Fast Start-up of a Combined-Cycle Power Plant: a Simulation Study with Modelica

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Abstract

The paper deals with the modelling and simulation of fast start-up transients of a combined-cycle power plant. The study is aimed at reducing the start-up time while keeping the life-time consumption of the more critically stressed components under control. The structure of the model, based on the ThermoPower library, and the main modelling assumptions are illustrated. Selected simulation results are included and discussed.[1]

1 Introduction

The on-going process of deregulation on the electrical power grids throughout Europe demands for more aggressive operation policies for existing and future power plants. Faster start-up and load change transients can be beneficial to remain competitive on an increasingly open power market. In this context, the present work is aimed at understanding how to improve the current start-up procedures for the typical combined-cycle power plant installed on the Italian grid.

For the purposes of the present study, the plant model must have the following features:

- be able to represent the whole start-up procedure, including boiler start-up, turbine start-up, and load pick-up;
- include a model of thermal stresses in critical components, which pose a lower bound to the start-up time;
- include a simplified model of the plant control system;
- neglect phenomena and components which are not critical for the start-up phase, in order to keep the model complexity at a reasonable level.

The plant model, based on the ThermoPower library [1]-[3], has been parametrised with design and operating data from a typical unit, and validated by replicating a real start-up transient, as recorded by the plant DCS. The model has then been used to test faster start-up manoeuvres, with the objective of either reducing the plant life-time consumption at equal start-up times, or reducing the start-up time at the same level of plant life-time consumption. This study has been carried out by trial-and-error, but the long term goal is to couple the model (or a suitably simplified version of it) to state-of-the-art optimisation software, to compute the optimal transients.

2 The plant model

The plant under investigation is composed by a 250 MWe gas turbine unit (GT), coupled to a heat recovery steam generator (HRSG) with 3 levels of pressure, driving a 130 MWe steam turbine (ST) group. The limiting factors to a reduction of the start-up time are:

- the maximum load change rate of the gas turbine;
- the thermal stress in thick components (in particular, the steam turbine shafts);
- the ability of the control system to keep their controlled variables within the allowable limits.

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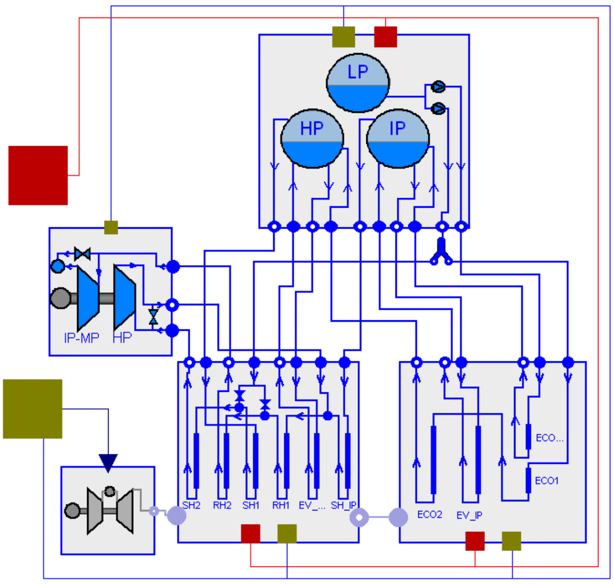

Fig. 1: The plant model at the system level.

At the system level, a detailed representation of all the parts of the plant working with low-temperature fluids is not required, since they are not critical, as far as their control and their thermal stresses are concerned. Therefore, the low-pressure part of the HRSG, the condenser and the feed-water system will be represented in an extremely simplified way. The plant model is then obtained (Fig. 1) by connecting the models of the GT unit, HRSG unit (divided into three parts for convenience), and ST unit via thermo-hydraulic connectors. Sensor and actuator signals are collected from/to each unit by means of expandable connectors.

It is well-known that the HRSG start-up (several hours) is much slower than the GT start-up time (around 20 minutes). It is then possible to describe the GT unit in a highly idealised fashion, i.e. as an ideal source of hot flue gases, whose temperature and flow rate is prescribed as a function of the load level; the maximum load change rate is given by the unit specification, and is not a subject of the present study.

The steam turbine unit model (Fig. 2) is instead rather detailed, in order to correctly describe the various phases of the start-up transient. The high pressure turbine (HP) and intermediate-plus-low pressure turbine (IP) are modelled, as well as the turbine bypass circuits; the contribution of the low-pressure steam generator is instead neglected.

The most critical part of the plant, as far as the thermal stresses are concerned, is the turbine shaft in contact with the highest temperature steam, i.e. downstream of the first (impulse) stage of both turbines, which is then represented separately, and connected to a thermal stress model of the shaft section. The stress model contains a thermal model, based on Fourier's equation discretised by finite differences, to represent the radial distribution of the temperature; the thermal stress on the outer surface (which is the most heavily stressed part) is then computed as a function of the difference between the surface temperature and the mean temperature. The generator inertia and a simplified model of the connection to the grid complete the unit; a small torque is added in order to avoid the stopping of the steam turbine, which would lead to various model singularities.

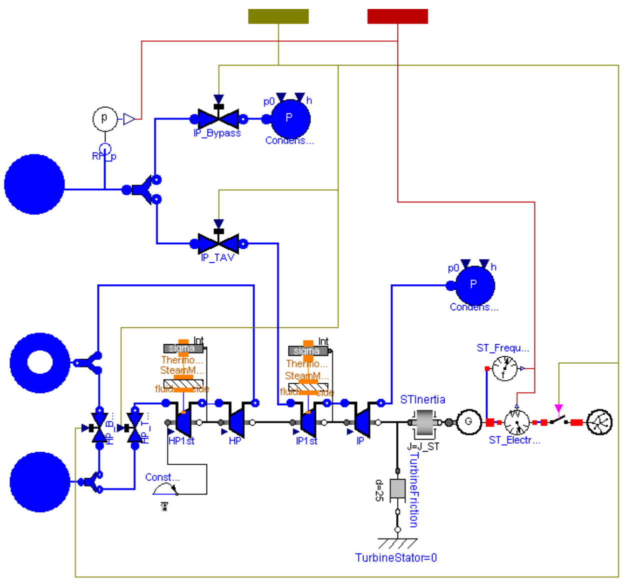

Fig. 2: The steam turbine unit model.

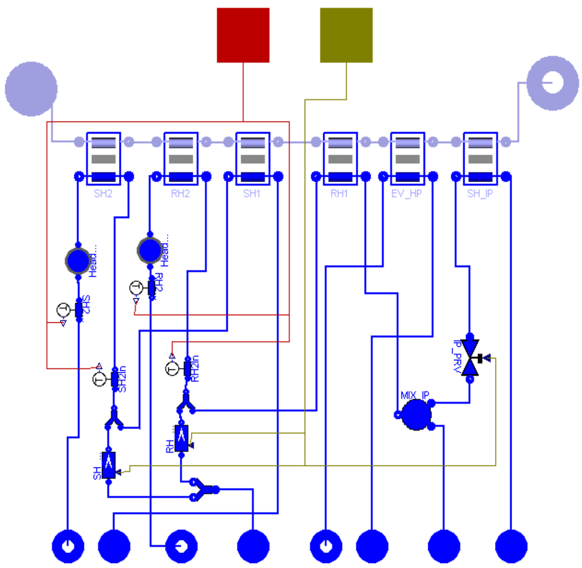

Fig. 3a: Heat exchanger units in the HRSG.

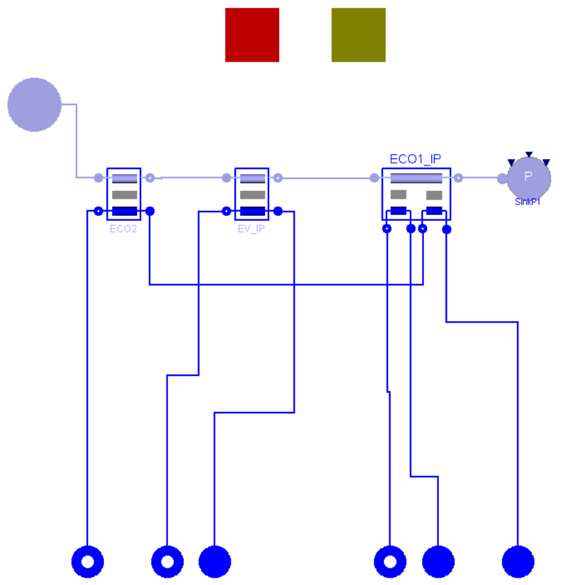

Fig. 3b: Heat exchanger units in the HRSG.

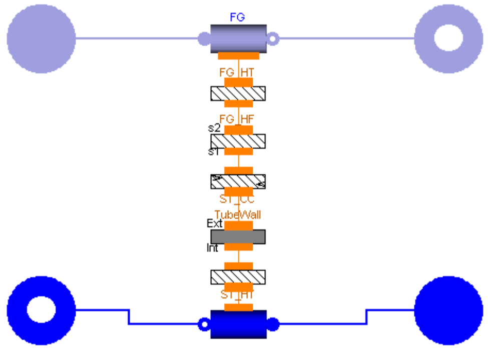

Fig. 4: A single, generic heat exchanger model.

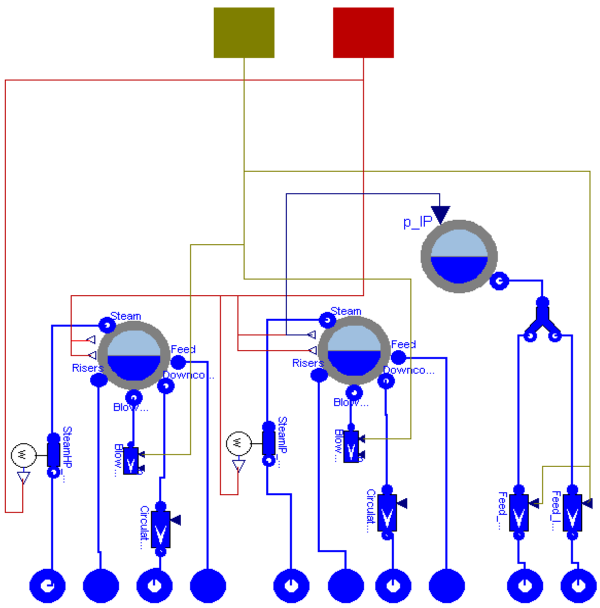

Fig. 5: The drum unit.

The heat exchangers along the flue gas path, i.e. the economisers, evaporators and superheaters, as well as the IP mixer and attemperators, are contained in two units for convenience (Fig. 3a-b). The structure of the each heat exchanger (Fig. 4) includes finite-volume models of the flue gas and water/steam side, as well as of the fluid-wall heat transfer, and of the wall thermal inertia. Since there is no draft control in the flue gas path, the associated dynamics is negligible; therefore, to reduce the number of states of the model, the flue gas side model is quasi-static. Note that the fluid side model is replaceable: a `Flow1D` model is used for the economisers and superheaters, while a `Flow1D2ph` model is used to describe the 2-phase flow in the evaporators.

The plant model is completed by the models of the boiler drums (Fig. 5). Since we are not interested in the high-frequency pressure dynamics, the high-pressure (HP) and intermediate pressure (IP) drums are based on mass and energy balances assuming thermal equilibrium between the two phases. The low-pressure part of the HRSG is neglected, so that an idealised model of the low-pressure drum is only needed as a boundary condition for the IP and HP circuits, i.e. to connect the inlets of the corresponding feed-water pumps (Fig. 5, on the far right). The LP drum pressure (and thus temperature) is determined as a function of the IP drum pressure, tuned from operational data.

3 Control system model

Given the type of plant, and the modelling assumptions, the control system can be hierarchically split into two levels.

3.1 Low level controllers

The lower level is quite straightforward, and is not the subject of the optimization. It contains five independent PI/PID loops, controlling:

- the HP and IP drum levels, using the corresponding feed-water flows;
- the HP steam pressure, using the HP turbine bypass valve;
- the IP steam pressure, using an intermediate valve at the outlet of the IP superheaters, before the mixing with the HP turbine exhaust;
- the reheater steam pressure, using the IP turbine bypass valve.

Note that the HP pressure controller is only active during the initial phase of the plant start-up, when the turbine admission valve (TAV) is closed, and the steam is dumped to the condenser. Once the steam turbine generator has been synchronised with the grid, the TAV is opened, the pressure diminishes, and the pressure controller reacts, eventually closing the bypass valve completely. The (continuous-time) PID controller model must then be able to operate correctly under saturation, providing suitable anti-windup action.

3.2 Supervisory control

The supervisory control level determines the actual start-up transient by acting on the following variables:

- GT load request;
- TAV opening (both HP and IP) *or* turbine speed set point;
- pressure controllers set points (HP and IP);
- generator-grid breaker;
- drum blow-down flow rates.

During the start-up transient, all of these variables are operated in an open-loop fashion, according to a pre-determined schedule which is the subject of the optimisation. The only exception is the TAV opening, which is determined in closed loop by a speed controller during the turbine start-up transient phase: in that case, a PI controller drives the TAV, and the scheduling variable is the speed set-point. It is therefore necessary that the PI controller provides a tracking mode as well, to manage the transitions between the off-duty, start-up and connected modes of operation of the turbo-generator in a correct fashion.

4 Model parametrisation and validation of the reference transient

The physical parameters of the model (dimensional data and operating points) have been set to match those of a real unit. Some data are known directly (e.g. number and dimensions of the tubes in the heat exchangers), other (e.g. the heat transfer coefficients or the valve and turbine flow coefficients) are computed from operating point design data.

The low-level controllers have been tuned in order to provide satisfactory performance (fast enough response with no significant oscillations and control overshoot).

The direct initialisation of the plant model in the shut-off state is numerically hard, due to the presence of low or zero flow rates and to the lack of knowledge of good guesses for the initial values. Therefore, the model has been initialised near the full-load steady state by setting the `start` attributes of the state variables (pressures, temperatures, flow rates, turbine speed, controller states), then brought to the full load steady state by running a stabilisation transient. The use of variable step-size integration algorithms allow to perform this task in a reasonable time (less than 10 seconds, CPU time). The steady-state values of the variables of interest (pressures, temperatures, flow rates, powers) are correct by construction, as the model has been parametrised using those same values. Incidentally, an attempt was performed to get the steady state directly by using initial equations, but the non-linear solver failed to converge, probably due to bad selection of the start values for some iteration variables.

A plant shut-down transient was then performed, bringing the model to a state corresponding to the warm start-up of the plant:

- steam turbines with no steam flow and (almost) at standstill.
- pressures around 1 bar in both the HP and IP circuit.
- GT “almost” shut down (a small flow rate of warm flue gas is kept to avoid singularities in the flue gas side model).

The temperature distribution of the turbine shafts was then reinitialized to the desired initial value (corresponding to 180 °C). This constitutes the initial state for the start-up transient simulation.

A reference simulation was then performed to replicate the recording of an actual start-up transient, which was replicated with acceptable fidelity, as far as the measured variables are concerned. Note that the study is not targeted to a specific plant, but rather to a whole class of similar plants, so that a high accuracy is actually not needed. Some results are reported here to give an idea of the degree of complexity of the transient.

Fig. 6 reports the net electrical power outputs of the GT, steam turbine (ST) and total. During the first 5000 s, the GT is running idle, so that there is no net electrical power output, but a certain flow of hot exhaust gases is already available to start up the steam generator. The steam turbine is then started and synchronised, and at time t = 13500 s the steam turbine starts picking up steam, while the GT load is increased up to the maximum.

Fig. 7 shows the pressure in the HP and IP drums. The steam generator start-up is split into two phases, where both pressures are controlled; during the load pick-up phase, instead, the HP circuit operates in sliding pressure mode, to avoid reducing the turbine efficiency due to throttling.

Fig. 8 shows the steam production rates (HP in blue, IP in green), as well as the flow rate through the HP turbine (in red). Until t = 9600 s all the steam is dumped to the condenser; subsequently, a small amount is used to accelerate the turbine (see the turbine speed, Fig. 9), and only after the load pick-up has started the TAV are fully opened, sending all the steam into the turbine. Fig. 10 shows the interplay between the TAV and the bypass valves; the former are gradually opened during the transient, while the latter are closed by the pressure controllers.

All the transients shown so far (except the last) correspond to actual measurements taken on the plant.

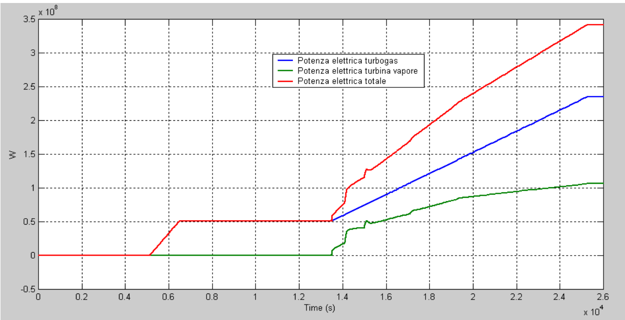

Fig. 6: Net electrical power outputs.

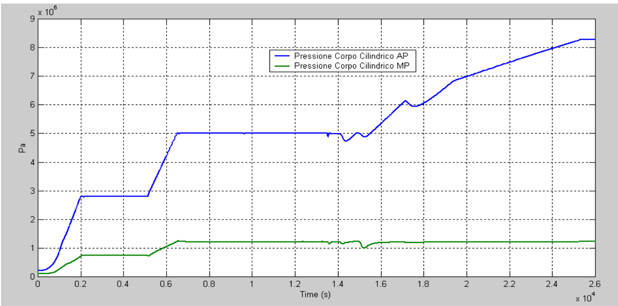

Fig. 7: HP and IP drum pressures

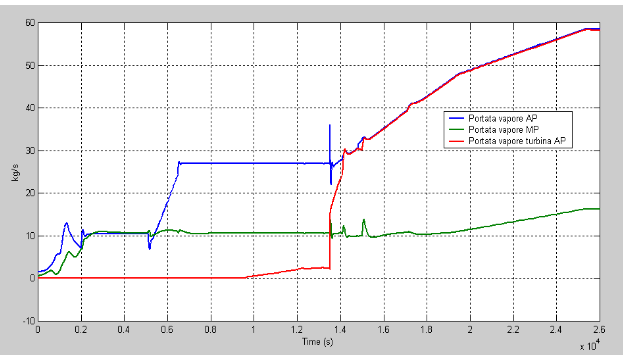

Fig. 8: Steam flow rates.

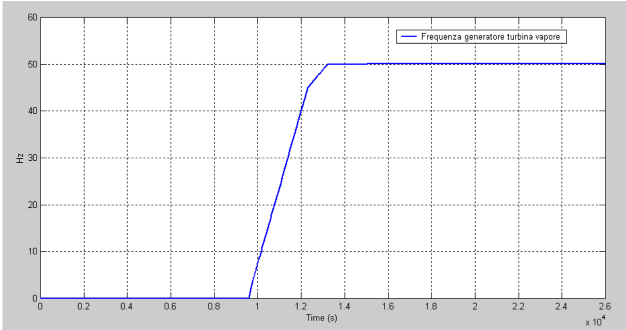

Fig. 9: Turbine speed.

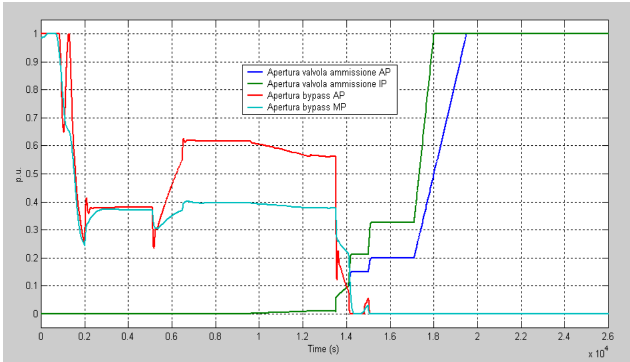

Fig. 10: Valve openings.

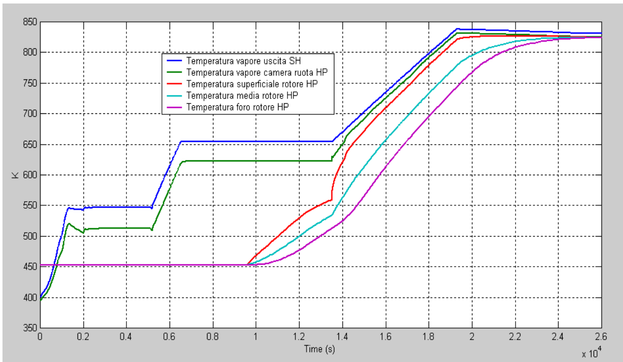

Fig. 11: HP turbine steam and rotor temperatures

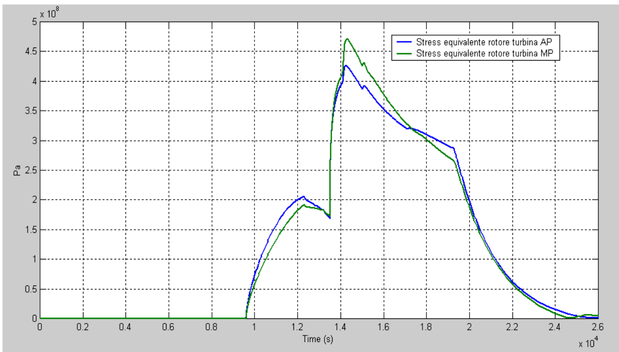

Fig. 12: HP and IP turbine rotor thermal stress.

The most interesting part of the simulation concerns the temperature distribution in the first section of the steam turbine shafts, and the corresponding thermal stress, which results in component fatigue and thus limits its useful lifetime. Fig. 11 shows the temperatures for the HP turbine: the three lower curves represent the internal, mean and external rotor temperatures, while the two upper curves represent the temperature of the superheated steam at the turbine inlet, and the (slightly lower) temperature of the steam downstream the nozzles of the first stage, which is where the steam comes into contact with the rotor. During the turbine start-up, the steam flow is very low, and so is the heat transfer coefficient; therefore, the rotor temperatures increase slowly; when the TAV are opened more aggressively, the external temperature gets much closer to the steam temperature. Note that, at time t = 19200 s, the steam temperature tops at 830 K; this is due to the GT exhaust temperature control, which keeps the hot flue gases at constant temperature for loads higher than 60%, by acting on the inlet guide vanes. At the end of the transient (steady state), all temperatures are equal, since there is no steady-state heat flow.

The corresponding thermal stresses at the rotor surface (for both HP and IP turbines) are shown in fig. 12. The peak values, which actually determine the lifetime consumption over the start-up cycle, are around 450 MPa, which is consistent with typical values estimated on the real plant during warm start-up transients. Note that this peak is reached at the beginning of the load pick-up phase.

5 Improving the start-up transient

The analysis of the reference transient shows that the current start-up procedure is quite conservative. There are a number of intermediate stops, which are not needed from a physical point of view, and have probably been provided to allow for ample margin against unexpected problems when starting up the plant. The current practice was in fact conceived when the plant was run in a vertically integrated context, which placed more emphasis on safety and availability rather than on efficiency and economy of operation. If those stops are completely eliminated, the corresponding transient can be run without any problems for the control system, resulting in a reduction of the start-up time from 25300 to 19200 s, and in fuel savings corresponding to the production of 208 MWh at full load (i.e. maximum efficiency). The same peak levels of stress are obtained. The details are omitted for the sake of brevity.

The "theoretical" minimum start-up time was then sought in two complementary ways:

a) minimising the start-up time subject to the constraint of getting the same stress peak (and thus lifetime consumption) of the reference transient;

b) reducing the stress peak (and thus increasing the lifetime consumption), without increasing the start-up time with respect to the reference transient.

To reach the first goal, it is essential to note that the stress peak is basically due to the thermal shock at the beginning of the start-up phase. Once this peak has been hit, the lifetime consumption is the same regardless how fast the stress goes back to zero. Therefore, the GT load pick-up rate has been increased from 1 MW/min to 1.5 MW/min, in order to keep the stress transient flat (see fig. 15). Furthermore, once the GT load has reached 60%, the flue gas temperature does not increase further, so that also the superheated steam temperature will not increase substantially (it will actually decrease a little bit, as the steam flow rate increases), and the stress will decrease no matter how fast the load goes up.

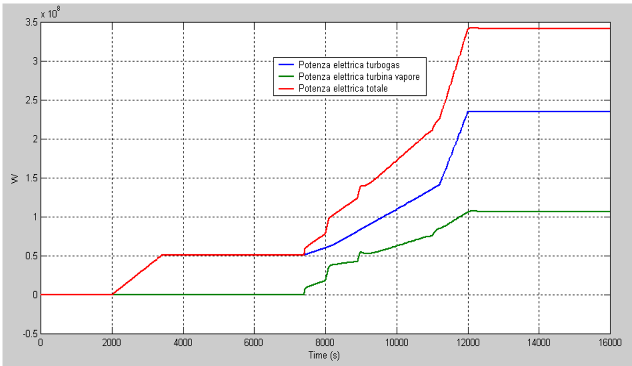

Fig. 13: Net electrical power outputs, fast start-up

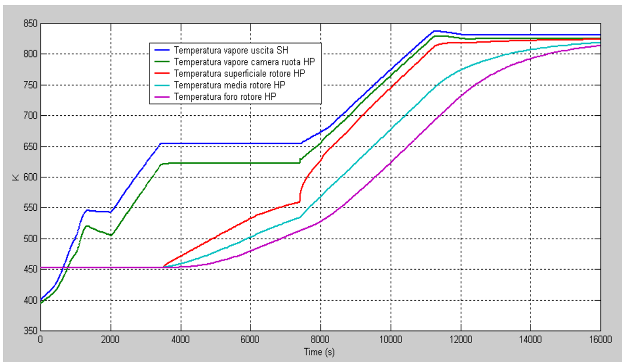

Fig. 14: HP turbine rotor and steam temperatures, fast start-up

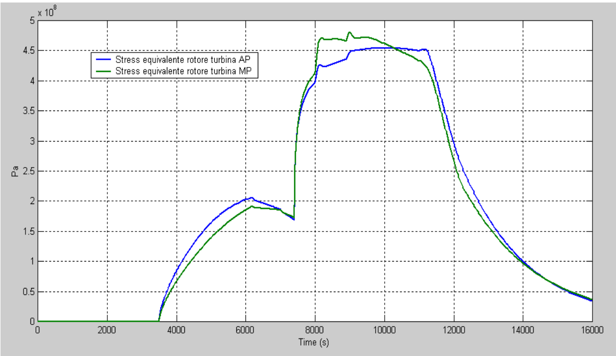

Fig. 15: HP and IP turbine rotor stress, fast start-up

Fig. 13 represents the net power outputs, while Fig. 14 and 15 represent the corresponding temperature profiles on the HP turbine rotor, and the corresponding thermal stresses on both turbines, respectively. It is possible to compare these figures with Fig. 6, 11, and 12. The peak stress is still around 450 MPa. The transients of all the other variables are similar to those already shown for the reference case, showing no particular problem as to the plant control.

Compared with the reference transient, the start-up time is reduced from 25300 s to 12500 s, and the fuel savings correspond now to 242 MWh at full plant load.

To reach the second goal (i.e. reduce the stress peak), it is necessary to reduce the thermal shock at the beginning of the steam turbine load pick-up. This can be obtained by allowing for a longer soak time for the turbine, i.e. once the turbine has reached full speed, it is kept running at no load so that the steam flow can heat up the rotor a little bit more. The turbine start-up is therefore initiated earlier than in the reference transient, and the turbine is kept idling for 3900 s. The load pick-up phase is then started, and the rate of change is adjusted to obtain a flat stress curve. Once the 60% level has been reached, the rate of change is increased to 7 MW/min, as in the previous case.

The resulting temperature and stress plots are shown in Fig. 16 and 17. In this case, two stress peaks are obtained (each one causing fatigue and thus lifetime consumption); however, the first peak value, around 200MPa, is only slightly higher than the limit of elastic behaviour (170 MPa), and thus correspond to a very low additional lifetime consumption, while the second, around 320 MPa, is well below the previous value of 450 MPa. The start-up time is reduced from 25300 s to 17500 s, while the fuel savings correspond to 114 MW/h at full plant load.

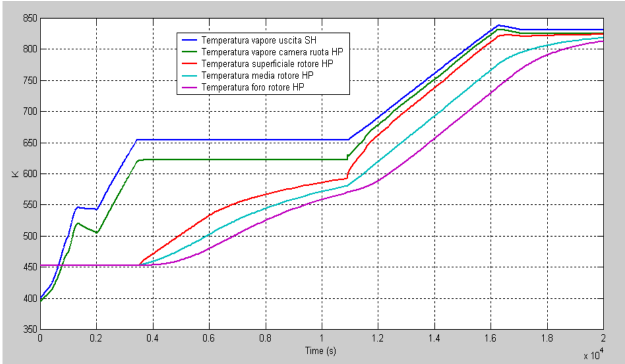

Fig. 16: HP steam and rotor temperatures, soaking.

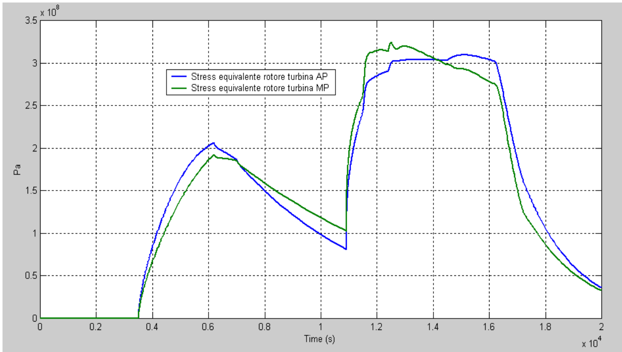

Fig. 17: HP and IP turbine rotor stress, soaking.

6 Conclusions and future work

The optimisation of the start-up procedure for a combined-cycle power plant has been studied, by means of a system simulator. The plant model was developed in Modelica using components from the ThermoPower library; the low-level control system model is based on continuous time PID controllers with anti-windup and tracking features. Compared to traditional simulation environments, it was relatively easy to customise the degree of detail of the model, both by writing extremely simplified component models where possible, and by developing ad-hoc models for the estimation of the thermal stresses in the steam turbines shafts. The final model has around 140 states and several thousands algebraic variables.

The simulation study was conducted using the Dymola [4] simulation tool, which allowed to compute the whole simulation transient in times around 400 s on a Pentium 3 GHz CPU.

The study is a first step towards the realisation of more simplified models, to be validated against the reference one, which will be employed together with optimisation software to automatically compute the optimal transients. A further step could then be the design of a closed-loop model-based control system, capable of attaining similar performance in real time and in the presence of uncertainty.

7 References

[1] F. Casella, A. Leva, "Modelling of Thermo-Hydraulic Power Generation Processes using Modelica", *Mathematical and Computer Modelling of Dynamical Systems,* v. 12, n. 1, pp. 19-33, 2006.

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[3] ThermoPower library home page, URL: http://www.elet.polimi.it/upload/casella/thermopower/

[4] Dynasim AB, Dymola v. 5.3.