SASEC
1st Southern African Solar Energy Conference
21-23 May 2012

HYBRID PRESSURIZED AIR RECEIVER FOR THE SUNSPOT CYCLE

Holger Kretzschmar[1], Paul Gauché[2]

[1] M. Sc Eng candidate, University of Stellenbosch, Private Bag X1 Matieland 7602, Stellenbosch, South Africa;

Phone: +27 (0)72 387 8676; E-Mail: 15025926@sun.ac.za

[2] M. Sc Eng, Senior Researcher & Coordinator, University of Stellenbosch; Solar Thermal Energy Research Group

Abstract

The Solar Thermal Energy Research Group (STERG) at Stellenbosch University is currently performing research on combined Brayton/Rankine cycle (called SUNSPOT) and the hybrid pressurized air receiver forms part of this research. The receiver concept is based on the well-established tubular receiver and open volumetric receiver technologies. It aims to eliminate the use of pressurized quartz window, whilst achieving high efficiencies and outlet air temperatures. Various receiver types have been outlined from which the hybrid receiver was derived and receiver concepts with close resemblance have been reviewed. Also a first principle energy balance has been set up to show the effect of reducing the convection and radiation losses on the receiver efficiency.

Keywords: CSP; hybrid pressurized air receiver; tubular receiver; open volumetric receiver; SUNSPOT; energy balance

1. Introduction

Central Receiver Systems (CRS) are large solar power plants where incoming solar irradiation is concentrated via multiple tracking mirrors, called heliostats. The concentrated solar rays are captured by the central receiver on top of a high tower. The receiver converts the concentrating solar irradiation to thermal energy. The thermal energy is absorbed by a heat transfer fluid and used in a conventional power cycle.

Various power cycles exist of which the most common cycles include Rankine cycles, Brayton cycles or combined Rankine/Brayton cycles. In Rankine cycles cold water is converted to steam, passed through a steam turbine to generate electricity and cooled down via a steam condenser to water. The Brayton cycle makes use of air as working fluid. Air is passed through a compressor, gains thermal energy by a heat source and flows through a gas turbine at extremely high temperatures. A combined Rankine/Brayton cycle utilizes the thermal energy that remains in the gas turbine outlet air flow to generate steam. The steam is used in the Rankine cycle. Thus, the Rankine cycle and Brayton cycle is combined to obtain a higher overall efficiency.

The efficiency of power cycles can be improved by increasing the temperature difference across the turbine. Since receivers form the primary heat source in solar power stations, high efficiency power cycles require receivers that are capable of providing high working fluid temperatures. The receivers as such also have to be efficient.

The following sections provide background on different types of receivers, a description of the hybrid pressurized air receiver, a first principle energy balance calculation and final conclusions are presented.

2. Background

Typical receivers found in literature are tubular receivers and volumetric receivers. Others include the Dual receiver concept and the novel annular reticulate porous ceramic (RPC) pressurized air receiver. The section below addresses these receiver technologies.

2.1. Tubular Receivers

Tubular receivers are the oldest and most mature receiver technology. They have been developed for 30 years and tested at numerous solar tower power plants worldwide. Tube receivers make use of steam as working fluid in first generation power plants. Second generation power plants utilize molten salts. Molten salt flows at atmospheric pressure inside the tubes. Therefore, the tube thickness can be less which enhances the heat transfer. Also, molten salts can accommodate higher solar fluxes, and therefore the entire receiver structure can be made smaller for the same output.

Recent studies have also investigated the application of tube receivers in pressurized air cycles (i.e. the SOLHYCO receiver development published by Uhlig et. al. [1]). This receiver was designed to produce an outlet fluid temperature of 800°C with a receiver efficiency of about 67.6% and metallic tube temperatures of up to 950°C [1]. Generally, tube receivers lack thermal efficiency and high receiver outlet temperatures. This is associated with the thermal conductivity of the tubes which restricts effective heat transfer to the working fluid. Also, material constraints of the receiver tubes due to high thermal stresses and non-uniform flux distribution around the tubes cause leakages and hot-spots are generated. However, they do not require a pressurized quartz window which would only increase the complexity and costs of the receiver.

2.2 Volumetric Receivers

Research and development has been performed on volumetric receivers over the last 20 years [2-5]. The operating principle for this receiver type is air forced through a porous medium. The medium is exposed to direct sunlight and thus gains thermal energy while absorbing the concentrated solar irradiation. Therefore, heat transfer occurs between the absorber medium and the air flow.

Volumetric receivers are more efficient than tubular receivers and obtain much higher temperatures (i.e. 80% and 1200°C) [4]. Thermal energy can be directly absorbed and is not subject to thermal conductivity losses through the absorber material. Also, due to the fact that the frontal surface temperature of the absorber is lowered through cold air penetrating into the porous medium, called "volumetric effect", less energy losses are obtained. Recent studies have shown that the volumetric effect is predominated by geometric properties such as mean cell sizes and porosity, and less due to thermal conductivity of the solid absorber material. Wu. et al. [3] showed that the surface temperature difference at the frontal absorber surface to a certain distance into the absorber, for 600K fluid outlet temperature and porosity variance from 0.7 to 0.9, was of 150K. This is due to solar irradiation penetrating deeper into the absorber material at higher porosities. Also, the mean cell size influenced the absorber frontal surface temperature significantly. The mean cell sizes stretch out the thermal non-equilibrium region deeper into the material. Here, a temperature difference of 50K was observed for different cell sizes. Furthermore, since porous media provide a large surface area for heat absorption the efficiencies and high fluid temperatures of volumetric receivers are supreme.

However, open volumetric receivers are only applicable to Rankine cycles. Studies and field experiences have also shown that open volumetric receivers can only sustain a certain amount of irradiation which is dependent on the pressure drop vs. depth through the porous medium [6]. It was found that the absorber surface would fail if not sufficient mass flow via pressure drop is ensured, especially at the centre of the receiver aperture where maximum solar flux is experienced. In comparison, much less pressure drop is required at the sides, close to the rims. This gives rise to a complicated control system problem where pressure drop versus irradiation needs to be regulated. Additionally, for higher temperatures the air density as well as heat capacity decrease and thus limiting the amount of heat absorption [6].

A way to overcome this problem is to make use of pressurized air which would increase the density of the working fluid. However, in this case a pressurized quartz window is required. Pressurized quartz windows pose several problems, including sensitivity to thermal shock loading, size limitation, thickness constraint in order to reduce radiation attenuation, accommodate high pressures and temperatures (quartz window only tolerates 800°C under pressurized conditions and thus requires active cooling [7]) and cleaning maintenance. Therefore, it would be beneficial to eliminate the use of a pressurized quartz window.

2.3 Dual and RPC Pressurized Air Receiver

The Dual receiver concept, proposed by Buck et al. [8], is an open volumetric air receiver with evaporator tubes placed in front of the receiver aperture [8]. Thus, steam is being generated not only by the hot air from the open volumetric receiver but also directly by the steam pipes. The steam pipes are primarily used to cool down the porous material at the frontal receiver aperture and thus reduce reradiation and convection losses. This concept showed an annual electricity production increase of 27%, compared to conventional open volumetric receivers [8].

The only other receiver concept that deviates from the receiver types mentioned above is the RPC pressurized air receiver. Here, the reticulate ceramic absorber foam is bounded by two concentric cylinders. This concept is more robust and less complex, compared to a typical closed-loop volumetric air receiver. Also, the use of a quartz window has been eliminated in this design. The receiver reaches an outlet temperature of 1000°C at 10 bars and an efficiency of 78% [7].

3. Hybrid Pressurized Air Receiver Concept

3.1 Concept Description

The Hybrid Pressurized Air Receiver (HPAR) consists of three zones, the secondary concentrator, the cavity and the secondary heat exchanger. The secondary concentrator, called compound parabolic concentrator (CPC), is optional to the design. The cavity consists of a transparent window (i.e. louvers or glass panes) and an array of absorber tubes. The third zone consists of a secondary heat exchanger, which is a counter-flow heat exchanger.

The receiver's thermodynamic cycle can be separated into two different loops, i.e. pressurized air loop, and the unpressurized air loop. Within the pressurized air loop compressed air is being preheated by the unpressurized air loop and enters the receiver cavity. Here, it is further heated by indirectly-irradiated tube bundles to the final maximum outlet temperature. The unpressurized air loop is similar to the open volumetric receiver concept where ambient air is being sucked into the receiver and is heated up by the absorber material. The enthalpy gained by the volumetric air stream is being recovered by means of a secondary heat exchanger. This also ensures that the residual heat in the air stream does not cause damage to the suction fan.

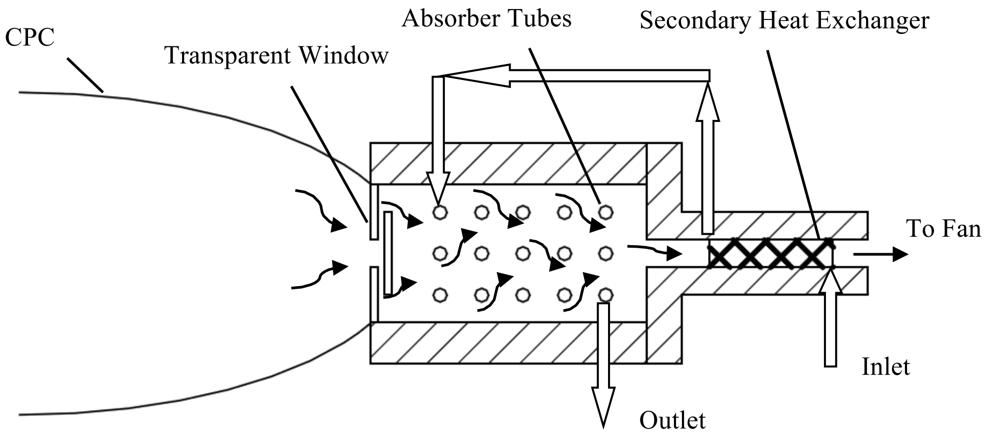

Fig. 1. Schematic layout of a preliminary concept of the hybrid pressurized air receiver

3.2 Advantages and Disadvantages

Due to the fact that the tubes on the irradiated side of the receiver are cooled higher solar flux densities can be accommodated. Thus the size of the receiver can be reduced which will lower the costs. Also, the ambient air flow stream aids in a more equalized temperature distribution around the periphery of each tube. Hot air on the irradiated side of the tube is dragged around the tube to the shadowed side and thus heats up the shadowed side while the irradiated side is cooled. Thus, thermal stresses on the tube material are reduced and buckling of tubes is minimized. Also, local hot-spots are less likely to occur not only due to the cold air stream but also since the tube material is thicker, compared to wires and foams.

Furthermore, the transparent window ensures that infra-red radiation emitted from the absorber tubes is recaptured. Therefore reradiation losses are minimized. The ambient air enters the receiver and cools down the transparent window whilst eliminating the use of active cooling mechanisms. Additionally, the air stream is preheated before striking the absorber tubes. This effect has been investigated by Pitz-Paal et al. [9] where quartz glass structures were placed in front of an open volumetric receiver. It was found that the efficiency would increase by 10% compared to typical volumetric receivers. Furthermore, since the transparent window remains unpressurized, higher temperatures are permissible. That is, unpressurized quartz glass can tolerate 1000°C, compared to 800°C for pressurized quartz glass windows.

A major disadvantage is that, since the HPAR makes use of tubes instead of a porous absorber material, the thermal conductivity of the tube material restrains the heat transfer to the air. This is a typical drawback for any type of indirectly-irradiated receiver. Also, the effective surface area for heat absorption is considerably less in the HPAR concept compared to a porous absorber material. Thus it is less likely for the HPAR to achieve higher temperatures than volumetric receivers. Also, in order to obtain effective heat conduction through the tubes walls the HPAR needs to make use of conductive metal alloys. These super-alloys (e.g. Inconel) achieve lower maximum temperatures compared to SiC foam in volumetric absorbers.

4. Energy Balance

The following section develops a first principle energy balance model to approximately predict the performance of the HPAR compared to a generalised cavity receiver. The relevant equations and results are presented.

4.1 Analytical Model

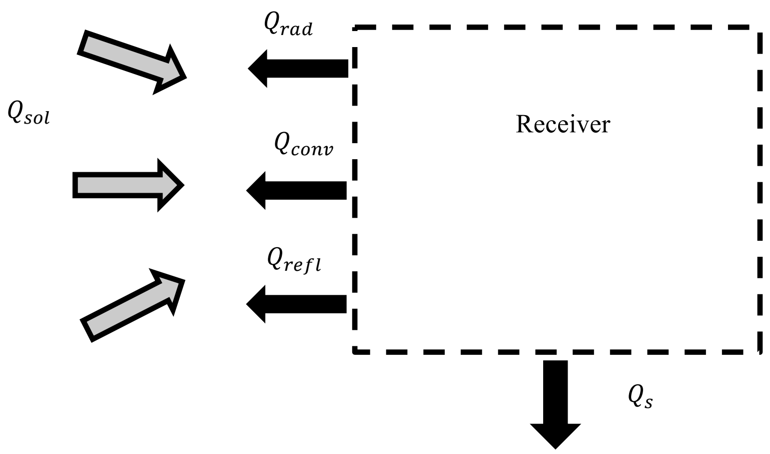

Fig. 2. Schematic diagram of lumped receiver model including the various energy terms

In order to obtain an indication of the various factors that influence the efficiency of a generalised cavity receiver a control volume around the entire receiver is placed. Then, the various energy terms have been identified, as shown in Figure 1, where the incident solar irradiation is given by equation 1.

$$Q_{sol} = G_{sol} \cdot n_{mirr} \cdot A_{mirr} \tag{1}$$

Here, the incident solar flux is denoted as G_{sol}, the number of mirrors used as n_{mirr} and each mirror area as $A_{ap.}$ Then, the reflection losses are determined by multiplying the incoming solar irradiation with a configuration factor and the reflective loss coefficient, as shown in equation 2.

$$Q_{refl} = Q_{sol} \cdot \rho_{refl} \cdot F_{i \to j} \tag{2}$$

A configuration factor of $F_{i \to j} \approx 0.2$ was calculated based on the assumption that the cavity depth is of the same size as the receiver aperture width and height. The reflective loss coefficient was typically defined as $\rho_{refl} = 0.05$. Then, further energy losses occur due to convection given by equation 3.

$$Q_{conv} = h_{nc} \cdot A_{ap} \cdot (T_s - T_{amb}) \tag{3}$$

Numerous studies have investigated the convection losses from cavities over the last few years [10-12]. These studies however differ significantly due to the variance in receiver sizes under consideration. Thus, for this analytical model the heat transfer convection coefficient was assumed to be $10 W/m^2.K$. This value is typical for air at ambient pressure and elevated temperatures. Convection losses for the HPAR energy balance model were neglected. It was assumed that the suction effect would eliminate these losses. Then, the radiation losses out of the receiver aperture for a generalised cavity receiver are calculated by equation 4.

$$Q_{rad} = \varepsilon \cdot A_{ap} \cdot \sigma \cdot F_{i \to j} \cdot ({T_s}^4 - {T_{amb}}^4) \tag{4}$$

The radiation loss calculation for the HPAR energy balance model includes a transparent window. Therefore, equation 5 is used [13].

$$Q_{rad} = \frac{\varepsilon \cdot A_{ap} \cdot \sigma \cdot F_{i \to j} \cdot ({T_s}^4 - {T_{amb}}^4)}{\left(\frac{1}{\varepsilon_1} + \frac{1}{\varepsilon_1} - 1\right) + \left(\frac{1}{\varepsilon_2} + \frac{1}{\varepsilon_2} - 1\right)} \tag{5}$$

The emissivity for the absorber material and glass were chosen as $\varepsilon_1 = \varepsilon_2 = 0.9$. It should be noted that the general radiation loss equation assumes that no reradiation occurs between the absorber material and the cavity walls, which is a highly conservative formulation. A better approach would be to split up the entire receiver into n- number of radiation zones and determine the energy balance for each zone independently. Nonetheless, despite the conservativeness, the general equation for radiation losses in this first principle analysis is sufficient. Similarly, a nodal surface temperature calculation would provide more accurate results, but the surface temperature was assumed to be equal to the fluid outlet temperature, $T_s \approx Tf_2$ instead. Here, Tf_2 was ranged from 25°C to 1000°C and therefore the radiation and convection losses could be determined. Then, the energy balance was used to calculate the useful heat transferred to the working fluid, as shown in equation 6.

$$Q_s = Q_{sol} - Q_{refl} - Q_{con} - Q_{rad} \tag{6}$$

It should be noted that the conduction resistance within the absorber material from the external absorber surface temperature to the internal absorber surface temperature was neglected. Also, the heat transfer between the internal absorber surface and the heat transfer fluid was assumed to be ideal at this stage. Finally, the efficiency at the specified temperatures is computed from equation 7.

$$\eta_{th} = \frac{Q_s}{Q_{sol}} \tag{7}$$

Table 1 is a list of the parameters that were used to plot the efficiency for the various receiver outlet temperatures.

Parameter	Value	Unit
n_{mirr}	50	-
G_{sol}	900	W/m^2
A_{mirr}	1	m^2
$F_{i \to j}$	0.2	-
ρ	0.05	-
ε	0.9	-
T_{amb}	300	K
L	1	m
h_{conv}	10	$W/m^2.K$
σ	5.678×10^{-8}	$W/m^2.K^4$
A_{ap}	1	m^2

Table 1. List of parameters used to determine the efficiency variance with respect to outlet temperature for a lumped model of the HPAR and a generalised cavity receiver.

A typical DNI value of $900W/m^2$ was chosen with 50 1m x 1m heliostats pointing at the receiver neglecting any field losses (e.g. shading, blocking and cosine losses). Also, ambient and fluid inlet temperatures were assumed to be at 300K.

4.2 Discussion and Results

Fig. 3 illustrates a plot of the receiver efficiency versus fluid outlet temperature. It can be seen that for a generalised cavity receiver, where radiation and convection losses are considered, efficiencies range between 60 and 40% for fluid outlet temperatures of 700 to 900˚C.

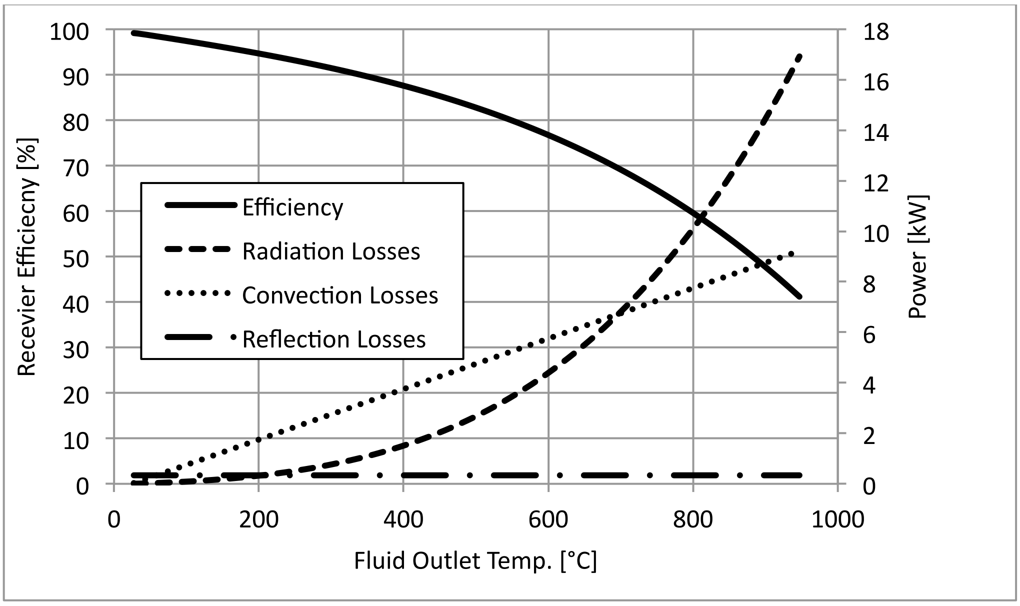

Fig. 3. Plot of the efficiency function versus radiation, reflection and convection losses for a generalised cavity receiver

Furthermore, radiation losses increase to the 4^{th} magnitude. Therefore, the reduction in radiation losses is a critical design consideration, especially at high temperatures.

Equation 4 shows the factors that influence the radiation losses. These include the emissivity coefficient, the configuration factor, the size of the aperture and the surface temperature of the emitting body. The emissivity of the absorber material is more or less given. The frontal surface temperature can be reduced by means of the suction flow. The size of the aperture can be reduced by means of a CPC. The configuration factor can be changed by altering the geometric configurations of the emitting bodies.

A sensitivity analysis has been performed to investigate the effect of changing the configuration factors. Figure 4 depicts the results. Note that the HPAR energy balance model neglects convection losses. Therefore, the receiver efficiency ranges from 30% to 80% with a configuration factors $0.1 < F_{i\to j} < 0.3$, at fluid outlet temperatures in the vicinity of 900˚C. The results are somewhat vague and thus a more sophisticated radiation model is required. However, it can be clearly seen that the efficiency drops drastically with high fluid outlet temperatures, especially at configuration factors greater than 0.2. Therefore, reradiation due to configuration of emitting bodies is a critical design factor.

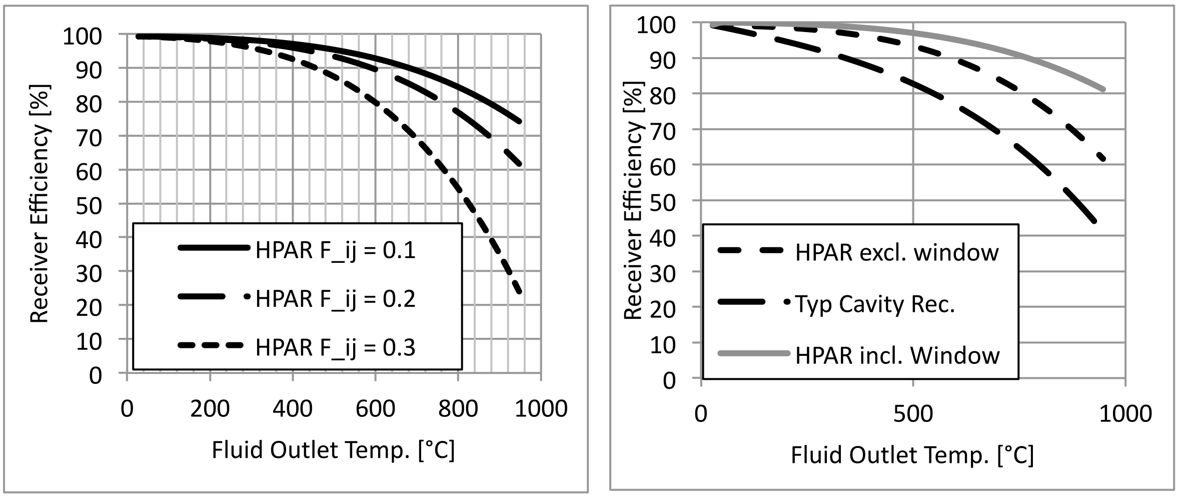

Fig. 4. Sensitivity analysis for radiation losses out of the HPAR with view factor configurations of $F_{i \to j} = 0.1$, $F_{i \to j} = 0.2$ and $F_{i \to j} = 0.3$; Receiver efficiencies for a generalised cavity receiver, the HPAR including a transparent window and the HPAR excluding transparent window, plotted over the fluid outlet temperature range of 0 to 1000 °C.

From earlier, reradiation can also be minimized by including a transparent window. The window transmits solar irradiation in the visible light spectrum. Then, once the light rays strike the absorber material heat is generated and wavelengths become shorter. The window is opaque to these shorter wavelengths and therefore reradiation is contained. Figure 4 illustrates the improvement and comparison for the HPAR including and excluding a transparent window with a typical cavity receiver. The results show that efficiencies of up to 80 % can be expected for the HPAR including a transparent window.

5. Future Work

Future work includes the development of a numerical simulation model that will investigate the flow patterns around the receiver in order to gain more insight on the natural convection losses, as well as the forced convection streams through the receiver tube bank. These results are to be coupled with a more sophisticated radiation heat transfer model. Also, experimental test are to be conducted on this concept for validation purposes

6. Conclusion

This paper presents a pressurized air receiver suitable for the SUNSPOT cycle, which is a cavity tubular receiver that makes use of the volumetric effect in order to minimize convection and radiation losses out of the receiver aperture. Various receiver types similar to this concept have been discussed and an energy balance approximation has been developed.

Acknowledgements

The authors would like to thank STERG, the DST/NRF solar spoke and the Stellenbosch University HOPE project for financially supporting this project.

References

[1] L. Amsbeck, G. Helsch, M. Röger, and R. Uhlig, “Development of a Broadband Antireflection Coated Transparent Silica Window for a Solar-Hybrid Driven Microturbine System,” Proceedings of the SolarPACES 2009, September, Berlin, 2009.

[2] R. Buck, “Solar-Hybrid Gas Turbine-based Power Tower Systems (REFOS) *,” vol. 124, no. February, 2002.

[3] Z. Wu, C. Caliot, G. Flamant, and Z. Wang, “Coupled radiation and flow modeling in ceramic foam volumetric solar air receivers,” Solar Energy, vol. 85, no. 9, pp. 2374-2385, September 2011.

[4] A. L. Ávila-Marín, “Volumetric receivers in Solar Thermal Power Plants with Central Receiver System technology: A review,” Solar Energy, vol. 85, no. 5, pp. 891-910, May 2011.

[5] J. Karni, A. Kribus, R. Rubin, and P. Doron, “The ‘ Porcupine ’: A Novel High-Flux Absorber For Volumetric Solar Receivers,” Journal of Solar Energy Engineering, vol. 120, 1998.

[6] A. Kribus, H. Ries, and W. Spirkl, “Inherent Limitations of Volumetric Solar Receivers,” Journal of Solar Energy Engineering, vol. 118, pp. 151-155, 1996

[7] I. Hischier, D. Hess, W. Lipiński, M. Modest, and a Steinfeld, “Heat Transfer Analysis of a Novel Pressurized Air Receiver for Concentrated Solar Power via Combined Cycles,” Journal of Thermal Science and Engineering Applications, vol. 1, no. 4, p. 041002, 2009.

[8] R. Buck, C. Barth, M. Eck, and W. Steinmann, “Dual-receiver concept for solar towers,” Solar Energy, vol. 80, no. 10, pp. 1249-1254, Oct. 2006.

[9] R. Pitz-Paal, J. Morhenne, and M. Fiebig, “Solar Energy Materials A new concept of a selective solar receiver for high temperature applications,” Solar Energy Materials, vol. 24, pp. 293-306, 1991.

[10] Taumoefolau, T., and K. Lovegrove. "An Experimental Study of Natural Convection Heat Loss from a Solar Concentrator Cavity Receiver at Varying Orientation." Proceedings of Solar 2002 Australian and New Zealand Solar Energy Society Annual Conference, Newcastle, Australia.

[11] S. Paitoonsurikarn and K Lovegrove. “Numerical Investi-gation of Natural Convection Loss in Cavity-Type Solar Receivers”. Annual Conference of the Australian and New Zealand Solar Energy Society, Newcastle, November 2002.

[12] Paitoonsurikarn, S. and K. Lovegrove, 2003, “On the study of Convection Loss from Open Cavity Receivers in Solar Paraboloidal Dish Application”, Proceedings of Solar 2003, ANZSES Annual Conference, Melbourne, Australia.

[13] Y. A. Cengel, “Heat and Mass Transfer: A Practical Approach, 3rd ed”, Singapore: Mc Graw-Hill, 2006, pp. 740-741.