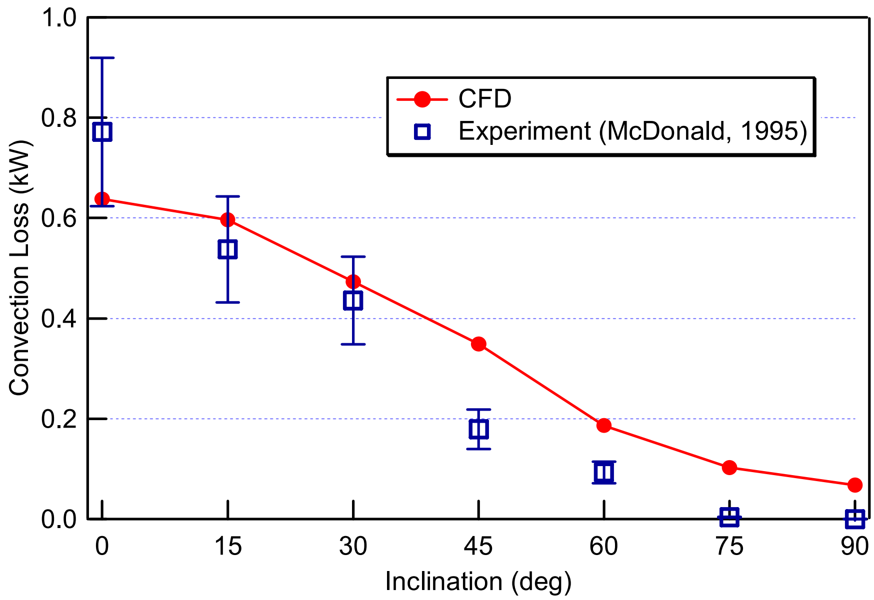


Figure 4 Comparison for the case of the McDonald receiver.

For the high inclination at which the heated air inside the cavity ascends along the cavity wall, the corrugated wall should exhibit the higher flow resistance than that of the smooth wall. Consequently, the resulting convection loss should be lower than predicted from CFD calculations for a smooth wall.

However, despite discrepancies between the numerical and experimental results which can be explained by the above arguments, it is confirmed that the numerical simulation is sufficiently good to predict convection loss that is not far different from a measured value for a wide range of receiver size, i.e. from the model receiver to the much larger McDonald receiver. Therefore, it gives confidence to rely on the numerical results of the other receiver geometries, which are used to develop the correlation in the subsequent section.

4. FREE CONVECTION CORRELATION

Free convection loss simulations were undertaken for the case of three receivers shown in Figures 1a-c, i.e. the model receiver, and the 20 m^2 and 400 m^2 dish receivers, with varying relevant parameters such as wall temperatures, sizes, to expand the database used in the correlation development. The total number of data points included was 210, which is ten times larger than that used to establish the original correlation.

As shown in two previous ANZSES papers (Paitoonsurikarn & Lovegrove, 2003; Paitoonsurikarn et al, 2004), it is possible to use the ensemble cavity length scale L_s to take into account the effects of cavity geometrical parameters and the inclination. Figure 5 shows the three relevant length scales used to define the ensemble cavity length scale L_s, i.e. L_1, L_2, and L_3, which are the average cavity diameter, the average cavity length, and the aperture diameter, respectively.

By fitting all available numerical data, it was found that the ensemble length scale L_s yields:

$$L_s = \left| \sum_{i=1}^{3} a_i \cos(\phi + \psi_i)^{b_i} L_i \right| . \tag{1}$$

The constants a_i, b_i, and ψ_i as summarised in Table 1. Note that the absolute sign is applied to the right hand side of Equation (1) because the summation can occasionally less than zero at the very high inclination, i.e. when ϕ approaches 90°, depending on the cavity geometry.

The modified correlation is in the following form:

$$Nu_L = 0.0196 \cdot Ra_L^{0.41} \cdot Pr^{0.13} , \tag{2}$$

where the Nusselt number Nu_L, and the Rayleigh number Ra_L are defined with the use of the ensemble length scale L_S:

$$Nu_L = \frac{hL_s}{k},\ Ra_L = \frac{g\beta(T_w - T_\infty)L_s^3}{\nu\alpha},\ \text{and the Prandtl number } Pr = \frac{\nu}{\alpha} , \tag{3}$$

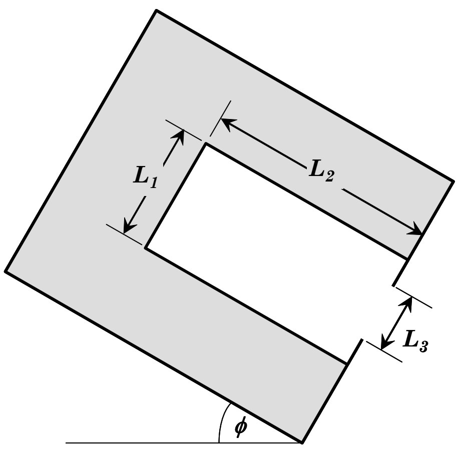


Figure 5 Denotation of cavity geometrical parameters used in the definition of the ensemble cavity length scale L_s.

Table 1. Constants in Equation (1) for the evaluation of the ensemble cavity length scale L_s.

i	a_i (-)	b_i (-)	ψ_i (rad)
1	4.08	5.41	-0.11
2	-1.17	7.17	-0.30
3	0.07	1.99	-0.08

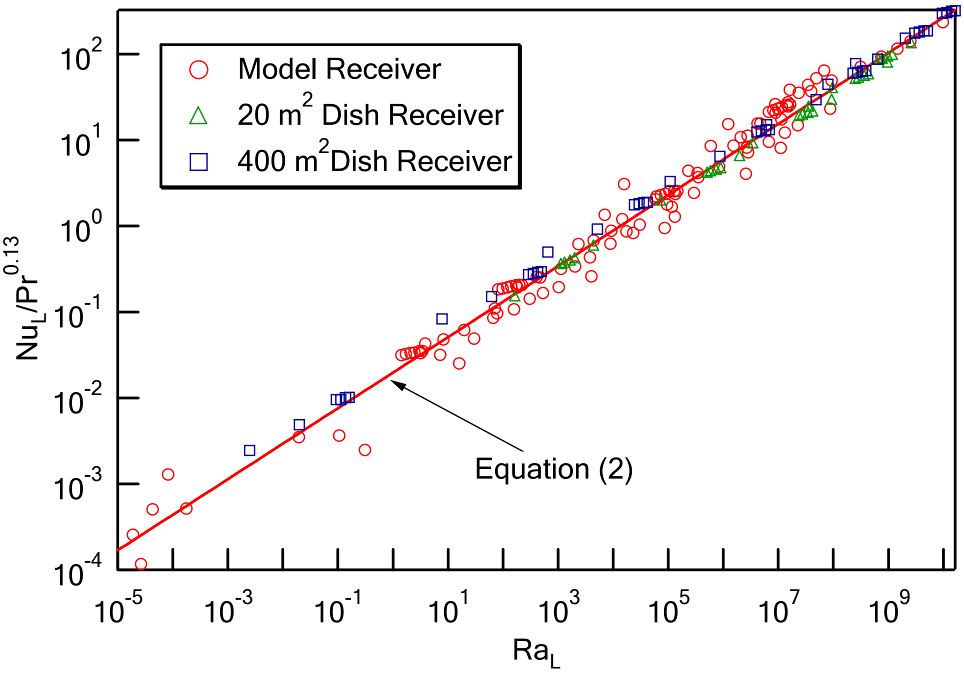


Figure 6 Comparison between the prediction by Equation (2) and the numerical results.

where h is the convective heat transfer coefficient. g is the gravitational constant. k, β, ν, and α are the thermal conductivity, the thermal expansion coefficient, the kinematic viscosity, and the thermal diffusivity of air, all of which are evaluated at the film temperature, i.e. the average between the cavity wall temperature T_w and the ambient temperature T_∞.

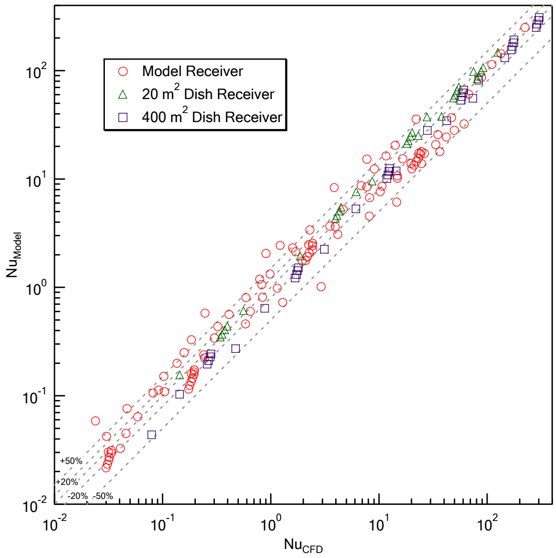
Figure 6 show the comparison between the prediction by Equation (2) and the numerical results from the three receivers. Overall, it correlates 103 data points within ±20%, and 184 data points within ±50%. Note that the large percentage error essentially occurs at very low Nu_L, where the inclination approaches 90°. This regime of Nu_L is rather insignificant since the magnitude of the convection loss is extremely low, compared to those at other inclinations.

Considering the prediction separately for each receiver, it is clearly indicated in Figure 6 that the maximum deviation is found in the case of the model receiver. The correlation correlates only 40% of the model receiver data within ±20%, and up to approximately 80% of the data within ±50%. On the other hand, the deviation in the case of other two receivers, i.e. the 20 m^2 and 400 m^2 dish receivers is relatively low. The correlation correlates as high as 70% of the data from both receivers within ±20%, and up to 100% coverage within ±50%.

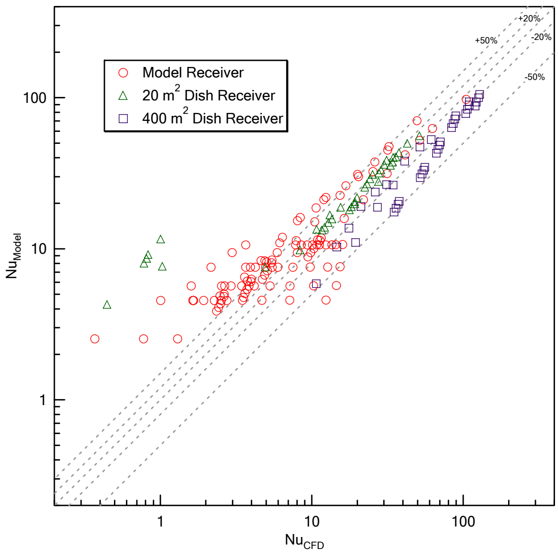
Although the correlation of Equation (2) does not yield an accurate prediction in some cases, particularly in the case of the small-scale model receiver, it does show a reasonably accurate prediction in the case of two full-scale receivers. Therefore, its application is rather justified, considering its simplicity and wider range of applicability.

Comparison of the present model with other previously proposed ones is shown in Figure 7. The following models are included:

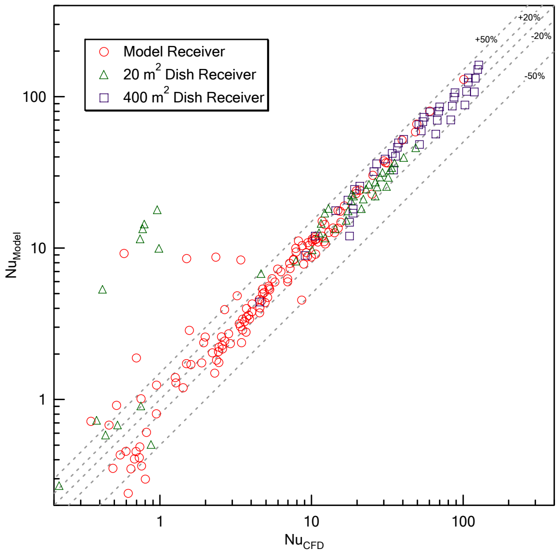
(1) Le Quere et al Model (Le Quere et al, 1981)
(2) Clausing Model (Clausing, 1983)
(3) Siebers & Kraabel Model (Siebers & Krabbel, 1984)



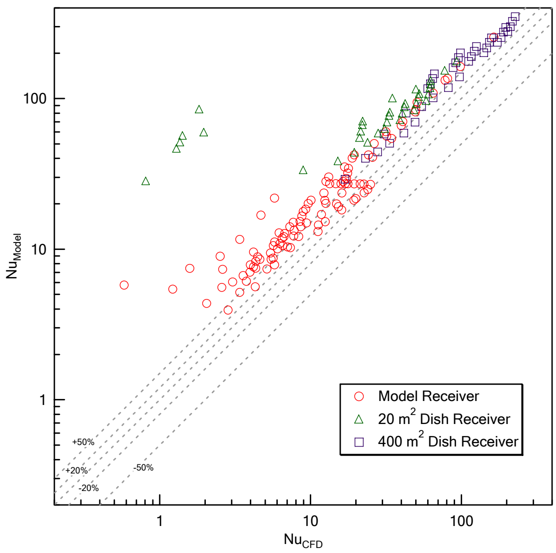
(a) New Model (Equation (2))



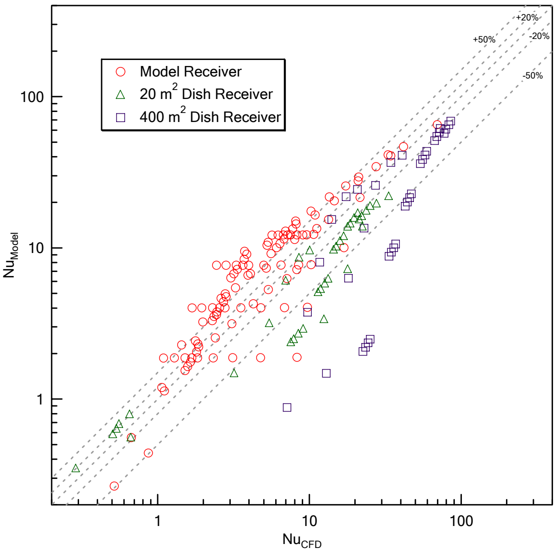
(b) Le Quere et al Model



(c) Clausing Model

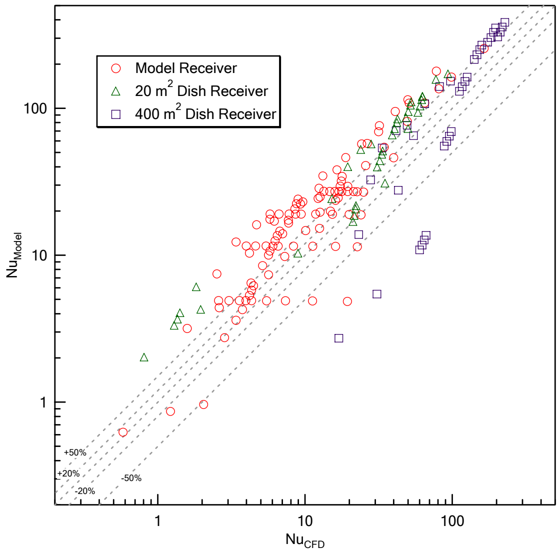


(d) Siebers & Kraabel Model

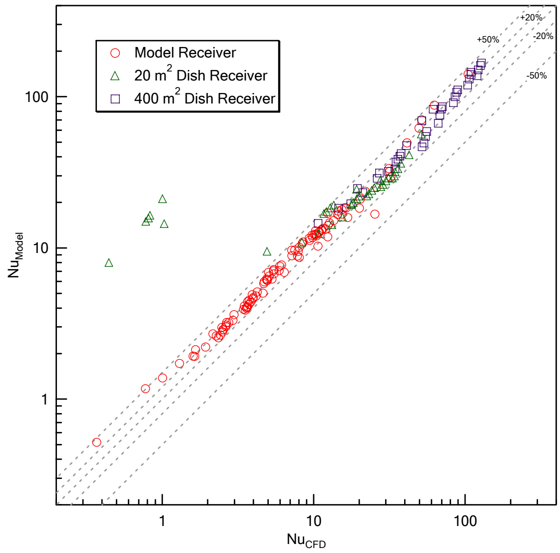


(e) Koenig & Marvin Model

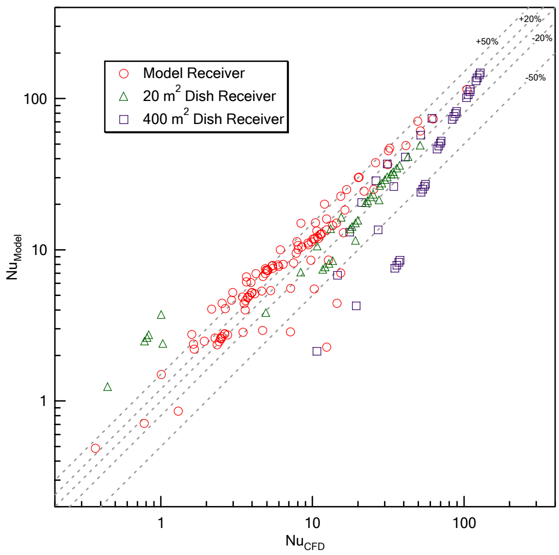
Figure 7 Comparison between the predicted results from various correlations Nu_{model} and the numerical results Nu_{CFD}.



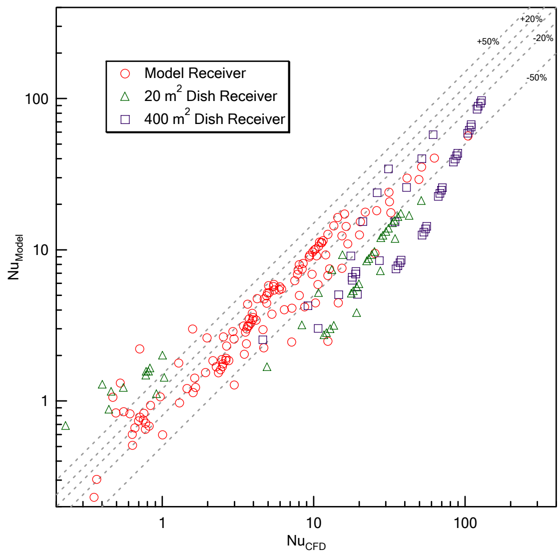
(f) Stine & McDonald Model



(g) Modified Clausing Model



(h) Modified Stine & McDonald Model



(i) Taumoefolau Model

Figure 7 (cont.) Comparison between the predicted results from various correlations Nu_{model} and the numerical results Nu_{CFD}.

(4) Koenig & Marvin Model (Koenig & Marvin, 1981)
(5) Stine & McDonald Model (Stine & McDonald, 1989)
(6) Modified Clausing Model (Leibfried & Ortjohann, 1995)
(7) Modified Stine & McDonald Model (Leibfried & Ortjohann, 1995)
(8) Taumoefolau Model (Taumoefolau, 2004)

It is obvious that the present model and the modified Clausing models give the best prediction of all numerical results of the three receivers, while all others show a considerable scatter. Overall, the modified Clausing model better correlates the results than the present model does, however it has large errors for some data points of the 20 m^2 dish receiver. The success of the modified Clausing model indicates that the concept of stagnant and convective zones originally proposed by Clausing (1981) does closely represent the actual mechanism of the convection loss in the cavity. The models that employ this concept promisingly show a wide range of applicability. However, they are very complicated and need considerable computational effort, especially in the determination of various areas defined in the models. Therefore, the use of Equation (2) is rather advantageous, as it is much simpler, and also yields a reasonably accurate prediction for all cavity geometries and operating conditions examined.

5. CONCLUSIONS

From the comparison with the experiments, it has been proven that the computational fluid dynamic (CFD) simulation can accurately predict the free convection loss from different open-cavity receivers of different sizes, geometries and operating conditions. The corresponding numerical results hence have been used in the development of an empirical model for loss prediction.

The concept of using the ensemble cavity length L_s in a convection loss correlation to account for the combined effect of the cavity geometrical parameters and the inclination is found to be applicable to a variety of receiver geometries. The resulting correlation has been shown to be more reliable and simpler to use than any of those published to previously by other authors.

6. REFERENCES

Clausing A. M. (1981), *An Analysis of Convective Losses from Cavity Solar Central Receivers*, Solar Energy 27, No. 4, pp. 295-300.
Clausing A. M. (1983), *Convection Losses from Cavity Solar Receivers-Comparisons between Analytical Predictions and Experimental Results*, Journal of solar energy engineering 105, pp. 29-33.
Fluent Inc. (2001), *Fluent 6.0 User Guide*.
Koenig A. A. and Marvin M. (1981), *Convection heat loss sensitivity in open cavity solar receivers*, Final Report, DOE contract No. EG77-C-04-3985.
Leibfried U. and Ortjohann J. (1995), *Convective Heat Loss from Upward and Downward-Facing Cavity Solar Receivers: Measurements and Calculations*, J. Sol. Eng. 117, 75-84.
Le Quere P., Penot F., and Mirenayat M. (1981), *Experimental Study of Heat Loss through Natural Convection from an Isothermal Cubic Open Cavity*, Proceedings DOE/SERI/SNLL workshop on convective losses from solar receivers, Sandia Laboratory Report SAND81-8014, Livermore, California, 165-174.
Lovegrove K., Taumoefolau T., Paitoonsurikarn S., Siangsukone P., Burgess G., Luzzi A., Johnston G., Becker O., Joe W. and Major G. (2003), *Paraboloidal Dish Solar Concentrators for Multi-Megawatt Power Generation*, Proceedings of ISES 2003, ISES Solar World Congress 2003, Sweden, 14-19 June 2003.
McDonald C. G. (1995), *Heat Loss from an Open Cavity*, Sandia National Laboratories.
Paitoonsurikarn S., Lovegrove K. (2002), *Numerical Investigation of Natural Convection Loss in Cavity-Type Solar Receivers*, Proceedings of 40th ANZSES Conference, 27-29 November 2002, Newcastle, Australia.
Paitoonsurikarn S., Lovegrove K. (2003), *On the study of Convection Loss from Open Cavity Receivers in Solar Paraboloidal Dish Application*, Proceedings of 41st ANZSES Conference, 26-29 November 2003, Melbourne, Australia.
Paitoonsurikarn S., Taumoefolau T., and Lovegrove K. (2004), *Estimation of Convection Loss from Paraboloidal Dish Cavity Receivers*, Proceedings of 42nd ANZSES Conference, 28 November - 1 December 2004, Perth, Australia.
Siebers D. L. and Kraabel J. S. (1984), *Estimating Convective Energy Losses from Solar Central Receivers*, Sandia National Laboratories, USA.
Stine W.B. and McDonald C.G. (1989), *Cavity Receiver Heat Loss Measurements,* presented at ISES World Congress, Kobe, Japan.
Taumoefolau T. (2004), *Experimental Investigation of Convection Loss from a Model Solar Concentrator Cavity Receiver*, Master Thesis, Department of Engineering, ANU, Canberra, Australia.
Taumoefolau T., Paitoonsurikarn S., Hughes G., and Lovegrove K. (2004), *Experimental Investigation of Natural Convection Heat Loss from a Model Solar Concentrator Cavity Receiver*, J. Sol. Eng. 126, 801-807.