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# Control design model for a solar tower plant.

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**Abstract**

This paper deals with the development of a control design model for a 1MW Solar Tower equipped with a heat storage facility. This model is precise enough to achieve a good prediction of the responses but is also simple enough to avoid computational burden. The paper presents the assumptions and equations used for the different components of the plant. The behavior of the model developed in Matlab/Simulink[tm] is qualitatively validated by closed loop simulations. The control used for these simulations is also given. It consists of two levels, the upper level being an automaton whose outputs are the set points of the lower level controllers.


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## 1. Introduction

The development of an advanced control strategy is an efficient means to improve the maneuverability of the superheated steam Concentrated Solar Power plant e.g. the performance, the reliability, the plant life consumption and the safety. The main objective for the control is to track the power demand and to limit the thermal load experienced by the main components. The design and tuning of such a control system is however particularly difficult due to the complexity of the phenomena involved in this kind of plant. Among the major concerns are the non linearity and the coupling between the variables which make the system highly complex. To deal with such a process a model is thus mandatory to test and tune the control design before an on-site implementation.

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The references [2, 3] give a detailed model for the heliostat field and the boiler for the CSP Plant considered in the following. This model is very useful for the design of the plant and for performing some detailed analyses. Different from this, a simpler model is suggested in the present paper, which is specifically used to design the control system. For this purpose, intensive simulations are necessary, on the one hand to manipulate and parameterize an appropriate control structure and on the other hand to check if the process can be safely maintained by the control at the desired operation conditions under different scenarios. For instance, if a process variable (pressure, level, temperature) oscillates during a simulation, one can try to decrease the corresponding controller gains. The response obtained in simulation with the new gain may however be too sluggish and another control structure can be considered. This trial and error approach to design the control loops can be time consuming and a fast model is needed to be able to make a lot of runs. The model simplifications lead to a shorter simulation time and lessen the numerical difficulties so that intensive simulations can be done to adjust the control design. Moreover, the simple model can also be used in advanced control solutions such as a Model Predictive Control [1] for example which generally requires a lot of simulation to compute the control sequences.

**Nomenclature**

| | |
|---|---|
| $Cv$ | Valve characteristic coefficient |
| $DNI$ | Direct Normal Irradiation (W/m$^2$) |
| $E$ | Exchange efficiency |
| $L$ | Level (mm) |
| $M$ | Inertia parameter (kg) |
| $N$ | Number |
| $O$ | Opening |
| $Q$ | Heat Flux (W) |
| $S$ | Heat surface (m$^2$) |
| $T$ | Temperature (K) |
| $V$ | Volume (m$^3$) |
| $W$ | Electric Power (W) |
| $c$ | Heat capacity(J/K) |
| $k$ | Proportional gain |
| $h$ | Specific enthalpy (J/kg/°C) |
| $h$ | Heat coefficient (W/°C/m$^2$) |
| $m$ | Mass (kg) |
| $p$ | Pressure (Pa) |
| $q$ | Flow rate (kg/s) |
| $\alpha$ | Steam fraction |
| $\lambda$ | Fluid thermal conductivity (W/m/°C) |
| $\rho$ | Density (kg/m$^3$) |
| $\eta$ | Efficiency |

Subscripts

| | |
|---|---|
| $EV$ | Evaporator |
| $SH$ | Superheater |
| $c$ | Cold or convection |
| $cd$ | Condensation |
| $d$ | Desuperheater |
| $dc$ | Downcomer |
| $ev$ | Evaporation |
| $f$ | Feedwater |
| $h$ | Hot or heliostat |
| $l$ | Latent |
| $loss$ | Loss |
| $m$ | Massic or metal |
| $oil$ | Oil |
| $r$ | Riser |
| $s$ | Steam |
| $sat$ | Saturation |
| $sd$ | steam under water level in the drum |
| $t$ | Turbine or total |
| $v$ | Volumic |
| $w$ | Water |

Upperscripts

| | |
|---|---|
| $in$ | Inlet |
| $out$ | Outlet |

In the 1$^{st}$ section the paper presents the process with a field of heliostat, a solar tower with a steam boiler, a steam and water storage system, oil circuits used to store the heat and a steam turbine. The 2$^{nd}$ section presents the simplified equations used to describe the different components. The model of the boiler is based on the drum boiler model developed by Aström & Bell [4]. The steam accumulator model is described by a two phase system representing the condensation and the evaporation of steam and water and is based on the work of Stevanovic & al. [5]. The turbine is represented by a model that is classically used for control. Simple models for the superheater and desuperheater are also given. The implementation of the model in Matlab Simulink$^{tm}$ as well as the main control loops (level, pressure and temperature) are presented in the 3$^{rd}$ section. A two-level control structure is proposed to manage the storage system. The upper level is an event-driven state chart which computes set points for the lower-level continuous controllers. This upper controller is implemented using Stateflow$^{tm}$. The last part of the paper presents simulation results obtained with the model on a charge and discharge scenario for a solar flux variation. Different graphs show the response of the main parameters of the process. The conclusion points out the future developments.

## 2. Process description

The process considered in this paper is the Badaling power plant for which a detailed description can be found in [2, 3]. The plant described in Fig. 1 consists of a heliostat field, a receiver located at the top of the tower, a storage system and a power block. The steam is condensed in the condenser at the steam turbine exit and the water is sent to the deareator where the oxygen contained in the water is withdrawn. The feedwater pump then carries the water to the receiver located at the top of the tower. The receiver is composed of seven evaporator and four superheater panels. In Fig. 1 three equivalent heat exchangers, one evaporator and two superheaters separated by a desuperheater, are considered. A second desuperheater is located at the outlet of the receiver. The steam produced is sent directly to the power block through the direct valve VDirect or sent to the steam accumulator by opening the valve VStock. When the steam produced by the receiver is insufficient, the steam stored in the steam accumulator is released by opening the valve VDestock and heated by the auxiliary oil circuit. The superheated steam admitted by the turbine is controlled by opening the throttle VTur. A steam bypass valve VBypass is activated in case of high pressure.

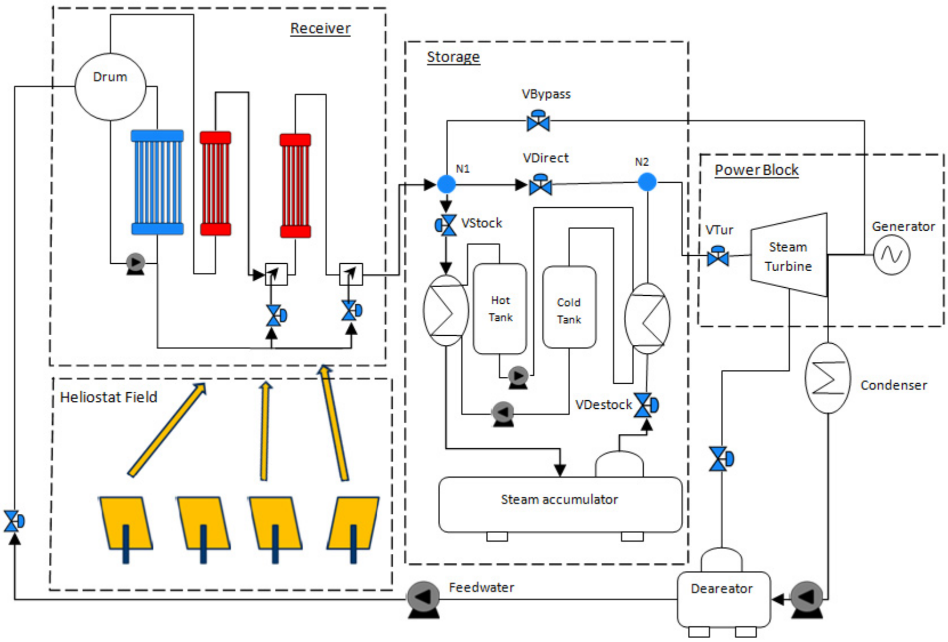


Fig. 1. Schematic diagram of the power plant.

## 3. Modeling

The plant is divided into four main components: the heliostat field, the receiver, the storage and the power block. In this section equations for each of the components which are implemented and connected in Matlab/Simulink$^{tm}$ are given.

Heliostat field. A very simplified model is considered for the solar field. The main reason for this assumption is that the angle positions of the heliostat are locally controlled by the tracking systems and are considered as perturbations for the rest of the plant. Each heliostat is supposed to be able to track the sun perfectly and to concentrate the solar rays inside the receiver. The total flux received by the receiver is supposed to be given by equation (1) where $k_h$ is a coefficient which depends on the field configuration (heliostat, solar position, etc.) and will be estimated with measurements in the real plant.

$$Q = k_h . N_h DNI \tag{1}$$

Evaporation loop (Fig. 2a). The model used is based on the work of Aström & Bell [4]. It represents the pressure and level of the drum. The basic equations are explained below; the complete derivation can be found in the original paper. The main assumptions adopted are: the mass steam fraction $\alpha_m$ is linearly distributed in the riser, which supposes that the heating is uniform; the metal temperature gradients $dT_m/dt$ are equal to the steam temperature gradients $dT_v/dt$; the flowrate in the downcomer is imposed by the circulation pump; the steam at the outlet of the drum is saturated dry steam. The steam and water properties are given by the IAPWS-IF97.

The mass and energy balance equations for the global system (2) are used to compute the volume of steam ($V_{st}$) and water ($V_{wt}$) in the loop. The mass and energy balance equations for the riser are used to compute the volume of water in the riser $(1-\overline{\alpha}_v)V_r$, where $\overline{\alpha}_v$ is the average volume steam fraction. Hence, the volume of water in the drum itself can be calculated knowing the total volume of water ($V_{wt}$) and the volume of water in downcomer. The volume of steam in the drum under the liquid surface ($V_{sd}$) is important to reproduce the shrink and swell effect that will be explained later in the control section. Formula for the condensation rate in the liquid phase $q_{cd}$ and for the steam flow rate through the water surface $q_{sd}$ as well as those giving the different vapor fractions are derived in [4]. The total volume of steam and water in the loop being constant, the independent state variables are the pressure $p$, the volumes $V_{wt}$, $V_{sd}$ and the mass steam fraction $\alpha_r$ giving a 4$^{th}$ order model for the evaporation loop.

$$\begin{aligned}
&\frac{d}{dt}\left[\rho_s V_{st} + \rho_w V_{wt}\right] = q_f - q_s \\
&\frac{d}{dt}\left[\rho_s h_s V_{st} + \rho_w h_w V_{wt} - pV_t + m_t c_m T_m\right] = q_f h_f - q_s h_s + Q_{EV} \\
&\frac{d}{dt}\left[\rho_s \overline{\alpha}_v V_r + \rho_w (1-\overline{\alpha}_v) V_r\right] = q_{dc} - q_r \\
&\frac{d}{dt}\left[\rho_s h_s \alpha_v V_r + \rho_w h_w (1-\overline{\alpha}_v) V_r - pV_r + m_r c_m T_m\right] = q_{dc} h_f - q_r (h_l + \alpha_r h_c) + Q_{EV} \\
&\frac{d}{dt}\left[\rho_s V_{sd}\right] = \alpha_r q_r - q_{cd} - q_{sd}
\end{aligned} \tag{2}$$

Superheater & desuperheater. The superheater model is given by equations (3). No steam accumulation is assumed; the flow is related to a static pressure loss between the inlet and outlet. To avoid implicit equations, $\rho_s$ is the steam density at the inlet. The temperature of the steam $T_v$ and of the metal $T_m$ are computed by energy balance equations. The convection coefficient $h_c$ is given by the Dittus-Bolter correlation defined for single phase heat transfer.

$$\begin{aligned}
&q_s = k_{SH}\sqrt{\rho_s\left(p_s^{in} - p_s^{out}\right)} \\
&V\frac{d\rho h_s}{dt} = q_s\left(h_s^{in} - h_s^{out}\right) + h_c S\left(T_m - T_v\right), \quad m_m c_m \frac{dT_m}{dt} = Q_{SH} - h_c S\left(T_m - T_v\right)
\end{aligned} \tag{3}$$

The desuperheater is described by equations (4); the steam flow rate at the outlet $q_s^{out}$ is the sum of the steam flow rate at the inlet $q_s^{in}$ and the water injection flow rate $q_d$; the output enthalpy is given by an energy balance equation where $M_d$ is an inertia parameter taking into account the steam and the metal contributions. The value of this inertia has to be estimated with measurements.

$$M_d \frac{dh_s}{dt} = q_s^{out}\left(h_s^{out} - h_s^{in}\right) + q_d h_d, \quad q_s^{out} = q_s^{in} + q_d \tag{4}$$

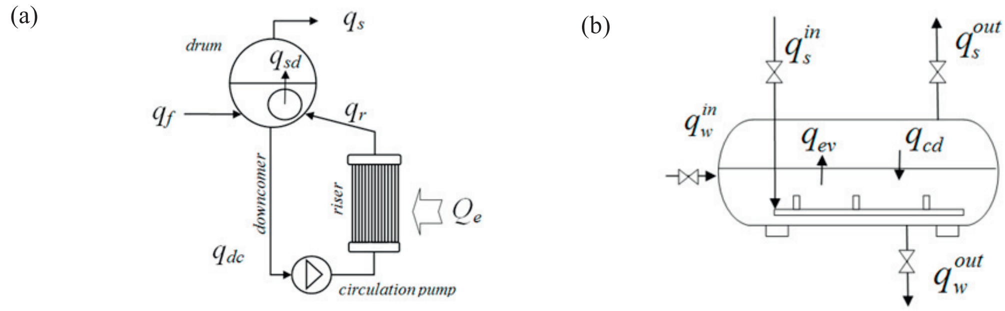


Fig. 2. (a) Evaporation Loop;(b) Steam Accumulator

Steam valve. The steam valves are modelled by static equations and isenthalpic expansion is assumed. The mass flow rate through the valve is given by (5) where $Cv$ is the valve characteristic provided by the vendor or estimated with on site measurements and depends on the opening of the valve $O$.

$$q_s = Cv(O)\sqrt{\rho_s\left(p_s^{in} - p_s^{out}\right)} \tag{5}$$

Pressure node. The pressure at a node of constant volume $V$ is given by mass and energy equations (6). The fluid is assumed to be superheated steam and a time varying density $\rho_s$ is considered. It is supposed that there is no flow inversion e.g. the steam flows from the inlet to the outlet. This can be assured in practice by check valves which allow the fluid to flow in only one direction. For the Matlab/Simulink[tm] implementation the equations are written with $p$ and $h_s$ instead of $\rho_s$ and $h_s$. The variable change implies a matrix inversion to obtain an ODE.

$$V\frac{d\rho_s}{dt} = \sum_{in} q_s^{in} - \sum_{out} q_s^{out}, \quad V\frac{d}{dt}\left[\rho_s h_s - p\right] = \sum_{in} q_s^{in} h_s^{in} - \sum_{out} q_s^{out} h_s^{out} \tag{6}$$

Heat exchanger. The NTU method is applied to compute the heat transfer in the oil and steam exchangers. The output temperatures are given by equations (7), where $E$ is the effectiveness, and $c_h.q_h$ and $c_c.q_c$ are the products of the flow and the specific heat for the hot and the cold fluids. The hot fluid is either the oil or the steam depending on the heat exchanger. The specific heat coefficients are mean value and may vary with the operating point.

$$\begin{aligned} Q &= E.\min(c_h \cdot q_h, c_c \cdot q_c).\left(T_h^{in} - T_c^{in}\right) \\ T_h^{out} &= -\frac{Q}{c_h.q_h} + T_h^{in}, \quad T_c^{out} = \frac{Q}{c_c.q_c} + T_c^{in} \end{aligned} \tag{7}$$

Oil Tank. It is assumed that there is no stratification in the tanks and that perfect mixing is achieved. The heat storage system consists of two oil tanks simply described by incompressible mass and energy balance equations (8). The oil level in the tank can be calculated knowing the mass of oil in the tank, the area and the oil density. The oil density and specific enthalpy are fitted with the discrete property specifications by 2nd order polynomials.

$$\frac{d}{dt}\left[M_t\right] = q_{oil}^{in} - q_{oil}^{out}, \quad M_t \frac{d}{dt}\left[h_{oil}^{out}\right] = q_{oil}^{in}.(h_{oil}^{in} - h_{oil}^{out}) - Q_{loss} \tag{8}$$

Steam accumulator (Fig. 2b). The steam accumulator equations are based on the work of Stevanovic & al [5]. It consists of two phases at the same pressure. $V_w$, $h_w$ are the volume and enthalpy of the liquid part and $V_s$, $h_s$, the volume and enthalpy of the steam part. The difference between the liquid and steam temperatures generates a heat flux $Q_c$ (9). The condensation and evaporation flow rates are given by empirical equations (10), where $\tau_c$ and $\tau_e$ can be identified with plant measurements. These evaporation and condensation flow rates also generate a heat transfer $Q_{evcd}$ (9). The volume of water and steam as well as the enthalpy are given by mass and balance equations (11). The state variables are $p$, $h_s$, $h_w$, $V_s$, $V_w$. As the total volume is constant, $V_s$ can be eliminated leading to a 4th order model.

$$Q_c = hS(T_s - T_w), \quad Q_{evcd} = q_{ev} h_{s_{sat}} - q_{cd} h_{w_{sat}} \tag{9}$$

$$q_{ev} = \max\left(\frac{\rho_w V_w}{\tau_e} \frac{h_w - h_{l_{sat}}}{h_{v_{sat}} - h_{l_{sat}}}, 0\right), q_{cd} = \max\left(\frac{\rho_s V_s}{\tau_c} \frac{h_{v_{sat}} - h_s}{h_{v_{sat}} - h_{l_{sat}}}, 0\right) \tag{10}$$

$$\begin{aligned} &\frac{d}{dt}\left[\rho_s V_s\right] = q_s^{in} - q_s^{out} - q_{cd} + q_{ev}, \quad \frac{d}{dt}\left[\left(\rho_s V_s h_s - V_s p\right)\right] = q_s^{in} h_s^{in} - q_s^{out} h_s^{out} + Q_{evcd} - Q_c \\ &\frac{d}{dt}\left[\rho_w V_w\right] = q_w^{in} - q_w^{out} + q_{cd} - q_{ev}, \quad \frac{d}{dt}\left[\left(\rho_w V_w h_w - V_w p\right)\right] = q_w^{in} h_w^{in} - q_w^{out} h_w^{out} - Q_{evcd} + Q_c \end{aligned} \tag{11}$$

Power Block. A simple model is used to compute the flow $q_s$ and the power $W_t$ delivered by the turbine (12); the flow is described by a critical expansion with parameters to be calibrated on site. A linear characteristic is assumed for the valve which supposes that the valve is tuned accordingly. The outlet enthalpy is a limit condition for the model and is supposed to be the enthalpy of the saturated steam at the condenser pressure.

$$q_s = \frac{k_{st} O_t p_s^{in}}{\sqrt{T_s^{in}}}, \quad W_t = \eta \cdot q_s \cdot \left(h_s^{in} - h_s^{out}\right) \tag{12}$$

## 4. Control system

In this paper a classical controller structure is specified to check the sizing of the plant and the performance of the model. The main objective for the control is to limit the variations in temperature and pressure of the receiver due to the solar flux perturbations on the one hand, and to limit the pressure variation at the turbine inlet on the other hand. In practice, the control systems for power plants are developed in a hierarchical way as in Fig. 3a which consists of several layers. At the lower level, not shown in Fig. 3a, are the actuator controllers (valve opening, pump speed, etc.) The dynamics of these local loops are in general fast. At a higher level of the hierarchy are the group controllers which apply to process variables like level, temperature, pressure by setting references for the actuator controllers. In Fig. 3a, group controllers are defined for the receiver (level, temperature), the turbine (pressure) and

the storage (pressure and oil flow rate). The heliostats are supposed to be independently controlled by tracking controllers and no automatic group functions are defined for them. The upper level is the unit control. It consists of a controller or an automaton that sends set-points to the different subsystems in order to accomplish general tasks (start-up, shut-down, load following, etc.).
Some definitions are given in Fig. 3. For continuous control (Fig. 3b), *P* is the process transfer function, *K* is the controller, *y* is the controlled variable, *u* is the manipulated variable, $y_r$ is the set point and *e* is the regulation error. For a discrete-event controller or automaton (Fig. 3c), *A* and *B* are state, $C_1$ and $C_2$ are transition conditions. The state chart representation is used in this paper because it is a very intuitive and readable language more suited for design. However automata can also be described by block diagrams like continuous controls but are in this format more difficult to read and maintain. Continuous and discrete-event control can be defined at each level of the hierarchy. For example an automaton can be used to control the *on/off* of a pump or to manage the whole unit for automated startup. In the following we describe the controllers developed for the subsystems and the unit. The PI controllers used in this paper and shown in the next figures have a transfer function $K(s) = K_p\ (1+I/s)$. Information on boiler control can be found in Gilman [6]. An important feature shown in Fig. 4a is the antiwindup system which stops the integral action when the controller is in a manual or saturated state

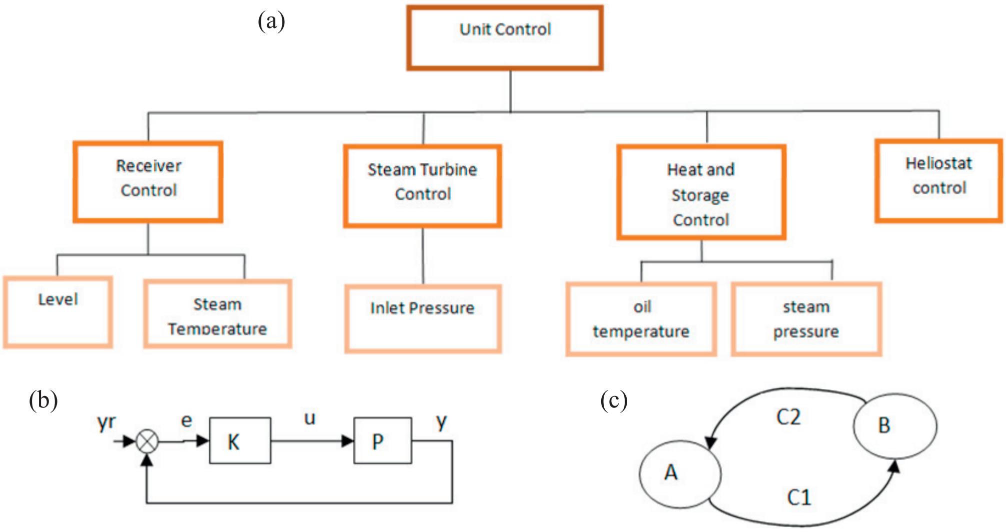


Fig. 3. (a) Hierarchical Control for CSP Plant; (b) Continuous Controller; (c) Discrete Controller (Automaton).

<u>Temperature Controller (Fig. 4b):</u> The controlled variable is the final temperature. The controller outputs are the water flow-rate set points for the desuperheaters. The adopted strategy is to use the first desuperheater in priority. When the flow-rate demand exceeds the capacity of the 1st desuperheater, the 2nd desuperheater is activated. Another strategy would be to use the final desuperheater in priority. A solution with activation of the two desuperheaters at the same time could be envisaged too but it would be more technically difficult. Indeed, a rule of thumb says that to avoid control problems such as hunting, the number of manipulated variables must be equal to the number of controlled variables, which is one.

<u>Level controller (Fig. 4c)</u>: the controller is a three-element controller. The steam flow going out of the receiver is compensated directly by a feed water variation. The PI level controller introduces a slow compensation to guarantee a null steady state error of the drum level. The control of the level can be tricky in the case of a pronounced shrink and swell effect which is taken into account by the model. Due to the variation of the vapor fraction under the water surface ($V_{sd}$), the level is going in the inverse direction during the first instants. For instance, the level decreases when water is added, or increases for an increase in steam demand. This kind of response designated as non-minimum phase, introduces severe limitations on the achievable performances and sometimes necessitates advanced control solutions.

Inlet and Drum Pressure Controller The pressure in the drum must be kept within bounds and its gradient must be limited to avoid thermal fatigue of the thick wall components. We define a pressure controller for the drum and for the turbine inlet (Fig. 4d). These pressure controllers are active depending on the state of the power plant. For instance, when the storage is not on line, the pressures in the drum and at the turbine inlet are strongly coupled and it is better to control only one pressure leaving the other one free. In this case the control for the sliding pressure is put on manual mode by setting the Auto/Manu (AM_Tur, AM_Direct, etc.) and the output value (VTur, VDirect, etc.).

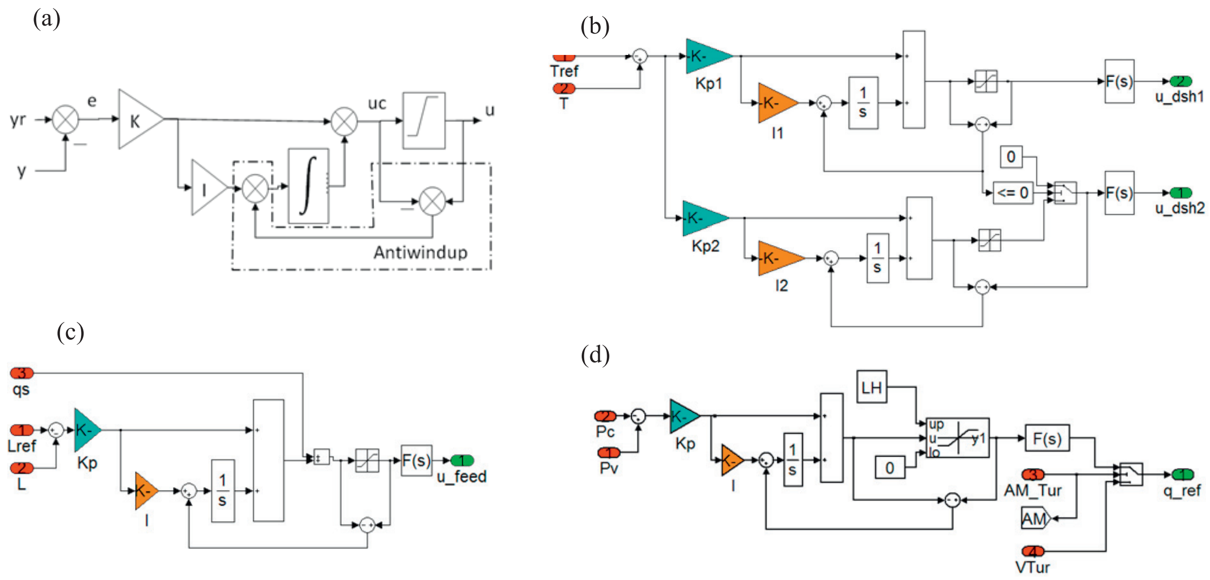


Fig. 4. (a) PI controller with antiwindup (b) Temperature Control (c) Level Controller (d) Inlet Pressure Controller

Unit controller. The unit controller developed in the paper (shown Fig. 5) defines six operating modes. DIRECT, CHARGE, DISCHARGE, BYPASS, TURBINE DOWN, STOP. A mode INIT is created to initialize the model from scratch. When the simulator is stabilized after a transient, the steady state is saved and can be used as a starting point for the next simulations.

In the DIRECT configuration, the drum pressure is not directly controlled but the throttle controls the turbine Inlet Pressure which is linked to the drum pressure by the pressure drop. When the pressure is too high (condition [Pv>=3200] in the Stateflow[tm] model Fig. 5), the unit control goes to the CHARGE mode; the pressure controller opens the charging valve in the accumulator. When the accumulator is full (condition [Paccu > 2700]) and the pressure still high, the bypass is opened (mode BYPASS) until the operator reduces the flux by withdrawing heliostats; then the system goes back to the DIRECT mode. When the pressure is too low [Pv<2800] or the temperature too low [Tv_s<Tvmin], the system goes to the DISCHARGE mode in order to decrease the flow rate out of the receiver; the discharge valve (VDestock) is opened, the pressure in the drum being controlled by the direct valve (VDirect) and the turbine inlet pressure being controlled by the throttle (VTur). When the solar flux is again high enough (clouds disappear or additional heliostats projected), the pressure in the receiver increases. The system reverts back to the DIRECT mode as soon as the pressure reaches a sufficient value [Pv> 3000]. If the solar flux remains low and the steam accumulator is empty, the unit goes to the TURBINE_DOWN mode where the heat continues to be stored in the steam accumulator until full or until the operator decides to stop the plant.

## 5. Simulation results

The simulation scenario is a fluctuation of the solar flux received by the receiver (Fig. 6). The initial power value is about 6 MW and the control is in DIRECT mode. Then the flux increases to around 9 MW. The control switches to the STORAGE mode and then the BYPASS mode when the accumulator is full. After 83 min the flux decreases to about 3 MW for seven minutes. The control switches to DISCHARGE mode decreasing the accumulator pressure. The flux is then maintained at high level until the time 167 min, with the control unit on BYPASS mode until the solar flux returns to the lower level for a longer period. The unit control switches to the DISCHARGE mode until the hot oil tank reaches its lower limit at 237 min. The turbine is then shut down and the heat is stored until the accumulator reaches the high pressure limit (TURBINE_DOWN mode) at 266 min. At this point the plant is stopped and the heliostats put in the upright position. During this scenario the simulated pressure of the drum is kept between 28 and 32 bar and no severe fluctuations are observed during the short flux variation. The turbine flow is maintained at around 2kg/s during the fluctuation with a limited impact of the solar flux on the inlet pressure and power. A fluctuation can be seen when the discharge valve is opened. The optimization of the control loops (optimization of the transition by coordinated control) may solve this problem. For longer periods with low sunshine, the drum pressure is maintained and the turbine inlet pressure is slowly decreased following the accumulator pressure until the turbine shuts down (at 237 min). The steam temperature peak value (Fig. 6) is limited which shows that the risk of creep is mitigated. However there exist rapid variations for instance at 100 min and 230 min. These fast gradients do not last long and hence may not excessively consume the life of the thick-walled components. The consequences of these variations have to be studied in detail and a control adaptation will be done if the impacts prove to be a concern.

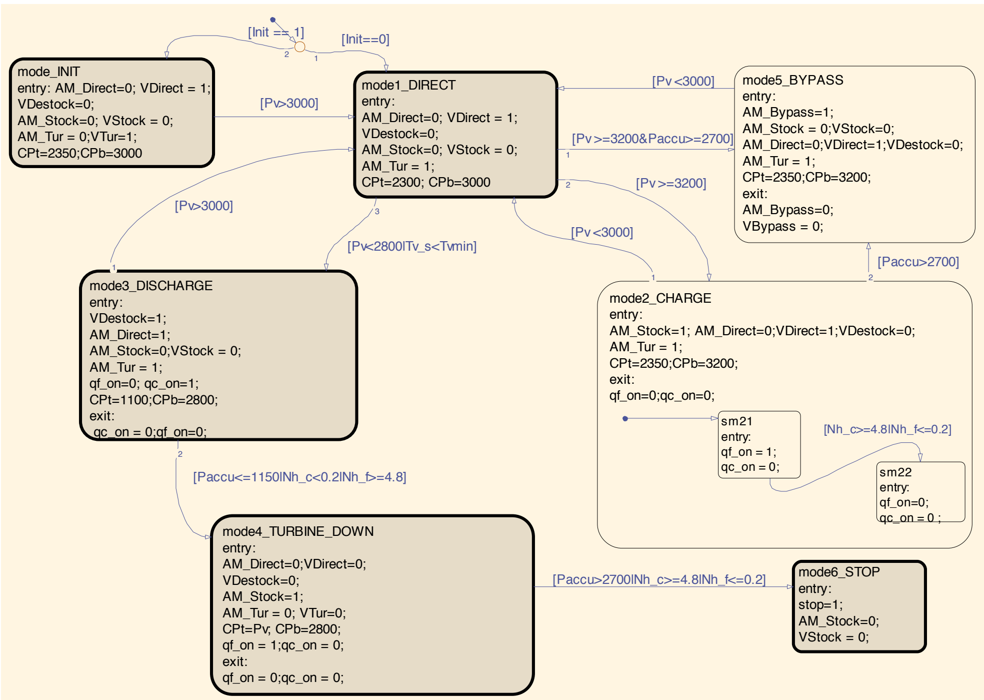


Fig. 5. Stateflowtm Unit Control (nota: the pressures are in kPa)

## 6. Conclusion

A simplified physical model is developed to design the control loops. The model is suited to test control solutions because the simulation time is short (in average 10 times faster than real time) which allows interactive simulations. The model can also be used for online optimization in an MPC control strategy. For highly complex systems such as CSP plant, a physical model is easier to tune than the black box transfer functions also used in MPC control, because the parameters can be quantified by physical dimensions such as mass, volume, etc. and few functioning parameters such as valve characteristics, inertia, time constants. The model is characterized by low order equations (4$^{th}$ order for the evaporation loop, 2nd order for the superheater, 4th order for the steam accumulator) leading to easier simulations. The price to pay is limited precision and parameters which have to be identified with on site measurements.

The control solution developed in the paper will be refined. An extension will be envisaged by integrating the heliostats in the unit control. The next steps will be the construction of a simplified heliostat field model, extended validation of the process model with on site experiments, fine tuning of the local controllers and of the unit control with model-based advanced solutions and implementation of the control solutions on a DCS. The procedure envisaged to validate the model is to apply step variations on each control input on the real process. The parameters which correspond to empirical equations will be tuned to fit the measured data. If the gap between the model and the measurements is too big, a model adaptation or new tests may be done depending on the analysis results. The model will be considered valid when the model error is compatible with the desired control objectives. Some comparisons will also be made with detailed models to assess the validity of the control design model.

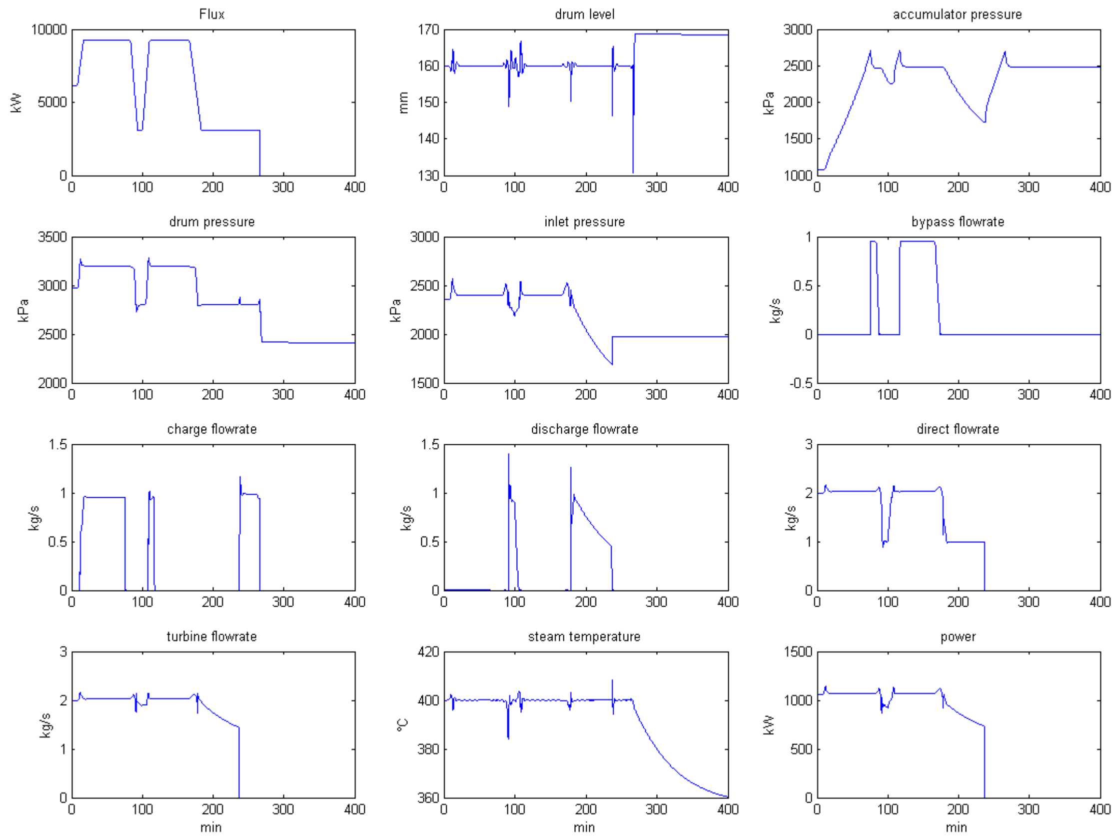


Fig. 6 Simulation results

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