HEAT TRANSFER MODELING IN RECEIVER/ABSORBER OF A DISH/STIRLING

André Leandro de Souza, als_shark@yahoo.com.br
Osvaldo José Venturini, osvaldo@unifei.edu.br
Vladimir Rafael Melian Cobas, vlad@unifei.edu.br
Electo Eduardo Silva Lora, electo@unifei.edu.br
Luis Sebastian Mendoza Castellanos, sebasmen@hotmail.com
Universidade Federal de Itajubá, Av. BPS, 1303, Itajubá - MG, 37500-903, Brazil.

Abstract. *Brazil, in the same way that the remaining of the world, is undergoing a growing on electricity's demand, mainly due to the technological progress as well as due to the constant seek for improvement of its socio-economic indicators. The increasing energy demand, together with the scarcity of water resources, the reduction in the fossil fuels reserves and the increasing concern with the environment, are challenging the scientific community to develop and/or to improve the usage of Renewable Energy, environmentally-friendly and low cost energy sources, such as the Solar Energy. This is the largest known source of clean energy, which can used in different ways. Some researches describe Brazil as a privileged country, due to its geographical location and its proximity to the equatorial axis of earth, having high solar radiation coverage year-round. This work focuses on the harnessing of Heliothermic Energy, aiming to analyze a Dish-Stirling system, more specifically the receiver/absorbed of this device. A dish-Stirling system consist of a parabolic solar concentrator, which concentrates the solar energy on a receiver that forms the hot part (the absorber) of a Stirling engine. The receiver/absorber is subjected to many thermal losses, which reduce its efficiency, and consequently the whole system performance. The quantification of these losses is one of the objectives of this work. So, an analysis will be performed using data obtained from a mathematical model, developed to quantify these thermal losses, together with experimental data, obtained from experiments conducted in the dish/stirling systems installed in the Heliothermic Energy Laboratory of the UNIFEI - Federal University of Itajubá, located in Itajubá, Minas Gerais.*

Keywords*: Heliothermic Energy, Dish/Stirling, Solar Energy, Renewable Energy, Receiver Cavity, Heat Transfer, Mathematical Modeling, Thermal Losses.*

1. INTRODUCTION

Currently still has the world's heavy dependence on fossil fuels for the development of numerous activities of various possible this global dependence on oil and fossil fuels cause a huge concern related to environmental issues, because the processes that use these fuels cause great CO2 emissions, on the other hand has also increased prices of these fuels, which generates an economic concern. Therefore it can be concluded that it is extremely important to reduce this dependence that still exists in the use of fossil fuels, and increase research related to clean energy or renewable energy. Thus the researchers are challenge to develop research related to clean energy such as heliothermic energy, which is nothing more than using solar thermal energy to produce electricity indirectly.

The usage of the Heliothermic energy can be based on various technologies of solar light concentration, including the Solar Tower, concentrators of Fresnel type, Cylindrical Parabolic and Parabolic collectors and Dish/Stirling, the latter is the system that this work intends to study. The Dish/Stirling system transforms solar energy into mechanical work by means of a Stirling engine, and this mechanical work is converted to electricity through a generator coupled to the motor. The Dish / Stirling system considered in this study is installed in Heliothermic Energy Laboratory (LEH) UNIFEI - Federal University of Itajubá, which also includes other devices, all related to research into renewable energy, such as a parabolic cylindrical system working with ORC (Organic Rankine Cycle).

This paper analyzes the thermal losses occurring in the receiver/absorber of the Dish/Stirling system, knowing that these losses generate a large decrease of the system efficiency. The heat transfer by conduction, convection and radiation that occur in the receiver/absorber are analyzed in order to determine the conditions in which the greatest losses occur, aiming to orientate future research in order to optimize the design of these systems to reduce such losses.

2. DISH/STIRLING SYSTEM

The Dish/Stirling system that in installed at LEH-UNIFEI has a solar concentrator of 3.75 m in diameter, determining an operational area of 9.58 m². It is constructed with multilayer aluminum mirrors, providing an optical efficiency of approximately 90%, according to the manufacturer. Figure 1 shows the Dish/Stirling system installed on LEH of the UNIFEI.

The solar collector shown in Fig. 1 concentrates the sun's rays on a focal point, where the Stirling receiver/absorber is located. The heat absorbed in receiver/absorber is conducted to the hot part of the Linear Free Piston Stirling Engine, expanding the working fluid, in this case the helium, and thus generating mechanical work. Subsequently the mechanical work is converted into electricity through a linear generator coupled to the engine shaft.

Figure 1 - Dish/Stirling system.

Figure 2 illustrates the Stirling engine used in Dish/Stirling system of LEH-. The manufacturer of solar concentrator is the Innova - Applied Brilliance, and model of the solar collector is the Trinum. The Stirling engine is manufactured by Microgen Engine Corporation, model DM 1.2S, with 01 kW of capacity.

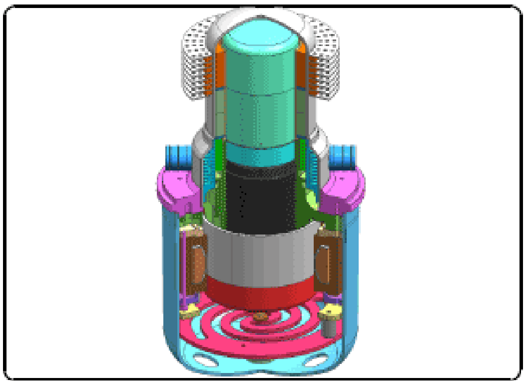

Figure 2 - Stirling engine (Linear Free Piston Engine Generator).
(http://www.microgen-engine.com/, Data: 18/06/2016)

Figure 3 shows the geometric characteristics of the receiver/absorber of the Dish/Stirling system, containing the actual dimensions of the system installed on LEH-UNIFEI. The material of the receptor is copper, the isolation is manufactured of glass wool and the external housing is manufactured with stainless steel.

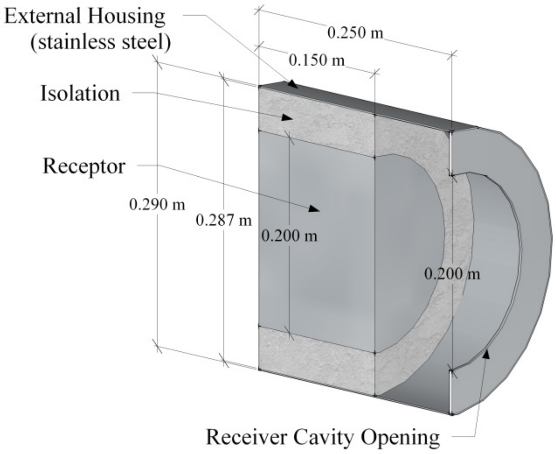

Figure 3 - Receiver geometric characteristics.

3. MATHEMATICAL MODEL

The mathematical model to quantify the heat losses in the existing receiver/absorber of the Dish/Stirling system was developed based on the following assumptions: fixed volume control around the receiver, steady state, constant temperature and laminar flow. A computational code was developed to solve the system formed by the heat losses governing equations.

The total energy used by the Stirling engine can be calculated by Eq. (1) (Castellanos et al., 2013).

$$Q_{gas} = Q_{rec} - \left(Q_{cond} + Q_{conv} + Q_{rad}\right) \tag{1}$$

Where: Q_{gas} : energy received by the Stirling engine through the receiver, [W].
Q_{rec} : concentrated solar power in the receiver cavity, [W].
Q_{cond} : heat loss by conduction, [W].
Q_{conv} : heat loss by convection [W].
Q_{rad} : heat loss by radiation, [W].

The amount of energy that reaches the receiver coming from the parabolic concentrator (dish) can be quantified by Eq. (2) according to Castellanos (2012).

$$Q_{rec} = \eta_{op} * A_{dish} * I_{sol} \tag{2}$$

Where: η_{op} : it is the optical efficiency of solar concentrator.
A_{dish} : it is the surface area of the dish concentrator, [m²].
I_{sol} : is the direct radiation incident on the dish concentrator, [W/m²].

Not all of the energy concentrated by the parabolic dish into the receiver/absorber can converted into mechanical energy, because there are many heat losses in this receiver, which will be examined below. So, the radiation losses from the collector can be calculated using Eq. (3) (Pavlovic and Penot, 1991).

$$Q_{rad} = \varepsilon * \sigma * A_{arec} * (T_{rec}^{4} - T_{amb}^{4}) \tag{3}$$

Where: ε: it is the emissivity of the receiver material.
σ: it is the Stefan-Boltzmann constant ($5.67*10^{-8}$ [W/m^2*K^4]).
A_{arec} : receiver aperture area, [m²].
T_{rec} : is the temperature in the receiver cavity, [K].
T_{amb} : the ambient temperature around the receiver, [K].

Besides these radiation losses, should be considered the heat losses by convection. To perform the analysis of the convection losses it is necessary to consider which type of convection is predominant: natural or forced convection.

According to Incropera *et al.* (2013), to determine which type of convection heat transfer is predominant, it is necessary to check the ratio between Grashof (*Gr*) and the square of the Reynolds number (*Re*), according to Gr/Re^2. Thus, if the ratio $Gr/Re^2 \ll 1$, natural convection can be neglected. If the ratio $Gr/Re^2 \gg 1$, forced convection can be disregarded in the calculations. Natural and forced convection and must consider if $Gr/Re^2 \approx 1$.

Before determining which convection regime is dominant, should be calculated the Reynolds number according to Eq. (4) (Jilte et al., 2014).

$$Re = \frac{\mathrm{v} * L_c}{\nu} \tag{4}$$

Where: v : it is the velocity of the air flowing near the receiver, [m/s].
L_c : is the characteristic length, [m].
ν : it is the kinematic viscosity of the air, [m²/s].

And to calculate the Grashof number (*Gr*) it is possible to use Eq. (5) (Pavlovic and Penot, 1991):

$$Gr = \frac{g * \beta * (T_{rec} - T_{amb}) * L_c^{3}}{\nu^2} \tag{5}$$

Where: g : acceleration of gravity, [m/s²].

β : is the volumetric thermal expansion coefficient, [K^{-1}].

The convection heat loss (Q_{conv}) occurring in the receiver can be calculated by Eq. (6) (Castellanos et al., 2013):

$$Q_{conv} = (Q_{i,n} + Q_{e,f}) \tag{6}$$

Where: $Q_{i,n}$: heat transfer by natural convection occurring inside the receiving cavity, [W].
$Q_{e,f}$: heat transfer by forced convection that occurs on the outside of the receiver cavity, [W].

Hence, to calculate $Q_{i,n}$, the loss by convection in the receiver cavity, use the Eq. (7) how in Prakash *et al.* (2009):

$$Q_{i,n} = h_{i,n} * A_{i,cav} * (T_{cav} - T_{amb}) \tag{7}$$

Where: $h_{i,n}$: is the convection heat transfer coefficient in the inner part of the receiver [W/m²*K].
$A_{i,cav}$: is the inner area of the receiver cavity, [m²].

The convection heat transfer coefficient in the inner part of the receivers is calculated according to Eq. (8) (Prakash et al., 2009):

$$h_{i,n} = \frac{Nu_{i,n} * k_{ar}}{L_c} \tag{8}$$

Where: k_{ar} : is the thermal conductivity of the air, [W/m*K].
$Nu_{i,n}$: is the local Nusselt number.

According McDonald (1995) the local Nusselt for natural convection can be calculated by the correlation shown in Eq. (9), where exponent m can be calculated according to Eq. (10):

$$Nu_{i,n} = 0.088 * Gr^{\frac{1}{3}} * \left(\frac{T_{cav}}{T_{amb}}\right)^{0.18} * \cos(\theta)^{2.47} * \left(\frac{d_{apert}}{d_{cav}}\right)^{m} \tag{9}$$

Where: d_{apert} : it is the diameter of the receiver aperture, [m].
d_{cav} : it is the diameter of the receiver cavity [m].
θ : it is the angle of inclination of the cavity.

$$m = 1.12 - 0.982 * \left(\frac{d_{apert}}{d_{cav}}\right) \tag{10}$$

According to Caballero *et al.* (2014), losses by forced convection on the outside of the receiver can be calculated by Eq. (11):

$$Q_{e,f} = h_{e,f} * A_{e,cav} * (T_{cav} - T_{amb}) \tag{11}$$

Where: $A_{e,cav}$: is the aperture area of the receiver cavity, [m²].

The heat transfer coefficient by forced convection on the outside of the receiver ($h_{e,f}$) can be obtained by Eq. (12) according to Caballero *et al.* (2014):

$$h_{e,f} = f(\theta) * v^{1.401} \tag{12}$$

And, $f(\theta)$ is the coefficient related to the receiver gradient, where θ is the angle of that inclination and can be obtained according to Caballero *et al.* (2014) by Eq. (13):

$$f(\theta) = 0.163 + \big(0.749 * \sin(\theta)\big) - \big(0.502 * \sin(2*\theta)\big) + \big(0.327 * \sin(3*\theta)\big) \tag{13}$$

Finally, the heat losses that occur by conduction, and further to the environment by convection, can be obtained by an analogy with a multilayer cylinder, and considering he thermal resistances of each layer (Eq. 14) how in Çengel and Ghajar (2012):

$$Q_{cond} = \frac{T_{cav}\text{-}T_{amb}}{\frac{ln\left(\frac{r_{iso}}{r_{rec}}\right)}{2*\pi*L*k_{iso}} + \frac{ln\left(\frac{r_{ext}}{r_{iso}}\right)}{2*\pi*L*k_{ext}} + \frac{1}{h_{e,c}*A_{e,c}}} \tag{14}$$

Where: r_{rec} : it is the radius of the receiver (inner cylinder, 1st layer), [m].
r_{iso} : it is the external radius of the thermal insulating material (2nd layer), [m].
r_{ext} : it is the outside radius of the collector (3rd layer, external housing of stainless steel), [m].
L : is the length of the receiver wall, [m].
$A_{e,c}$: is the external area of the receiver. [m²]
k_{iso} : it is the thermal conductivity of the insulation, [W/m*K].
k_{ext} : it is the thermal conductivity of the external housing (stainless steel), [W/m*K].
$h_{e,c}$: is the heat transfer coefficient by convection outside the cavity, [W/m²*K].

The natural convection coefficient of the external housing of the receiver ($h_{e,c}$) can be calculated by Eq. (15) according to Caballero *et al.* (2014).

$$h_{e,c} = \frac{(Nu_{e,c} * k_{ar})}{L_{rec}} \tag{15}$$

Where: L_{rec} : is the characteristic length of the receiver. [m]
k_{ar} : it is the thermal conductivity of air, [W/m*K].
$Nu_{e,c}$: is the Nusselt number

According to Hussain *et al.* (2016) the Nusselt number, for the case considered in this work, can be obtained by the correlation given in Eq. (16), which is valid for Prandtl number ≥ 0.6.

$$Nu_{e,c} = \left(0.664 * Re^{\frac{1}{2}} * Pr^{\frac{1}{3}}\right) \tag{16}$$

4. RESULTS

This work has the purpose to develop mathematical model to perform the calculation of the existing heat losses in the receiver of a Dish/Stirling system. These calculations were conducted based on the heat losses governing equations presented above, and using a computational code developed in Matlab to solve the resulting system of equations. With the computational code developed, is possible to assess these losses in different operation conditions of the Dish/Stirling system.

In Figure 4, it is shown the heat losses by conduction, radiation, and convection occurring in the receiver. The latter, heat loss by convection, already takes into account the natural convection inside the receiving cavity and also the forced convection outside the cavity. The values shown in Fig. 4 were calculated using data obtained from an experimental analysis where the values of ambient temperature vary between 298.25 K and 306.95 K; temperature of the receiver cavity vary between 459.75 K and 746.25 K; and wind speed vary between 1.7 m/s and 8.2 m/s. These experimental tests were performed the operation of Dish/Stirling system, on December 23, 2015, between the morning 7:38 and 12:14 hours. In Figure 4 can be identified that, the highest losses of the receiver are concentrated in the cavity, and occur by convection. The radiation loss also shows high values, especially when the temperature of the cavity increases. In addition, loss by conduction is the lowest.

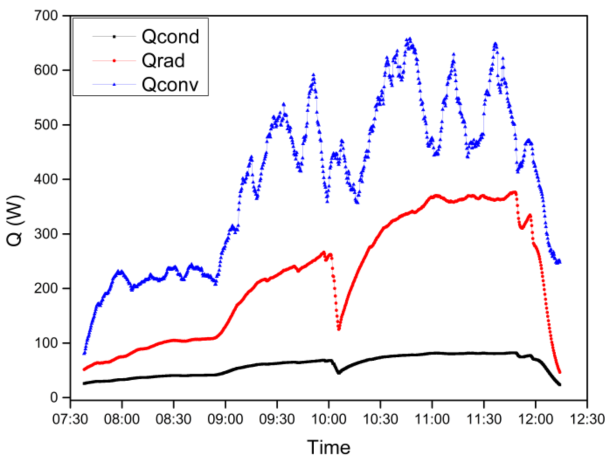

Figure 4 - Losses by heat transfer in the Dish/Stirling receiver.

It is possible to see that both losses, by convection and radiation, are greatly influenced by the temperature of the receiver cavity, what can be seen in Fig. 5. The convection losses, besides suffering the influence of the temperature's changes in the receptor cavity, also suffer the influence of wind speed, as can be seen in Fig. 6.

The losses by convection, besides being influenced by temperature variations and wind speed, are also affected by the receptor slope. Fig. 7 shows the influence of the receptor slope on heat losses by natural convection in the receiver cavity, and Fig. 8 shows the losses by forced convection on the outside of the receiver cavity. The slope of the Dish/Stirling occurs because of its sun tracking system. In order to maximize the system efficiency the Dish/Stirling system continuously tracks the sun position (solar azimuth and elevation), throughout the day along the year. In Figure 7 and in Figure 8 it is possible to observe that the receiver slope causes a decrease in the heat losses by convection as the tilt angle increases, especially inside the cavity receptor. There is also a decrease in losses on the outside of the receiver cavity, but with less intensity.

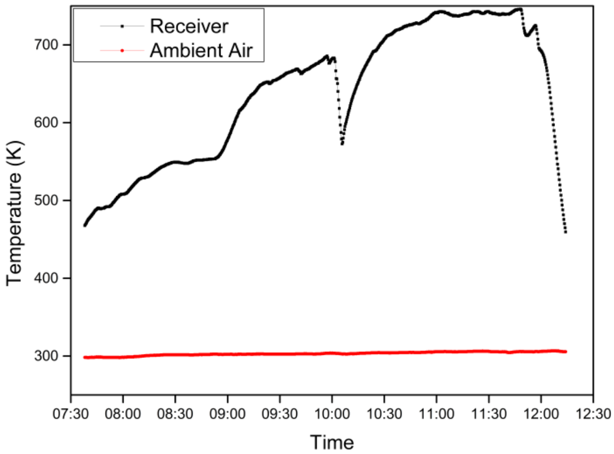

Figure 5 - Temperature variation, inside of the receiver and outside in ambient air.

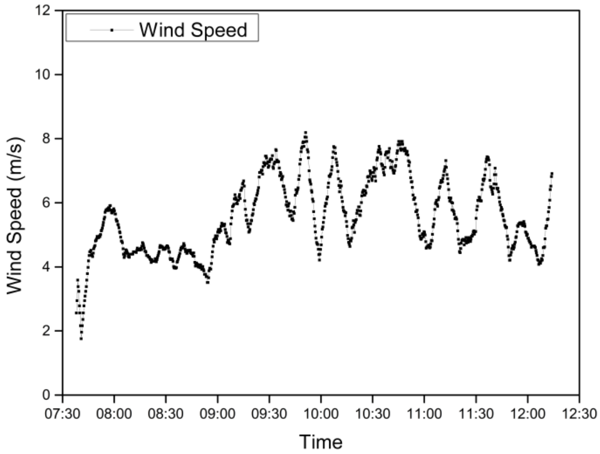

Figure 6 - Wind speed variation on the day of experimental analysis.

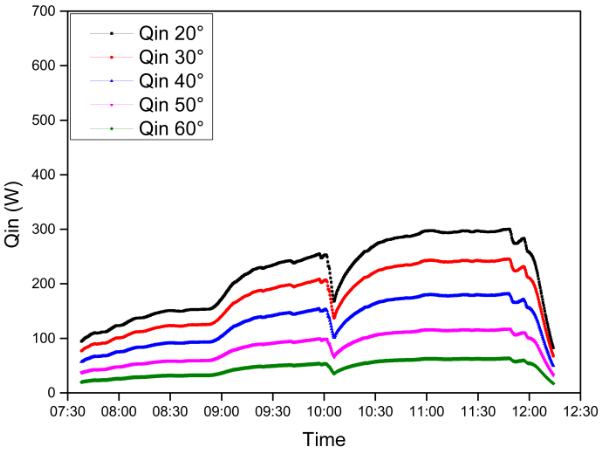

Figure 7 - Influence of the receptor slope on heat losses by natural convection in the receiver cavity.

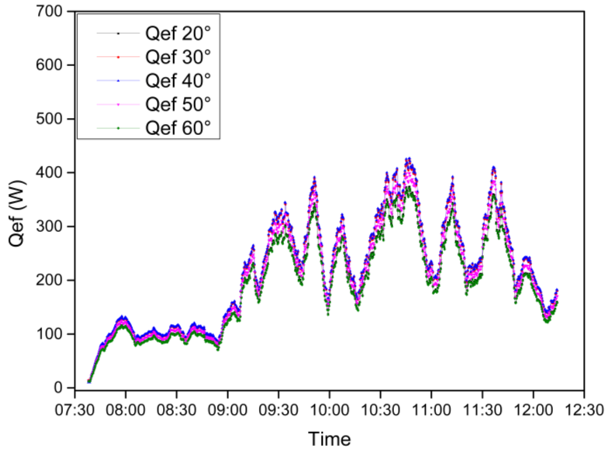

Figure 8 - Influence of the receptor slope on heat losses by forced convection on the outside of the receiver cavity.

Figure 9 highlights the influences of receptor slope in the total convection losses, where it is possible to observe a considerable influence of the receptor slope in total losses by convection.

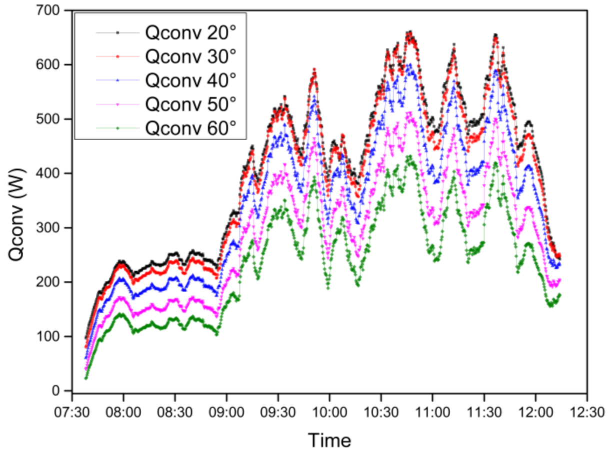

Figure 9 - Influence of the receptor slope on heat losses by convection.

5. CONCLUSION

This paper presents a mathematical model to describe and quantify the thermal losses in the receiver/absorber of Dish/Stirling systems. The heat losses by conduction, convection and radiation are one of the factors that decreases the conversion efficiency of system, so it is very important be able to describe the behavior of these losses to propose changes in system design/operation to minimize them.

The results obtained make it possible to verify that, the maximum losses typically occur when there are higher temperatures in the receiver cavity. It was found that the conduction losses reach a maximum value of 73 W, radiation losses reach 376 W, while losses by convection are higher than 657 W, when the receiver is on the slope at 30º, and 431 W with 60º tilt. The maximum value of the total combined heat loss is approximately 1087 W for a 30º tilt, and 857 W for a slope 60º of the receiver, both considering that the maximum temperature in the receiver is 739 K. Considering these values, it is possible to conclude that there are significant changes in the heat losses related to the changes in the receptor slope.

The receptor's slope only modifies the convection losses that occur in the receiver opening of the Dish/Stirling system, suggesting that a way to increase the thermal efficiency of energy conversion is to propose a modification of the geometry of the aperture of the receiver. This geometry modification/optimization is being evaluated in the study under development by the authors of this paper.

6. ACKNOWLEDGEMENTS

The authors want to thank the National Council of Technological and Scientific Development (CNPq), the Companhia Paulista de Forca e Luz (CPFL), the Coordination of Improvement of Higher Education (CAPES) and the Foundation for Research Support of Minas Gerais State (FAPEMIG) for their collaboration and financial support that enabled the realization of this work.

7. REFERENCES

Caballero, G.E.C., Martinez, A.M., L.S., Lora, E.E.S., Cobas, V.R.M., Venturini, O.J., 2014. “Análisis de Un Sistema de Dish Stirling Trabajando Con Receptor DIR y Con Receptor Heat Pipe.” In *Convención Internacional de Ciencias Técnicas*, Santiago de Cuba, Cuba, Vol. 1, pp. 1-6.

Castellanos, L.S.M., 2012. Modeling Systems for Generating Electricity from Solar Energy Using Parabolic Dishes and Stirling Engines (Solar Dish). MSc. Dissertation. Department of Pos-graduation Energy Engineering, Federal University of Itajubá, Itajubá, MG, Brazil.

Castellanos, L.S.M., Caballero, G.E.C., Lora, E.E.S., Cobas, V.R.M., 2013. “Modeling of a Cavity Receiver of a Solar Dish / Stirling System.”. In 22nd International Congress of Mechanical Engineering - COBEM 2013. Ribeirão Preto, SP, Brazil. pp. 714–724

Çengel, Y.A., Ghajar, A.J., 2012. Transferência de Calor E Massa: Uma Abordagem Prática. 4. ed,. AMGH Editora Ltda, Porto ALegre, RS.

Hussain, T., Islam, M. D., Kubo, I., Watanabe, T., 2016. “Study of Heat Transfer through a Cavity Receiver for a Solar Powered Advanced Stirling Engine Generator.” Applied Thermal Engineering, Vol. 104, pp. 751–757.

Incropera, F.P., Dewitt, D.P., Bergman, T.L., Lavine, A.S., 2013. Fundamentos de Transferência de Calor E de Massa. 6 ed,. LTC, Rio de Janeiro, RJ.

Jilte, R.D., Kedare, S.B., Nayak, J. K., 2014. “Investigation on Convective Heat Losses from Solar Cavities under Wind Conditions.” Energy Procedia, Vol. 57, pp. 437–446.

McDonald, C.G., 1995. Heat Loss from an Open Cavity. College of Engineering California State Polytechnic University, Pomona, CA.

Pavlovic, M.D., Penot, F., 1991. “Experiments in the Mixed Convection Regime in an Isothermal Open Cubic Cavity.” Experimental Thermal and Fluid Science, Vol. 4, pp. 648–655.

Prakash, M., Kedare, S.B., Nayak, J.K., 2009. “Investigations on Heat Losses from a Solar Cavity Receiver.” Solar Energy, Vol 83, pp. 157–170.