MODELLING AND SIMULATION OF A DISH STIRLING SOLAR ENGINE

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Abstract: Motivated by the necessity to study a Dish Stirling solar real plant, in this paper a model suitable for simulation and control is worked out. The model obtained from basic thermodynamical equations has the main disadvantage of being a stiff system. A handy and significant model is therefore derived by an analytical approximation approach. The comparison with experimental data is very satisfactory.

Keywords: Control system design, dynamic behavior, engine modelling, model approximation, power generation, simulators.

1. INTRODUCTION

In the field of solar based energy suppliers, a device of major interest in our days is the so-called Dish Stirling engine. In such engine, a parabolic mirror focuses the incoming sun rays towards a receiver acting as a thermal source for a thermodynamical machine based on the Stirling cycle (two isovolumic and two isothermal transforms). In the receiver and in the whole thermodynamical machine, helium gas circulates, pushed by the alternate movement of pistons oscillating inside two cylinders in V configuration. In this paper, it's first supplied a model of such an engine by describing the behavior of the various components with ordinary differential equations. A feature of this system is that the involved state variables can be grouped in two categories: those related to the walls' dynamics and those related to the gas behavior. The former exhibit the typical (slow) dynamics of thermal exchanges between metal and gas, metal and thermal source, metal and environment. The latter are also subject to mechanical effects, so that they can be seen as "fast" variables. In fact, they are periodically varying at $30Hz$ as a consequence of the piston oscillations taking place at that frequency. The different behavior of these two groups of state variables is a remarkable obstacle for the use of the model in practice. Indeed, for the fast variables one can conceive only the control of their average behavior. Furthermore, the model does not lend itself to explicate the correlations between the exogenous variables (such as emitted power) and internal state variables. Last, the model performs very slowly in simulation due to the fast sampling required for the gas variables. To overcome these problems, we propose to approximate the fast oscillating variables with sinusoidal signals with average values, amplitude and phase varying in time (*cisoids*). By solving the differential equations of the original model with such an approximation, it is possible to capture the slow "average behavior" of the fast variables. In this way, one can work out a new model comprising the slow state variables of the original model plus the average variables describing the thermal dynamics of the gas. The so obtained model is suitable for simulation, since the dynamics of all variables have the same rate, and for control, since it describes the essential features of the gas variables. In this way, as it will be described in the sequel, it is possible to work out correlations among the variables which clarify the theoretical behavior of the Dish Stirling solar engine. Despite the problem of modelling and simulate the behavior of the Stirling Engine has been dealt for a long time (see, for example ((Rodgakis *et al.*, 2002)), or ((Hirata *et al.*, 1997))), to the author's knowledge this is the first paper where the *cisoid approach* is adopted to such a device leading to these general results. The results obtained by our model have been compared to experimental trials performed with the Dish Stirling plant of CESI-Milan. Indeed, the research activity presented herein is the outcome of a collaboration between CESI and the Milan Institute of Technology (Politecnico di Milano). The CESI engine is a $10kW$ plant with a dish of $8.5m$.

This paper is organized as follows: in Section 2, the main components of the plant are described; the key equations required to describe the "thermo-mechanical" subsystems of the engine are presented in Section 3; in Section 4 we apply the "cisoid approach" to pass from the original model to the simplified one; thanks to such a model one can work out an expression for the generated mechanical power; moreover, it is possible to design a control system satisfying the specifications of the plant (Section 5); finally, some simulation trials are performed in order to compare the model behavior to the CESI plant data (Section 6). The comparison indicates that the model worked out is very satisfactory.

2. THE DISH STIRLING ENGINE PLANT

The thermo-mechanical part of the Dish Stirling Engine is an external combustion thermodynamical machine, whose thermal source is represented by insulation. The sun radiation is concentrated on the *receiver* by a *concentrator*, with a parabolic mirror (*Dish*) that rotates according to the sun position (Figure 1).

The *engine* (see (Walker, 1980)) is based on the *Stirling thermodynamical cycle* (consisting of two isovolumic and two isothermal transformations) where the working gas is helium. The engine is a two-cylinders engine (in "V configuration", see Figure 2). The helium gas is delivered by two *valves* from the so-called *helium bottle* to the compression cylinder where the temperature is about $20°C$ (low temperature).

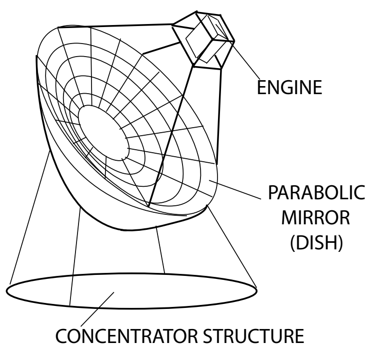

Figure 1. Plant scheme

The first (isovolumic) phase of the thermodynamical cycle takes place when the gas, in the *compression cylinder* (*cold cylinder*) is pushed by the corresponding piston to the chamber of the second *expansion cylinder* (*hot cylinder*), passing through a device called *regenerator* (which gains heat while gas passes from the heat source to the cold cylinder, and "looses" heat while gas passes from the cold cylinder to the heat source) and then the *receiver* (heat source). Along this path, the temperature of the gas increases from about $20°C$ (in the cold cylinder) to about $600°C$-$800°C$ (in the hot cylinder). The second (isothermal) phase is the expansion of the hot gas in the expansion cylinder. Such expansion is due to the heat accumulated in the receiver and produces mechanical work. Then the hot gas goes back on the same path "loosing" heat, so performing another isovolumic transformation. The last phase is the isothermal compression taking place in the cold cylinder. The cold cylinder's walls are kept at a low temperature thanks to a suitable *cooling system*. The produced mechanical energy is transformed into electrical power by an *induction motor*, by which the plant supplies energy to the power network. The whole engine is equipped with a *control system* the main scope of which is to keep the receiver's temperature constant, at a given set point value.

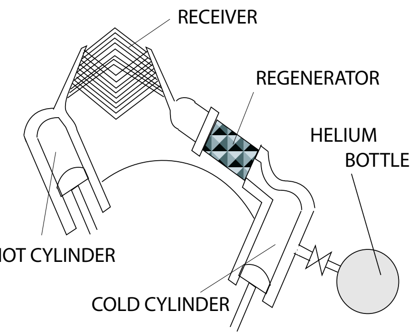

Figure 2. Scheme of the thermo-mechanical part of the engine

3. THE BASIC MATHEMATICAL MODEL OF THE PLANT

3.1 Main Variables

To work out the model of the plant, the following assumptions are introduced. First of all, the temperature of the gas within each chamber of the two cylinders is considered uniform, as well as the temperature of the walls of the cylinders. Furthermore, the uniformity assumption is made for the receiver too. What about the regenerator, whose thermodynamical behavior is more complex than that of the other components (see (Mayzus *et al.*, 2002) and (Organ, 2000)), it is subdivided it into 10 "slices", in each of which the temperature is assumed to be uniform. As far as the pressure is concerned, the spatial variation along the engine is relatively small (due to gas friction) and it is neglected for the equations. This corresponds to considering a uniform mean pressure in the whole engine. Of course, the gas friction is taken into account in the expression of the generated power. The adopted symbols are:

- $T_{c,g}(t)$: cold cylinder gas temperature,
- $T_{reg,g,i}(t)$: regenerator i-th slice's gas temperature ($i = 1, 2, ..., 10$),
- $T_{rec,g}(t)$: receiver homogeneous gas temperature,
- $T_{h,g}(t)$: hot cylinder gas temperature,
- $p(t)$: engine's mean gas pressure,
- $T_{c,w}(t)$: cold cylinder walls temperature,
- $T_{reg,w,i}(t)$: regenerator i-th slice's walls temperature ($i = 1, 2, ..., 10$),
- $T_{rec,w}(t)$: receiver homogeneous walls temperature,
- $T_{h,w}$(t): hot cylinder walls temperature.

Moreover, there are further variables representing the helium flow in the chambers. For instance:

- $z_c(t) = \bar{z}_c cos(\omega t)$, $z_h(t) = \bar{z}_h cos(\omega t + \varphi_h)$: pistons' positions in the cold and hot cylinder ($\omega = 30Hz$),
- $u_c(t), u_h(t)$: pistons' speeds ($u_c(t) = \dot{z}_c(t), u_h(t) = \dot{z}_h(t)$). It's assumed that $u_c > 0$ implies: cold piston ingoing, $u_h > 0$ implies: hot piston outgoing),
- $w_c(t)$, $w_h(t)$: the gas mass flows $[Kg/s]$ between the compression cylinder and the receiver, and between the receiver and the hot cylinder, respectively. Obviously $w_c(t) = A_c \rho_c u_c(t)$, and $w_h(t) = A_h \rho_h u_h(t)$, where A_c e A_h are the sections of the cold and hot cylinders, ρ_c and ρ_h are the gas densities in the cylinders ($\rho_c = \frac{p}{\tilde{R}T_{c,g}}$ and $\rho_h = \frac{p}{\tilde{R}T_{h,g}}$),
- w_{in}, w_{out}: gas mass flows from the gas bottle to the cold cylinder ($w_{in} \geq 0$) and from the cold cylinder to the helium bottle ($w_{out} \geq 0$), respectively.

3.2 Basic equations

The typical approach used in modelling such a device is formally describing his thermodynamical behavior using conservation laws (see, for example (Organ, 2002)). Account taken of the above simple schematization of the thermodynamical machine, we have led to consider 1 state equation related to the gas pressure, 13 state equations for the walls' temperatures, 13 state equations for the gas temperatures.

The gas pressure p equation is obtained by the mass conservation principle for a gas in a pipe. Precisely, let A the section area of the pipe, T_g the gas temperature (uniform in the pipe). Furthermore, denote by w_1 and w_2 the gas mass flows passing through the extremal points of the tube with coordinates z_1 and z_2. The conservation law is expressed as follows:

$$\frac{A(z_2(t) - z_1(t))}{\tilde{R}T_g}\dot{p} - \frac{p}{\tilde{R}T_g^2}A(z_2(t) - z_1(t))\dot{T}_g = \\ = w_1 - w_2 \qquad (1)$$

where $\tilde{R} = R/p_m$, R being the Boltzmann constant and p_m being the molar weight of helium. Note that in this equation, when applied to the cylinder chambers, will present time varying coordinates z_1 and z_2.

Passing to the 13 wall temperatures, the basic equation is the energy conservation law for a metal (at temperature T_w, uniform in the pipe) exchanging heat with a gas and a cooling system of with the external environment, and subject to a generic source supplying heat at the rate Q_{in}, and with a generic heat release Q_{out}:

$$\begin{aligned} c_{wall} M_{wall} \dot{T}_w &= \\ &= -\gamma_{mg}\Omega(T_w - T_g)(z_2(t) - z_1(t)) + \\ &\quad -\gamma_{cw} S_{cw}(T_w - T_{water}) + \\ &\quad +Q_{in} - Q_{out} \end{aligned} \qquad (2)$$

Obviously, in this equation c_{wall} is the specific heat of the metal, M_{wall} the corresponding mass, T_{water} the temperature of the cooling water, γ_{mg} and γ_{cw} are the heat exchange coefficients between the gas and the metal, and the cooling water and the metal, S_{cw} being the exchange surface between water and metal.

Finally, for the third group of variables the basic equation is the energy conservation law for a gas in a pipe. Precisely, let c_v the specific heat of the gas, and denote by T_i and ρ_i the temperature and density of gas at the extremal point of the tube, denoted by the coordinate z_i ($i = 1, 2$).

$$\begin{aligned} c_v A \rho \dot{T}_g(z_2(t) - z_1(t)) &= \\ &= \gamma_{mg}\Omega(T_w - T_g)(z_2(t) - z_1(t)) + \\ &\quad -c_v w_2(T_2 - T_g) + c_v w_1(T_1 - T_g) + \\ &\quad +p\left[\left(\frac{w}{\rho}\right)_1 - \left(\frac{w}{\rho}\right)_2\right] \end{aligned} \qquad (3)$$

3.3 The overall basic model

By exploiting the above equations, the overall model is a state space lumped parameter model with 27 state equations. Focus now on the last 13 equations. Among them, two equations refer to the gas temperatures in the cylinders. These equations are affected by the main exogenous variables, the oscillating ($30Hz$) pistons' positions. This implies that the temperatures of the cylinders present an oscillatory behavior too. Moreover, the gas mass flows w_c and w_h obviously depend on the piston velocities, and therefore they are both oscillating variables. Finally, w_c and w_h act as exogenous variables in the remaining 11 gas temperature equations. This means that all temperatures in the regenerator plus the temperature in the receiver exhibit oscillatory behavior. In conclusion all 13 gas temperature variables have oscillations at $30Hz$.

In turn, this implies that the pressure is oscillating too. As for the temperature of the cylinders' walls and the temperature of the receiver, the thermal inertia is so high that the effects of oscillations of the internal gas temperatures are filtered out, so that the 3 temperatures have slow dynamics. As for the last 10

temperatures (regenerator), the oscillatory behavior is maintained since the thermal inertia is not extremely high.
Summing up, of the 27 variables in the model, 24 present oscillations at $30Hz$, and 3 have slow dynamics.

3.4 Power

One main challenge is to find an expression describing thermal conversion into kinetic energy, so to derive the expression for the generated power (see, for example (Wei *et al.*, 2002)). To do this, denote by $p_c(t)$ ($p_h(t)$) the pressure of gas in the cold (hot) cylinder. Then, the instantaneous power absorbed by the compression piston is

$$W_c(t) = A_c p_c(t) u_c(t) \tag{4}$$

whereas the instantaneous power generated in the expansion piston is:

$$W_e(t) = A_h p_h(t) u_h(t) \tag{5}$$

So that the gross generated power is:

$$W_{mech}(t) = W_e(t) - W_c(t) \tag{6}$$

Notice that, above, it is assumed that the pressure is everywhere uniform (p(t)). However, when dealing with the computation of the power, one cannot neglect friction and therefore one should distinguish between the two pressures $p_c(t)$ and $p_h(t)$.

4. A SIMPLE MODEL OF THE PLANT

4.1 Working out a mean value model by a cisoid approach

The basic model has been implemented in a SIMULINK framework and several simulations trials have been performed. Such simulator requires a huge computational effort, mainly due to the stiffness of the system. Therefore the basic model is not only difficult to deal with in sampling and simulation, but it is also of no use for control design purposes. However, the performed simulations show that the oscillating variables have a sinusoidal-like behavior, with higher harmonics practically negligible, as it is also supported by experimental results (see (Bonnet *et al.*, 2002)). So we have explored the possibility of modelling the fast variables as if they were in a "cisoid regime". Indicating by $x_i(t)$ a generic fast variable, the following has been imposed:

$$x_i(t) = \bar{x}_i(t) + \tilde{x}_i(t) cos(\omega t + \varphi_{x_i}(t)) \tag{7}$$

As it can be seen, this corresponds to consider the fast variable as composed by a "slowly varying" average value $\bar{x}_i(t)$ plus a sinusoid with given frequency ($30Hz$) and "slowly varying" amplitude and phase $\tilde{x}_i(t)$ and $\varphi_{x_i}(t)$. Then, by analytic computations one can work out algebraic explicit expressions supplying the amplitude and phase $\tilde{x}_i(t)$ and $\varphi_{x_i}(t)$ in terms of the average values of the various fast variables $\{\bar{x}_j(t)\}$, as it will be illustrated in the subsequent point with reference to the pressure variable. Second, it is possible to derive a new set of differential equations where the derivatives of $\bar{x}_i(t)$ are given in terms of the variables $\{\bar{x}_j(t)\}$. In this way, the original basic model, consisting of 3 "slow" equations plus 24 "fast" equations (in the state variables $x_i(t)$), is replaced by a new system where the 3 slow equations are complemented by 24 new equations in the mean variables $\{\bar{x}_j(t)\}$. To such a system, a number of algebraic equations have to be added for the derivation of $\tilde{x}_i(t)$ and $\varphi_{x_i}(t)$ in terms of $\{\bar{x}_j(t)\}$. The remarkable advantage of such a model is that the dynamics of all variables are comparable, so that this model is effective for simulation and control design. Furthermore, the *mean value model* enables to evidence the effects of the geometry of the plant components and of the thermodynamical characteristics of the gas on the generated power. This is most useful in the early phase of the Dish Stirling solar plant design.

4.2 Pressure and power

By applying the cisoid approach to the pressure equations, taken as a significant case, the following expressions for $\tilde{p}$ and φ_p in terms of $\bar{p}$ can be derived.

$$\begin{cases} \varphi_p = arctan\left(\dfrac{-\frac{A_h \tilde{z}_h sin(\varphi_h)}{\tilde{R}\bar{T}_{h,g}}}{\frac{A_c \tilde{z}_c}{\tilde{R}\bar{T}_{c,g}} - \frac{A_h \tilde{z}_h sin(\varphi_h)}{\tilde{R}\bar{T}_{h,g}}} \right) \\ \tilde{p} = \dfrac{-\frac{A_h \tilde{z}_h sin(\varphi_h)}{\tilde{R}\bar{T}_{h,g}}}{\frac{M_{He}}{\bar{p}} sin(\varphi_p)} \bar{p} = K(\bar{T}_{gas}, \varphi_h)\bar{p} \end{cases} \tag{8}$$

where

$$M_{He} = A_c \bar{z}_{cm} \bar{\rho}_c + V_{reg,i} \sum_{i=1}^{10} \bar{\rho}_{h,i} + \\ + V_{ric} \bar{\rho}_{ric} + A_h \bar{z}_{hm} \bar{\rho}_h$$

Such results are useful to obtain a simplified formula for the mean power generated by the engine $\bar{W}_{mech}$. Precisely, in equations (4) and (5) one can adopt a cisoid approach for the fast variables p_h and p_c, to find out the following expression:

$$\bar{W}_{mech} = \frac{\omega}{2} A_h \tilde{p}_h \tilde{z}_h sin(\varphi_{ph} - \varphi_h) + \\ - \frac{\omega}{2} A_c \tilde{p}_c \tilde{z}_c sin(\varphi_{pc})$$

Note that $\tilde{p}_c$ and $\tilde{p}_h$ can be expressed in terms of $\tilde{p}$, which in turn is a function of $\bar{p}$, as shown in equation (8). In this way, one comes to the following expression:

$$\bar{W}_{mech} = \bar{W}_{mech,id} - \bar{W}_{mech,lam} - \bar{W}_{mech,turb}$$

where $\bar{W}_{mech,id}$ is the ideal power, generated as if the friction were absent, $\bar{W}_{mech,lam}$ ($\bar{W}_{mech,turb}$) is the loss in the generated power caused by the friction due to laminar flow (turbolent flow). As for the ideal power, one obtain:

$$\bar{W}_{mech,id} = \frac{1}{2}\frac{A_c\bar{z}_c A_h\bar{z}_h}{\tilde{R}}\frac{\bar{p}}{M_{He}}\omega\bar{p}\left(\frac{1}{\bar{T}_{c,g}} - \frac{1}{\bar{T}_{h,g}}\right) \quad (9)$$

where $A_c\bar{z}_c$ and $A_h\bar{z}_h$ are the volumes swept by the pistons inside the cylinders. This formula points out the dependence of the power upon the main variables of the plant. In particular, it shows how the power is determined by the mean gas pressure $\bar{p}$, the hot and the cold cylinder gas temperatures $\bar{T}_{h,g}$ and $\bar{T}_{c,g}$ and the geometric characteristics of the plant. A further analytical elaboration leads to the following equivalent expression:

$$\bar{W}_{mech,id} = \\ = \frac{1}{2}\frac{M_{He,swept,c}M_{He,swept,h}}{M_{He}}\omega\tilde{R}\left(\bar{T}_{h,g} - \bar{T}_{c,g}\right) \quad (10)$$

where $M_{He,swept,c}$ and $M_{He,swept,h}$ are the masses of Helium gas swept by the pistons in the cold cylinder and in the hot cylinder.

The two terms describing the losses in power due to friction effect are:

$$\bar{W}_{mech,lam} = \omega^2\left(k_{fl1}A_c^2\bar{z}_c^2 + k_{fl4}A_h^2\bar{z}_h^2\right) \quad (11)$$

$$\bar{W}_{mech,turb} = \frac{\omega^3}{2\tilde{R}}\bar{p}\left(\frac{k_{ft1}A_c^3\bar{z}_c^3}{\bar{T}_{c,g}} + \frac{k_{ft2}A_h^3\bar{z}_h^3}{\bar{T}_{h,g}}\right) \quad (12)$$

5. CONTROL SYSTEM

The engine can be seen as a controlled process as outlined in Figure 3. The input variables are the ingoing helium flow w_{in} and the outgoing helium flow w_{out}. The output variables are the gas mean pressure $\bar{p}$ the temperature $T_{rec,w}$ of the receiver, and of course the produced mechanical power $\bar{W}_{mech}$. Observe that, among the three output variables, only two are to be regulated, namely $\bar{p}$ and $T_{rec,w}$. As we intend the plant to supply energy to a power network, it is a common assumption to consider infinite load. Load variations are thus not consider. Therefore, the only disturbance that affect the process is insulation, which is measurable. Normally, the goal of the control system is to

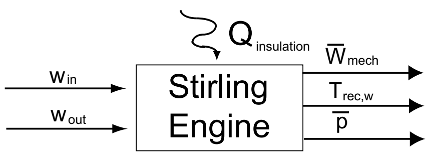

Figure 3. The Stirling Engine seen as a controlled process

keep the receiver's walls temperature ($T_{rec,w}$) constant at a given set point value.

The control scheme that has been applied is the cascade control system of Figure 4. (T_{ref}). The adopted

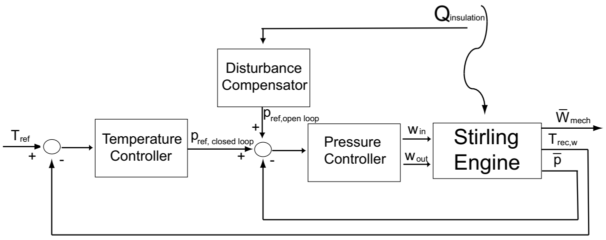

Figure 4. The Stirling Engine control system

rationale is that the inner loop is designed to control the helium mean pressure, with the pressure set point imposed by the outer loop. Moreover, being the disturbance measurable, the control system is also constituted by an insulation compensator. In conclusion, the pressure set point is given by the superposition of a reference pressure value decided by means of a look up table plus an additional term to deal with non standard situations, especially transient behaviors. The inner and the outer loop regulators are designed with a standard frequency domain approach.

6. SIMULATED MODEL VS REAL DATA

The overall model of the plant has been simulated in C language, in the Matlab SIMULINK environment. It consists of various submodels for the *thermodynamic part* (described by the mean value model above outlined), the *helium bottle*, the *cooling system*, the *induction motor*, *valves and transducers*, *controller*. Some heat exchange or geometric parameters have been numerically calculated analyzing the physics of the model and of the phenomena that take place in the engine (see, for example, (Thomas and Bolleber, 2000)). Some uncertain parameters have been tuned by comparing the simulation outcoming with the real data measured on the CESI Dish Stirling plant. Note that the CESI plant is equipped with a controller of unknown characteristics. The comparison between the so obtained simulator and the real plant has been made with reference to an interval of time with enough insulation as depicted in Figure 5. The comparison

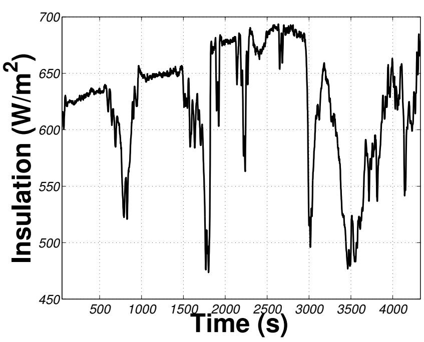

Figure 5. Insulation real data

between the simulated signals and the true ones is performed in terms of helium mean pressure (Figure 6) and produced power (figure 7), with very satisfactory results. In the Figure 8 the simulated nominal control

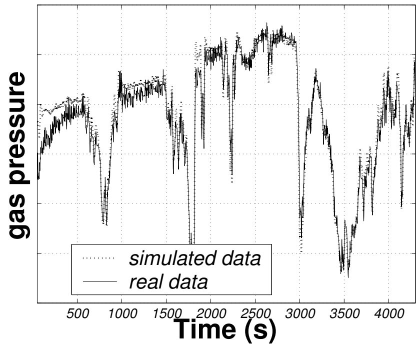

Figure 6. Simulated gas pressure versus real pressure of the Helium gas in the engine

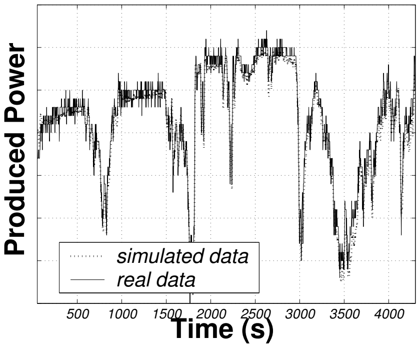

Figure 7. Simulated produced power versus really produced power by the plant

performances of the designed regulator is shown by the plot of the controlled receiver temperature T_{ric}.

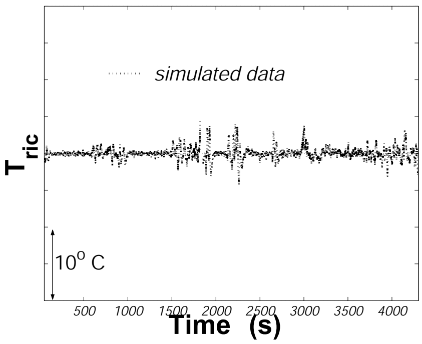

Figure 8. Simulated and real temperature of the receiver's walls

7. CONCLUSIONS

This paper discuss the main results obtained as far as the modelling of the Dish Stirling Solar Engine is concerned. While doing this, the model is simplified using steady state periodic behavior to remove stiffness introduced by fast dynamics that are not relevant in the time scale of interest. The so obtained model is suitable for numerical solution, to desume fundamental correlations that clarify the behavior of the considered engine, and to design a control system which could be successfully applied to the process. The overall control system has been extensively tested in simulation and comparisons of simulated results to real measurements are satisfactory. Experimental results are shown in the paper.

8. ACKNOWLEDGEMENTS

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