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Simulation of a Power Regulation System for Steam Power Plants

Lorenzo Damiani*, Alessandro Pini Prato

DIME, University of Genoa, Via Montallegro 1, 16145 Genoa Italy

Abstract

Renewable energy sources, presently constituting about 23% of the total Italian power production, are featured by very discontinuous supply during the day that, to avoid grid malfunctions, must be compensated by fossil fuelled power plants. The latter must hence be able to rapidly control power supply.
This paper proposes a power regulation system for coal power plants, consisting in the bypass of the low pressure pre-heaters in order to increase the steam flow-rate in turbine. The main advantage of this system is the limited thermo-mechanical stress induced in the pre-heaters. The solution effectiveness is investigated through a Matlab-Simulink model.

Keywords: Steam power plant, Ultra Super Critical, simulation model, power control system, grid code.

1. Introduction

In the field of electric power production, the last years have been characterized by an increasing interest in the renewable energy sources, both for the concern about fossil fuels depletion and for the problems connected to pollutant emissions.

Nomenclature

A	Turbine passage area [m^2]
c_p	Specific heat capacity [kJ/kg K]
E_r	Equivalent in water [kW K]

* Corresponding author. Tel.: +39-0103532549; fax: +39-0103532566.
E-mail address: Lorenzo.Damiani@unige.it

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H	Enthalpy of steam [kJ/kg]
J	Thermal inertia [kW]
k	Proportionality constant between pressure drop and mass flow rate [kg s^{-1} $bar^{-1/2}$]
K	Thermal transmittance [kW/m^2 K]
LPH	Low pressure pre-heater
HPH	High pressure heater
$\dot{m}$	Mass flow rate [kg/s]
M	Mass [kg]
p	Pressure [bar]
$\dot{q}$	Thermal power [kW]
ρ	Density [kg/m^3]
S	Surface [m^2]
T	Temperature [°C]
ΔT_m	Logarithmic Mean Temperature Difference [°C]
Subscripts	
d	Drain water
e	Extracted steam
in	Inlet
H_2O	Water
met	Metal
out	Outlet
tb	Turbine
v	Vapour
Superscripts	
i	i^{th} element
j	j^{th} element

According to the official data [1], the Italian fuel mix is largely dependent on oil and gas, differently from the other European Union (EU) countries. The exploitation of coal for power production remains largely below the EU average, whereas is remarkable the large employ of expensive fossil sources as oil and gas; this implies a high cost of electricity.

Instead, the amount of energy produced by renewable sources is in line with the EU countries average value, being around 23%. This constitutes, however, a problem for the national electric grid balancing, since the amount of electric power produced by sun and wind is not controllable and is subjected to rapid oscillations during the 24 hours. The sudden lack of renewable power production needs to be rapidly balanced by the increase of power produced by fossil fired plants, in order to avoid malfunctions in some zones of the national grid.

In this scenario, the Italian energy authorities are planning to increase the power produced by coal in the next decades, with the construction of new plants and the re-vamping of the older ones [2,3,4]. In particular, the new plants built will implement the state of the art technology, such as the ultra-supercritical steam cycle (260 bar maximum pressure and 600 – 610 °C maximum temperature) [5,6,7].

The aforementioned grid balancing issue is faced by a new regulation of the grid code imposing, for the power plants rated more than 10 MW, the requirement of primary frequency control. According to the regulation [1], these plants need to be able to:

- increase (or decrease) their power by up to 4% of the design power in an interval of 30 s, with a variation of at least 2% in the first 15 s;
- keep the increased (or decreased) power level for up to 15 minutes.

In this paper, a power regulation system able to satisfy the grid code requirements is applied to an ultra-supercritical (USC) pulverized coal power plant of 460 MW design power. As known, once-through boilers employed in the USC plants are much more rapid in following the load changes than steam drum boilers; however, a power increase of 4% in an interval of 30 s is not possible acting only on the coal burners. Therefore, the rapid power increase maneuver needs to first involve other plant components. Three solutions are possible:

1) bypass of the turbine 1st stage, increasing the expanding steam flow-rate thanks to the larger passage area available. This solution may lead to stresses on the turbine and on its regulation components;
2) bypass of the high pressure feed-water pre-heaters (HPH) to reduce steam bleeding from the turbine. This solution involves components subjected to high pressure and temperature, resulting in high stresses; moreover, this maneuver affects directly and instantly the boiler water inlet temperature;
3) bypass of the low pressure feed-water pre-heaters (LPH) to reduce steam bleeding from the turbine.

This paper analyzes the 3rd solution, proposing a feed-water bypass of three LPH to an auxiliary water storage tank. The feed-water bypass is activated as soon as the request for power increase arrives; in the meantime, the coal burners need to be regulated in order to increase the boiler load, so that the higher power level reached can be maintained for the time required.

To study the proposed solution effectiveness, a Matlab-Simulink concentrated parameters model of the low pressure feed-water line was worked out, in order to understand the system regulation characteristic and dynamic behavior.

The proposed system may be utilized not only for the maximum power increase required by the grid code, but also for smaller power increases; being the system able to assure power increases with a continuous range of values, the implemented model was also used to study the system regulation capability. Finally, the stresses on the LPH were evaluated, in terms of temperature difference variation and temperature time-derivative.

2. The proposed power regulation system

In Figure 1 is reported the scheme of the plant low-pressure part, including the proposed bypass line, composed by the valve V and the tank R.

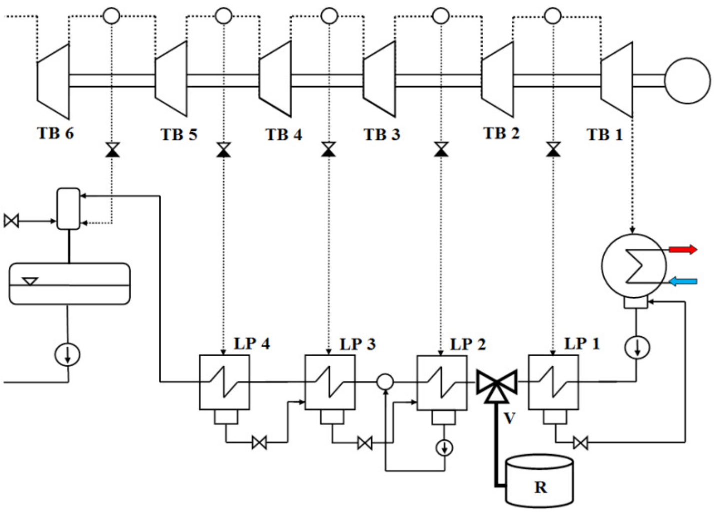

Fig. 1. Scheme of the low pressure part of the plant with auxiliary bypass tank.

As visible, the low pressure feed-water is pre-heated by four heat exchangers fed by steam bleedings coming from the turbine body. A fifth steam bleeding is mixed, at design conditions, with the whole flow-rate in the de-aerator, to provide the elimination of incondensable gases.

In case of demand for rapid power increase, the bypass involves the LP2, LP3 and LP4 heat exchangers. During the power increase maneuver, feed-water is split in two paths: at the end of the bypass maneuver, the larger flow rate bypasses the three pre-heaters and enters the auxiliary storage tank R, which needs to be thermally isolated; the remaining flow-rate continues its original path crossing the pre-heaters and entering the de-aerator shell. The reduction of the cooling flow in the LPH bundles causes, after a thermal transient, the decrease of the heat exchanged between water and steam and thus of the heat exchangers condensation capacity; hence, the steam flow rates bled from the turbine decrease, consequently increasing the flow-rate crossing the turbine and, as will be demonstrated in the following, the power produced.

The feed-water flow-rate still crossing the LPH bundles is necessary to keep a small steam flow-rate in the heaters shells, in order to keep the metal surfaces as close as possible to design temperature and avoid thermal stresses. The feed-water flow-rate still circulating in the LP1, LP2 and LP3 heaters reaches the de-aerator and enters in direct contact with the steam from the turbine bleeding. The condensation of the bled steam in the de-aerator, assured at design conditions by a large feed-water flow-rate, is now limited. Therefore, the de-aerator pressure rises, thus reducing the bled steam flow rate from the turbine. The temperature of the feed-water tank below the de-aerator is not expected to change significantly after the maneuver.

Contemporarily to the actuation of the power increase maneuver, the coal burners must increase their load in order to let the boiler reach the increased power level; of course, a proper regulation of the feed-water pump rotational speed must occur to assure the increase of live steam flow-rate. As live steam flow-rate rises, the feed-water flow-rate in the LPH needs to be properly restored, in order to keep the turbine power at the increased level. The by-pass water, stored in the tank, will be re-injected in the loop by means of an auxiliary pump. Attention should be paid to the plant main feed-water storage tank, installed below the de-aerator: at design conditions it contains about 260 t of water; the lack of feed-water from the LPH might lead to its empting, since the feed-water pump continues to subtract water.

However, this paper does not account for the procedures following the load increase transient, also because no information about the boiler behavior is available.

The main advantages of the proposed solution are the limited stresses on the turbine – contrarily to the solution involving bypass of the turbine 1st stage – and the absence of solicitations on the HPH.

In the following section is described in detail the Matlab-Simulink model utilized for the calculations; the model represents the LPH line from the condenser to the plant main feed-water storage tank, including the six turbine bodies laying among the steam bleedings.

3. The simulation model

In the following, the equations implemented in the concentrated parameters model are described. All the components obey to the general equations of mass and energy conservation written in their transient form [8,9].

A numerical model, implemented in the Matlab Simulink environment, was built including all the equations below reported for the different components. The aim of the model is carry out the tests needed to evaluate the behavior of the proposed solution. The water and steam properties were calculated according to [10].

3.1. The heat exchanger

By hypothesis, the LPH units are modeled as zero-dimensional components. The temperature of the steam contained in the i^{th} heat exchanger shell is found according to Equation (1);

$$T_e^i = T_{H_2O\_in}^i + \frac{\dot{q}_v^i}{E_r\left(1 - e^{\frac{-KS}{E_r}}\right)} \tag{1}$$

where $E_r = c_{p\,H_2O}\,\dot{m}_{H_2O\_in}$ represents the equivalent in water, K and S respectively the thermal transmittance and the exchange surface and $\dot{q}_v^i$ the thermal flow exchanged by the steam, calculated through the cavity thermal balance in Equation (2):

$$\dot{q}_v^i = \left(\dot{m}_e^i H_{tb}^i + \left(\sum_j \dot{m}_d^j \right) h_d^{i-1} \right) - \left(\dot{m}_d^i \; h_d^i \right) \tag{2}$$

$\dot{m}_e^i$ is the extracted steam mass flow rate, having enthalpy content H_{tb}^i calculated by the turbine model. The drain water coming from the downstream heat exchangers (*j*) is mixed with the bled steam in the shell of the i^{th} element. $\dot{m}_d^i$ represents the drain flow from the heat exchanger, having enthalpy content equal to h_d^i.

The mass flow rates extracted from the turbine are calculated as proportional to the square root of the pressure difference existing between turbine and shell: $\dot{m}_e^i = k^i(\rho_{tb}^i(p_{tb}^i - p_e^i))^{1/2}$.

Similarly, the drain water mass flow rates depend on the pressure difference between the shells of two adjacent heat exchangers: $\dot{m}_d^i = k_v(\rho_d^i(p_e^i - p_e^{i-1}))^{1/2}$; the valve characteristic parameter, k_v , is controlled in order to keep the condensed water level constant within the heat exchanger. The same level control is utilized for the pump installed below the LP2 unit.

The feed-water temperature at the heat exchanger exit is calculated through Equation (3), while the relationship between water ($\dot{q}_{H_2O}^i$) and steam ($\dot{q}_v^i$) thermal flows is given in Equation (4).

$$T_{H_2O\_out}^i = T_{H_2O\_in}^i + \frac{\dot{q}_{H_2O}^i}{E_r} \tag{3}$$

$$\dot{q}_{H_2O}^i = \dot{q}_v^i + J_{met} + J_{H_2O} \tag{4}$$

The terms J_{met} and J_{H_2O} represent the thermal inertia of, respectively, the heat exchangers metal walls and the water contained in pipes. These terms are calculated according to the following Equations (5) and (6):

$$J_{met} = M_{met} \, c_{p\,met} \, \frac{d\,T_{met}}{dt} \tag{5}$$

$$J_{H_2O} = M_{H_2O} \, c_{p\,H_2O} \, \frac{d\,\bar{T}_{H_2O}}{dt} \tag{6}$$

where $T_{met} = T_e - \Delta T_m$, ΔT_m being the heat exchanger logarithmic mean temperature difference and $\bar{T}_{H_2O} = \frac{1}{2}\left(T_{H_2O_{in}} + T_{H_2O_{out}}\right)$.

3.2. The turbine

The turbine was modeled as composed by six bodies (see Figure (1)); each body reproduces the expansion between two steam bleedings.

The turbine bodies were modeled under the assumption of chocked flow, according to Equation (7):

$$p_{tb}^i = \frac{\left(\dot{m}_{tb}^i\right)^2}{(A^i)^2 \; \rho_{tb}^i \; F_i^2(k)} \tag{7}$$

where $\dot{m}_{tb}^i$ is the flow rate crossing the body, A^i is the inlet passage area, ρ_{tb}^i is the steam density at the turbine inlet and $F_i^2(k)$ is a quantity function of the steam isentropic expansion coefficient.

The expansion curve, implemented as a function $H_{tb}^i = f\left(p_{tb}^i\right)$,was considered fixed with the turbine load. This assumption will be modified as soon as more precise knowledge of the turbine efficiency in off-design is available.

The turbine dynamics was not modeled, being much more rapid than the transient involving the LPH, the latter being connected to thermal exchange phenomena.

3.3. The de-aerator, the plant main feed-water storage tank and the condenser

The de-aerator and the feed-water storage tank were modelled by the mass and energy balance equations, in the time-varying form.

The condenser and the hot-well below obey to the mass and energy balance equations in the time-varying form. The condensation temperature was calculated in function of the steam mass flow rate coming from the turbine, by the use of the condenser characteristic vacuum curves.

4. Results

Tests on the simulation model were carried out to understand the suitability of the proposed system; in particular, the tests and the results described in the following paragraphs refer to:

- the evaluation of the system power regulation capability; this issue was investigated by testing 5 different feed-water bypass ratios;
- the assessment of the system dynamics during the transient for the most relevant variables;
- the estimation of stresses induced in the heat exchangers metal parts by temperature variations.

The manoeuvre is effected opening the bypass valve; the opening operation lasts for a time interval of 15 s, considered long enough in order to avoid water hammer phenomena.

4.1. Evaluation of the system power regulation capability

To quantify the power increase and the regulation possibility obtainable with the maneuver, several tests with different bypass ratios were worked out. As “bypass ratio” is intended the ratio between the feed-water mass flow rate crossing the LPH and that flowing to the auxiliary tank through the bypass line. Five values of bypass ratio were tested: 3%, 22.4%, 41.8%, 61.2%, 80.6%. Table 1 shows the main parameters for each turbine body in design conditions and after the regulation maneuver, for a bypass ratio equal to 3%.

Table 1. Comparison between design conditions and bypass conditions for the turbine bodies, for 3% bypass ratio.

		Power [MW]	Turbine inlet flow [kg/s]	Flow in bleeding [kg/s]	Turbine enthalpy drop [kJ/kg]	Expansion ratio [-]
TB 1	Design	19.10	216.60	-	88.18	3.14
	Bypass	27.60	275.60	-	100.15	3.14
	Difference	8.50				
TB 2	Design	37.14	223.89	7.26	165.87	3.45
	Bypass	48.83	284.20	8.59	171.82	3.45
	Difference	11.69				
TB 3	Design	52.13	234.01	10.12	222.75	3.87
	Bypass	66.14	284.67	0.47	232.35	3.75
	Difference	14.02				
TB 4	Design	53.70	253.67	19.66	211.68	2.64
	Bypass	53.44	285.55	0.88	187.14	2.40
	Difference	-0.26				
TB 5	Design	50.24	269.68	16.01	186.29	2.20
	Bypass	49.80	286.15	0.59	174.03	2.06
	Difference	-0.44				
TB 6	Design	43.32	286.75	17.06	151.08	4.62
	Bypass	39.22	286.75	0.60	136.78	4.33
	Difference	-4.10				
TOTAL INCREASE		29.41				

The quantities shown in Table 1 are the body power, the mass flow rate crossing the body, the mass flow rate of steam bled from the body, the body enthalpy drop and its expansion ratio.

The table shows that TB4, TB5 and TB6 bodies experiment a slight decrease of the produced power. This is due to the fact that in bypass conditions, although the flow-rate increase, the enthalpy drop of the three high pressure bodies is lower than on design. However, the total power increase results equal to 29.41 MW, which is about 6.4% of the total design power of the plant (460 MW), largely above the increase required by the grid code.

For the tests conducted with the other bypass ratios, the system behavior is similar: in all the conditions tested, the TB5 and TB6 bodies provide slightly negative contributions to the total power increase. The tests show that, for 41.8%, 61.2% and 80.6% bypass ratios, turbine TB4 provides positive power difference values of respectively 0.002 MW, 0.649 MW and 0.872 MW.

Figure 2 shows the diagram of power increase with bypass ratio. This diagram is useful as it represents a sort of "characteristic curve" of the regulation system.

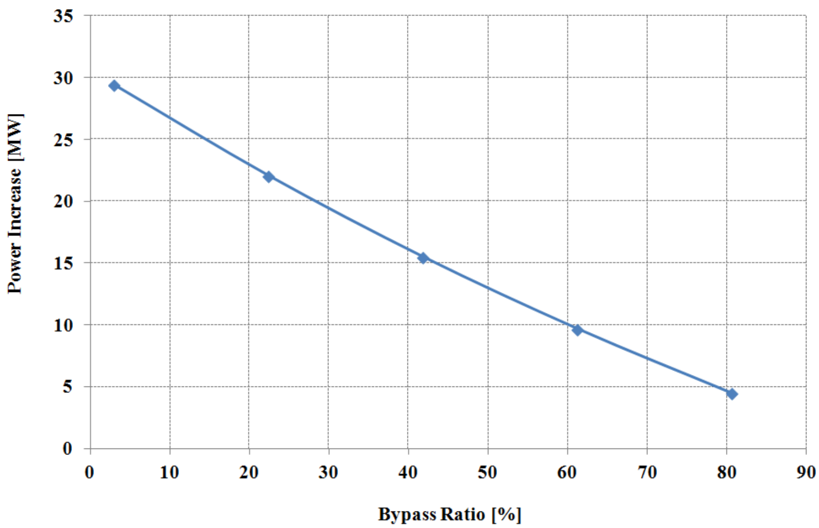

Fig. 2. Power increase versus bypass ratio: characteristic curve of the regulation.

4.1. Plant dynamic behaviour during bypass manoeuvre

In this section, an analysis of the system dynamic behavior during the transient time is provided for the most relevant quantities. The case analyzed is that providing a bypass ratio of 3% at the end of the maneuver. In Figure 3 is represented the total turbine power (sum for the bodies TB1 – TB6) with time.

From Figure 3 can be deduced that the power increase transient satisfies the requirements of the grid code. In fact, after 15 s the power increase is 12.63 MW, corresponding to 2.74% of the plant design power (460 MW). As remarked before, the total power increase resulting from the maneuver is 29.41 MW, corresponding to 6.4% of the plant design power. Such power increase is reached within the time interval of 30 s, as required by the grid code.

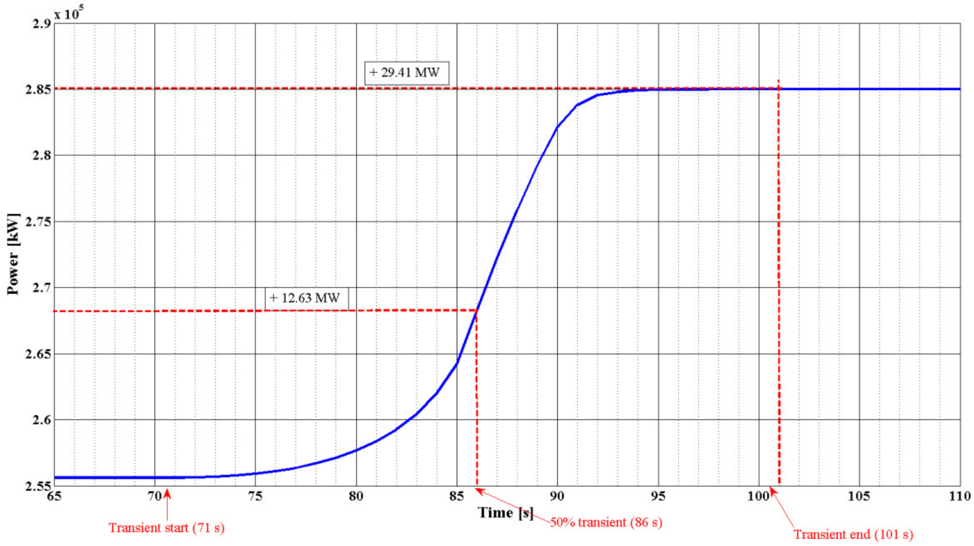

Fig. 3. Power of the bodies TB1 – TB6 versus time.

The main problem is the fact that the feed-water storage tank sees its water feeding dramatically reduced and may be subjected to empting if no other maneuvers are provided. According to the simulations, at the end of the 15 minutes transient, the mass stored in the tank reduces from 260 t to about 6.4 t if no other actions are performed; this attests that the boiler will not suffer from lack of coolant. In reality, as soon as the boiler load increases sufficiently, the bypass flow-rate needs to be reduced, down to zero. Future researches will account for the remaining part of the plant (i.e. the high pressure line, the boiler and the high pressure bodies of the turbine), not modeled in this work, to explain this open issue.

In Figure 4, the temperature of the feed-water leaving each LPH (T1, T2, T3 and T4) is diagrammed with time; also the condenser outlet temperature (T0) is provided.

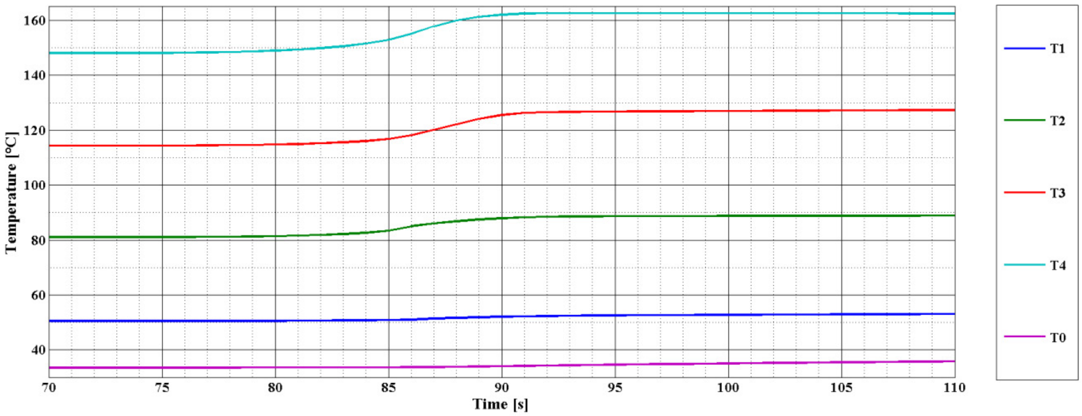

Fig. 4. Temperature versus time diagram for the low pressure feed-water line.

The inlet-outlet temperature difference to which the four heat exchangers are subjected is very similar before and after the feed-water bypass maneuver, as visible from Table 2.

Table2. Inlet-outlet temperature difference for the four heaters, before and after the bypass maneuver.

Inlet-outlet temperature difference [°C]	LP1	LP2	LP3	LP4
Design	16.97	30.55	33.31	33.73
Bypass	15.92	36.69	39.51	31.97

To demonstrate the stress levels induced by the maneuver, it is interesting to provide the maximum value of the LPH feed-water temperatures time derivative during the transient. This is presented in Table 3.

Table3. Maximum value of the feed-water line temperatures time derivative during the transient.

	d(T0)/dt	d(T1)/dt	d(T2)/dt	d(T3)/dt	d(T4)/dt
Temperature time derivative [°C/s]	0.11	0.44	1.63	2.09	2.82

The maximum derivative value affects the T4 temperature, calculated at the LP4 heater outlet. As visible, said value is lower than 3°C/s.

5. Conclusions

This paper presented a system for steam power plants power regulation, based on the bypass of the low pressure feed-water pre-heaters. The possibility of power regulation is today of interest also for the large steam power plants, as these are required to vary rapidly their produced power in order to provide a contribution to the grid frequency regulation.

To understand the proposed system efficacy, a Matlab-Simulink model of the low pressure feed-water line (from condenser to de-aerator) and of part of the turbine was implemented and used for testing different bypass ratios of the pre-heaters.

The tests results outline:

- the capability of the system to achieve the grid requirements: the power increase obtained through a bypass maneuver (with bypass ratio of 3%) is largely above 4% of the design power. According to the tests made, to reach the required power increase of 4%, a bypass ratio of 30% is sufficient;
- the limited stresses induced in the heat exchangers.

The main problem of the proposed solution regards the auxiliary water tank, which needs to store all the bypassed feed-water at a temperature of about 50°C. To size this component, information about the boiler dynamics is required.

The main issue still open regards the modeling of the remaining part of the plant, i.e. the high pressure pre-heaters, the highest pressure bodies of the turbine and the boiler. Once the model is completed, it will be possible to make decisions about the actions to be performed during the power increase maneuver.

References

[1] www.terna.it

[2] McMullan J T, Williams B C, Sloan E P, Strunk Clean Coal Technologies, , Proc. IMeChE, Journal of Power and Energy 1997, 211, p. 95-107.

[3] Franco A, Diaz A R, The future challenges for “clean coal technologies”: Joining efficiency increase and pollutant emission control, Energy 2009, 34 (3), p. 348-354.

[4] Masuyama F, History of Power Plants and Progress in Heat Resistant steels, ISIJ International 2001, 41(6), p. 612-625.

[5] F. Laubli, and F. H. Fenton, The Flexibility of The Supercritical Boiler as A Partner in Power System Design and Operation: Part I, IEEE Transactions on Power Apparatus and Systems1971, PAS-90 (4), p. 1719-1724.

[6] F. Laubli, and F. H. Fenton, The Flexibility of The Supercritical Boiler as A Partner in Power System Design and Operation: Part II IEEE Transactions on Power Apparatus and Systems 1971, PAS-90 (4), p.1725-1733.
[7] Mohamed O, Wang J, Guo S, Al-Duri B, Wei J, Modelling Study Of Supercritical Power Plant And Parameter Identification Using Genetic Algorithms, Proceedings of the World Congress on Engineering 2010 Vol II, WCE 2010, June 30 - July 2, 2010, London, U.K.
[8] Shirikawa M, Nakamoto M, Hosaka S, Dynamic Simulation and Optimization of Start-up Processes in Combined Cycle Power Plants, JSME International Journal2005, 48 (1), p. 122-128.
[9] T. Inoue, H. Taniguchi, and Y. Ikeguchi A Model of Fossil Fueled Plant with Once-through Boiler for Power System Frequency Simulation Studies IEEE Transactions on Power Systems 2000, 15 (4), p. 1322-1328.
[10] Wagner, W. K., Properties of water and steam/IAPWS-IF97. Berlin, Germany: Springer-Verlag; 1998.