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Dynamic simulation of combined cycle power plant cycling in the electricity market

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ABSTRACT

The energy markets deregulation coupled with the rapid spread of unpredictable energy sources power units are stressing the necessity of improving traditional power plants flexibility. Cyclic operation guarantees high profits in the short term but, in the medium-long time, cause a lifetime reduction due to thermo-mechanical fatigue, creep and corrosion. In this context, Combined Cycle Power Plants are the most concerned in flexible operation problems. For this reason, two research groups from two Italian universities have developed a procedure to estimate the devices lifetime reduction with a particular focus on steam drums and superheaters/reheaters. To assess the lifetime reduction, it is essential to predict the thermodynamic variables trend in order to describe the plant behaviour. Therefore, the core of the procedure is the power plant dynamic model. At this purpose, in this paper, three different dynamic models of the same single pressure Combined Cycle Gas Turbine are presented. The models have been built using three different approaches and are used to simulate plant behaviour under real operating conditions. Despite these differences, the thermodynamic parameters time profiles are in good accordance as presented in the paper. At last, an evaluation of the drum lifetime reduction is performed.

1. Introduction

In the last few decades, the global energy demand has risen to a level never reached before. This, in turn, has led to several environmental problems such as air pollution, global warming, the reduction of the ozone layer and the depletion of fossil fuels. These aspects have forced the international administrations to promote the liberalization of the energy markets (see [1,2]) and the spread of renewable energy sources (RES) [3].

A major result of this process is a high penetration of unpredictable energy sources such as wind and solar, with big impact on the electricity market. Then, as already discussed by the Authors [4–6], flexibility, availability and fast cycling have become fundamental concepts to be competitive in this new electricity market. For this reason, thermoelectric units need to switch from base-load to cycling operation: an operation mode characterized by fast load ramps, short start-up and shut-down time that permits to enhance the power plant's competitiveness and to maintain the grid stability.

As outlined by Balling [7], the grid stability is often compromised by the high number of power plants fed by unpredictable renewable energy sources and by the absence of large-scale energy storage systems. Thus, investments focused on this research field are necessary to guarantee the stability of electrical grids in the European framework. Nevertheless, as presented by Keatley et al. [8] for the case of Ireland, the grid stability is also of particular concern to the owners and operators of fossil-fuel power plants because cycling operation of these units is required to integrate very high levels of wind power. Moreover, Balling [7] states that Germany conventional power plants will have to be started up and shut down several times weekly, or even daily, in the next five years. Obviously, conventional power plants cycling and energy storage systems are fundamental to maintain the grid stability but another promising option is offered by cogenerative hybrid systems with energy storage [9,10] and waste heat recovery units [11–15].

Therefore, new operating requirements for fossil-fuel power plants arise (two-shift operation, island operation, load-follow operation, black start capability and severe start-up) in order to stabilize power grid dynamics and ensure economic electricity supply. This new kind of operation strategy guarantees high profits in the short term, but determines a significant reduction in the

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Nomenclature

C_3, C_5	coefficients in Eq. (5)
F_r	Froude number
P_r	Prandtl number
R_e	Reynolds number
T	temperature [°C]
$\dot{m}$	mass flow rate [kg s^{-1}]
Nu	Nusselt number
d	diameter [m]
h	heat transfer coefficient [W m^{-2} K^{-1}] or height [m]
l	pipe length [m]

Abbreviations

CCPP	Combined Cycle Power Plant
CS	Pump Control System
ECO	economizer
EVA	evaporator
HRSG	Heat Recovery Steam Generator
HX	heat exchanger
MSM	Matlab Simulink Model
PI	Proportional–Integral
RES	renewable energy sources
SAS	steam attemperator system
SH	superheater
ST	steam turbine
TP	ThermoPower library
TPL	Thermal Power Library

Greek letters

λ	thermal conductivity [W m^{-1} K^{-1}]
ξ	operator in Eq. (2)

Superscript

n	operator in Eq. (6)

Subscripts

c	cold
des	design
f	fin
g	gases
hyd	hydraulic
i	inner
m	mean
o	outer
s	surface

lifetime of the most critical power plant devices, which are those subjected to thermo-mechanical fatigue, creep and corrosion (see i.e. Salonen et al. [16] and Lefton et al. [17]).

As presented by Tica et al. [18] and Alobaid et al. [19], improving start-up performance, load ramps and shut-downs is essential to be competitive but, as underlined by Benato et al. [6], the availability of procedures able to predict the residual life of power plant devices, considering the combined effects of creep, thermo-mechanical fatigue, corrosion and oxidation, is essential to optimize plants' operation and maintenance scheduling.

Furthermore, in the liberalized energy market, power plant operators need simulation tools able to test different operation strategies which allow them to manage the plant without excessively compromising its residual life. These tools can be very useful not only during the plant design phase but also in the daily operation, in order to better schedule load ramps and shut-downs and increase the gap between peak power and technical minimum load without neglecting environmental constraints [20].

Being combined cycles the most rapid, efficient and widespread technologies, they are the most concerned when dealing with flexibility. Usually, they provide spinning and cold reserve services or two-shift operation (since they often work with daily start-up and shut-down) thanks to their intrinsic flexibility which is higher than that of steam power plants [16].

In combined cycle gas turbine units, Heat Recovery Steam Generators (HRSGs) [21] and gas/steam turbines are the most critical components being exposed to creep and low-cycle fatigue degradation [22,23]. In particular, in multiple pressure level HRSGs, high pressure steam drums are among the most stressed components as they are characterized by great thickness and present many weakness points (down-comers, risers, steam tubes) which determine high values of stress concentration factors. Each fatigue load cycle deteriorates the metal parts and the accumulated damage ends up causing breakdowns and thus determining unplanned maintenance interventions; to this end, Carazas et al. [24] present a method for the reliability and availability evaluation of HRSGs installed in combined cycle power plants, in order to better identify the components more subjected to failures.

Considering the new market scenario, where flexibility is paramount, and the flexibility related problems, the Authors have developed an innovative method (lifetime calculation procedure) able to predict the power plant behaviour during cycling operation modes and estimate the power plant components' lifetime reduction [6].

As said, to estimate the lifetime reduction of metal components due to cycling, it is essential to foresee the trends of the main thermodynamic parameters (such as water/steam mass flow rates, temperatures and pressures) that describe the plant behaviour. Therefore, the core of the lifetime calculation procedure is the power plant dynamic model. Nowadays, dynamic simulation and, in particular, power plant dynamic analysis, is an essential step to achieve the desirable performance under the various kinds of constraints related to system design, plant operation and environmental impact. In literature, several mathematical models were proposed to investigate the combined cycle power plants Heat Recovery Steam Generator (HRSG) behaviour using different simulation tools. Dumont and Heyen [25] developed a mathematical model of a once-through Heat Recovery Steam Generator while Ong'iro et al. [26] built a model of a two pressure level HRSG unit. Shirakawa et al. [27] built a dynamic simulation model able to optimize the start-up process of a combined cycle gas turbine unit. Alobaid et al. implemented a static and dynamic simulation model of a subcritical and supercritical Heat Recovery Steam Generator (see [19,28]) using the advanced process simulation software Apros [29] while, with the advanced processing simulation software Aspen Plus Dynamics [30], they investigate the Heat Recovery Steam Generator behaviour during start-up procedure [31]. A model of a natural circulation HRSG using the Modelica language was developed by Casella and Pretolani [32]. The study aimed at reducing the start-up time while keeping the life-time consumption of the more critically stressed components under control. The authors focused their attention on the thermal stresses of the steam turbine but extremely simplified the low pressure part of the HRSG. In addition, the heat exchangers behaviour is not considered. Conversely, Heimo [33] focused his attention on developing simplified dynamic components, in Modelica, that can be used to build plant models. Finally, a brief review of the modelling and simulation techniques for the analysis of both static and dynamic operation of power plants is outlined in [34] while the dynamic behaviour of small scale power plants (such as Organic Rankine cycle and air bottoming cycle units) has been investigated using Modelica language in [35–37].

In conclusion, notwithstanding the above-mentioned works, to the knowledge of the Authors, none of these studies presented a comparison among different dynamic models, with different degree of details, of the same single pressure combined cycle power unit in order to predict the plant behaviour and estimate the most stressed components lifetime reduction.

The present paper is organized as follows: the three dynamic simulation models, developed by two different research groups, are described, highlighting their peculiarities, in Section 2, whereas in Section 3 the models are validated taking into account both literature and experimental data. In Section 4 the dynamic models are used to simulate the plant behaviour during two typical transient conditions provided by an Italian operator while in Section 5, the procedure that has been developed to estimate the lifetime reduction of the most stressed components is briefly outlined and applied to study the effect of three different similar transient conditions on the steam drum. Finally, concluding remarks are drawn in Section 6.

2. Case study and methodology

The two research groups (from the University of Padova and from the University of Genova) have developed three dynamic simulation models of the same combined cycle power plant (49.9% rated electrical efficiency) characterized by a single pressure level HRSG and with the following main design data:

- Gas turbine power = 56 MW.
- Steam turbine power = 31 MW.
- Exhaust gases mass flow rate = 191 kg s^{-1}.
- Exhaust gases inlet temperature = 582 °C.
- Steam mass flow rate = 28.5 kg s^{-1}.
- Steam evaporating pressure = 38 bar.

The first and the second model have been developed by the University of Padova group, exploiting the software DYMOLA 2014 [38], whereas the third model, in the Matlab/Simulink [39] environment, has been developed at the University of Genova.

2.1. Thermal power HRSG model

The model has been built utilizing the commercial software DYMOLA 2014 [38]; the code is developed in the MODELICA language. Such language enables an object-oriented approach to modelling as it makes use of advanced object-oriented techniques such as inheritance, replaceability and reusability. These features facilitate the development of advanced models because pre-defined components (e.g. pipes) can be directly utilized as sub-components in more complex models (e.g. heat exchangers). The Thermal Power Library (TPL) [40] is used to characterize the different components of the Combined Cycle Power Plant (CCPP). The layout of the model is shown in Fig. 1.

The HRSG has a horizontal gas pass design; it is composed by three heat exchangers (the economizer (ECO), the evaporator (EVA) and the superheater (SH)), the drum, two pumps (FeedPump and CircPump), pipes (Tube_1–Tube_5), the steam turbine (ST), the condenser (Condenser) and so on. The geometry of the heat exchangers is completely implemented into the heat exchanger components. In Table 1 the geometry of the heat exchangers is summarized.

To evaluate the heat transfer among exhaust gases, water/steam and pipes metal wall (fins are taken into account), the wall material characteristics (AISI 304) are included in this model. The geometry and the wall material of the drum are also included (external radius = 750 mm, length = 10,800 mm, wall thickness = 30 mm, wall material P355GH EN10028-3). The steam turbine is modelled using the Stodola's equation [41] and setting nameplate operating conditions (such as mass flow rate, inlet and outlet temperature and pressure), mechanical and isentropic efficiency. Pumps are built using a module which describes a centrifugal pump with ideally controlled speed, either fixed or provided by an external signal. The feed pump works with variable speed while the circulation pump works at a fixed speed but in both cases the pump's maps have been established. The circulation pump is inserted to assist circulation into the evaporator tubes.

The dynamics of the shaft is modelled by applying the shaft dynamic balance; also the value of the moment of inertia of this component is set.

The model includes all the control systems, whose role is very important during cycling operation mode. Drum vent, admission valves and pump speed controller (CS) are implemented. They are in charge of maintaining the correct water level in the drum. The plant operates in sliding pressure mode.

Steam attemperator system (SAS) is needed to avoid that the steam temperature at the outlet of the HRSG exceeds the maximum admissible temperature (560 °C). The feed water pump is controlled by a PI controller which provides to maintain the drum level on a prefixed value. This pump, as said before, is centrifugal and is characterized by a design speed of 3000 rpm; its speed can change from 1500 to 4000 rpm during load variations in order to control the drum level while the steam turbine control maintains the turbine's speed at 3000 rpm. The composition of exhaust gases is modelled adopting Nasa Extended Exhaust Gas Model which is inserted into Thermal Power Library [40] while for water/steam fluid model, the IF 97 water model is used [42]. The heat transfer coefficient for the water/steam side in the economizer and superheater components is computed for each pipe segment with the equation:

$$h_c = Nu_m \cdot \frac{\lambda}{d_{hyd}} \tag{1}$$

where λ is the fluid thermal conductivity and d_{hyd} is the hydraulic diameter of the pipe. Nu_m is the average Nusselt number and is computed by means of Reynolds number using an equation put forward by Gnielinski [43] for heat transfer during turbulent flow of gases and liquids through pipes [44].

$$Nu_m = \frac{\left(\frac{\xi}{8}\right) \cdot Pr \cdot Re}{1 + 12.7 \cdot \sqrt{\frac{\xi}{8}} \cdot \left(Pr^{\frac{2}{3}} - 1\right)} \cdot \left[1 + \left(\frac{d_i}{l}\right)^{\frac{2}{3}}\right] \tag{2}$$

where Pr and Re are the Prandtl number and the Reynolds number; l is the pipe length and d_i is the pipe inner diameter. The friction factor ξ is given by:

$$\xi = (1.8 \cdot \log_{10}(Re) - 1.5)^{-2} \tag{3}$$

Note that Eq. (2) was obtained by modifying an equation that was derived from the theory of momentum transport by Petukhov and

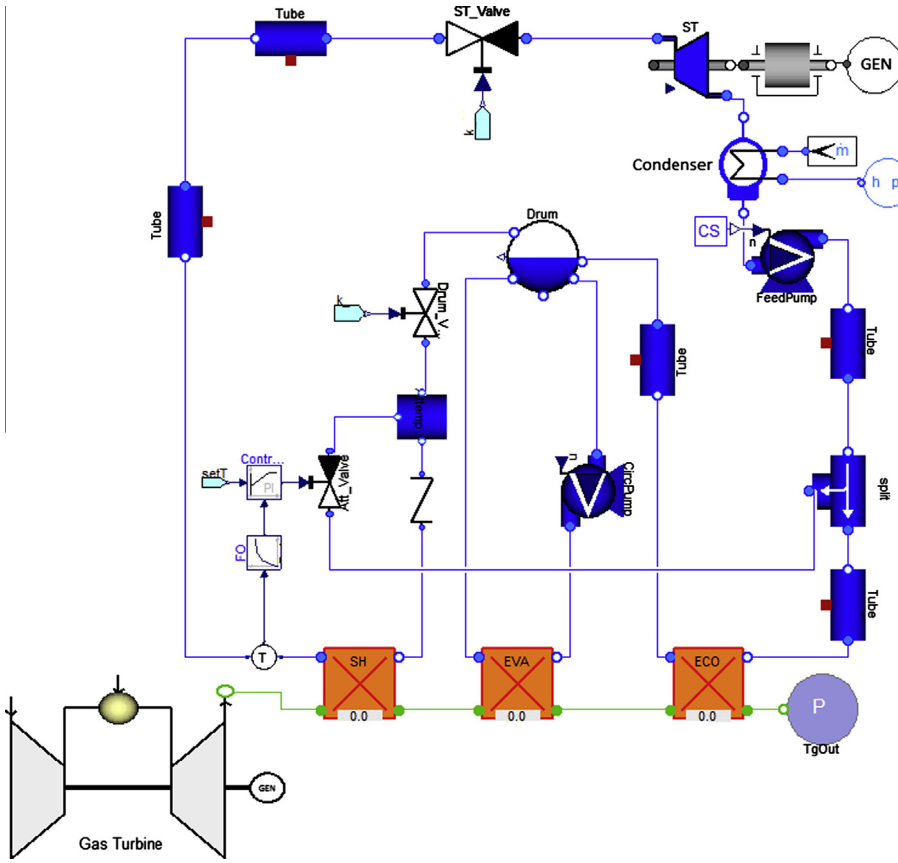

Fig. 1. Modelica object diagram of the single pressure Heat Recovery Steam Generator.

Table 1
Heat exchanger geometry data.

Device	Parameter	Value
Economizer	Outer diameter	25.0 mm
	Tube thickness	2.5 mm
	Tube length	10.0 mm
	Number of tubes	7560
	Longitudinal and Transverse tube pitch	78 and 90 mm
	Number of fins	236 fins m^{-1}
	Fin dimension	$12 \times 4 \times 1.2$ mm
	Fin pass	4.24 mm
Evaporator	Outer diameter	25.0 mm
	Tube thickness	2.5 mm
	Tube length	7.0 mm
	Number of tubes	12960
	Longitudinal and Transverse tube pitch	78 and 90 mm
	Number of fins	236 fins m^{-1}
	Fin dimension	$12 \times 4 \times 1.2$ mm
	Fin pass	4.24 mm
Superheater	Outer diameter	38.0 mm
	Tube thickness	6.0 mm
	Tube length	5.0 mm
	Number of tubes	2080
	Longitudinal and Transverse tube pitch	78 and 90 mm
	Number of fins	236 fins m^{-1}
	Fin dimension	$12 \times 4 \times 1.2$ mm
	Fin pass	4.24 mm

Kirillov [45] and is valid for completely developed pipe flow. It was enlarged by a factor proposed by Hausen [46] to take into account the dependence of the heat transfer coefficient on the length of the pipe. According to Konakov [47], the friction factor for turbulent flow in smooth pipes may be calculated from Eq. (3). Ranges of validity are reported as follows:

$$\frac{d_i}{l} \leqslant 1 \quad 10^4 < Re < 10^6 \quad 0.1 < Pr < 10^3 \tag{4}$$

where again Re and Pr are the Reynolds and Prandtl number, respectively.

The heat transfer coefficient during evaporation processes (which occur into the pipes) is computed by the correlation for convective boiling in vertical and horizontal tubes with a Froude number higher than 0.05. It uses a modified Dittus–Boelter equation for the heat transfer coefficient of the liquid and is multiplied by an enhancement factor which depends on the steam quality and the Boiling number. For horizontal tubes and Froude lower than 0.05 it also contains a multiplicative correction term (see [44,48]). Note that, the correlations are implemented into the heat exchangers models and, based on the fluid regime and state, the solver selects the appropriate equation.

The heat transfer coefficient on the gas side of the three heat exchangers, can be obtained with several correlations available in the open literature for solid or serrated finned tubes. Some equations for finned tubes with serrated and solid fins were developed by e.g. ESCOA [48] (Extended Surface Corporation of America) as well as by Nir [49]. The correlation proposed by ESCOA [48], to evaluate the Nusselt number for serrated and solid fins in staggered tube arrangement is employed.

$$Nu = \frac{1}{4} \cdot Re^{0.65} \cdot Pr^{\frac{1}{3}} \cdot C_3 \cdot C_5 \cdot \left(\frac{T_{gm}}{T_s}\right)^{\frac{1}{4}} \cdot \left(\frac{d_o + 2 \cdot h_f}{d_o}\right)^{\frac{1}{2}} \tag{5}$$

where Nu, Pr and Re are the Nusselt, the Prandt and the Reynolds numbers while d_0 and h_f are the outer diameter and the average fin height, respectively. The coefficient C_3 and C_5 are computed according to the method outlined in [48].

Pressure drops are also computed in the heat exchanger model with appropriate correlations taken from Verein Deutscher Ingenieure [44], ESCOA Corp. [48] and Weirman [50].

In conclusion, the model developed with extended Thermal Power Library (TPL model) components is really detailed and is able to simulate the plant steady state, part-load and transient behaviour and predicts the trends of the main thermodynamic flow parameters (such as mass flow rates, pressures, and temperatures). It is also able to compute the temperature values along the thickness of the heat exchanger (HX) walls, pipes and drum. Once the variation over time of the thermodynamic variables is evaluated, the associated trend of the underlying mechanical parameters (stresses and strains in each component) can be computed using the well-known relationships as a function of pressure, temperature difference between internal and external radius of pipes or drum, temperature gradient along tube length and considering the variations of these parameters over time.

2.2. ThermoPower HRSG model

In this case, the model is developed again with the commercial software DYMOLA but components are taken from ThermoPower library (TP model). The Modelica object diagram is depicted in Fig. 2.

The HRSG has the same configuration and control system modules described in Section 2.1 but, in this model, the heat exchangers geometry is not included.

In practise, the heat exchanger is implemented by combining basic ThermoPower modules: 1D-flow models for the gas side (top) and fluid side (bottom of the figure), and the 1D-thermal model for the tube bundle (middle). The heat exchange in ThermoPower is modelled with the so-called 1D-thermal ports (in orange[1] in Fig. 3); the counter-current model establishes the topological correspondence between the control volumes on the tube walls, and the control volumes on the gas flow model. The tube metal wall is modelled by a 1D dynamic heat balance equation, discretized by finite volumes. The flow models contain one-dimensional dynamic mass and energy balance equations, discretized by the finite volume method, assuming a uniform pressure distribution; the relatively small friction losses are lumped in an external component model. Here, the pressure drops at off-design conditions are estimated assuming a quadratic dependency on the volumetric flow. As reported in [51,37], due to their relatively small contributions, the thermal resistance in the radial direction and thermal diffusion in the axial direction are neglected in the dynamic models. The heat transfer coefficient between the gas and the outer pipe surface is much lower than the one between the inner pipe surface and the working fluid flow. Therefore, the overall heat transfer is essentially dependent on the gas side only, and the working fluid temperature is always close to the inner surface temperature of the pipe. For this reason, the heat transfer coefficient at the interface between the gas and the metal wall, at off-design conditions, is evaluated by the relation proposed by Incropera et al. [52].

$$h = h_{des}\left(\frac{\dot{m}}{\dot{m}_{des}}\right)^{n} \tag{6}$$

where $\dot{m}$ is the mass flow rate and h is the heat transfer coefficient. The subscript "*des*" refers to the value at design operating conditions, while the variable n is the exponent of the Reynolds number. In the equation n is assumed equal to 0.6. The thermal interaction between the wall and the working fluid is described by specifying a sufficiently high constant heat transfer coefficient, so that the fluid temperature is close to the wall temperature, and the overall result is dominated by the gas side heat transfer (more details can be found in Casella and Leva [53]).

The input parameters of the heat exchanger models are the mass and the heat capacity of the tube metal wall, the volumes occupied by the exhaust gases, and by the water/steam, the design-point heat transfer coefficients on the hot and cold side and the hot and cold surface areas. Reasonable figures for these variables are obtained applying the well-documented standardized design-procedure for shell and tube heat exchangers outlined in Coulson et al. [54].

The steam turbine, the condenser, the pumps and the other components are implemented into the TP model as outlined in Section 2.1 while water/steam and exhaust gases are implemented with models described in Casella et al. [55].

2.3. Model in Matlab/Simulink

This dynamic simulation model of the combined cycle power plant has been built by means of the Matlab/Simulink [39] software by the University of Genova group; it is based on a non-linear lumped parameter mathematical model, described by a set of algebraic and partial differential equations, as reported in previous papers [6,56]. The main subsystems of the simulator are: the gas turbine, the three sections of the HRSG (economizer, evaporator and superheater), the steam turbine, the condenser, the pump and the control system. The water/steam properties are computed by means of the XSteam tool [57–59], whereas the specific heat and enthalpy of the exhaust gases are evaluated as a function of temperature and chemical composition. In the simulator, the gas turbine is a simple black box which gives as input to the HRSG the exhaust gas flow rate and temperature at the gas turbine discharge. Regarding the HRSG, as described in [6,56], the thermal storage in the metal parts of heat exchangers is considered, and both the economizer and the superheater (simulated as counter-current heat exchangers) are divided into multiple sections in order to guarantee, for all operating conditions, that the wall temperature of each section is greater than the water/steam outlet temperature and lower than the gas outlet temperature. The steam turbine is modelled as into the Modelica models, the condenser pressure is calculated as a function of the steam flow rate, while the system composed by the water pump and the valve downstream is managed by a PI controller that acts on the error between the feed-water mass flow rate and the steam flow rate at the drum exit.

As for the model developed in Modelica language with components taken from ThermoPower library, the Matlab/Simulink model has simplified heat exchangers while the TPL model is characterized by in-depth discretized heat exchangers with a really detailed geometry. This aspect guarantees to compute the temperature values along the thickness of the heat exchanger walls, pipes and HX collectors. Once the variation versus time of the thermodynamic variables is evaluated, the associated trend of the underlying mechanical parameters (stresses and strains in each component) can be computed using the well-known relationships as a function of pressure, temperature difference between internal and external radius of pipes or drum, temperature gradient along tube length and considering the variations of these parameters versus time.

3. Models validation

The three models described in Section 2 have been successfully validated taking into account both literature results and experimental data coming from power plant operators [6]. For this

[1] For interpretation of colour in Figs. 3, 11 and 12, the reader is referred to the web version of this article.

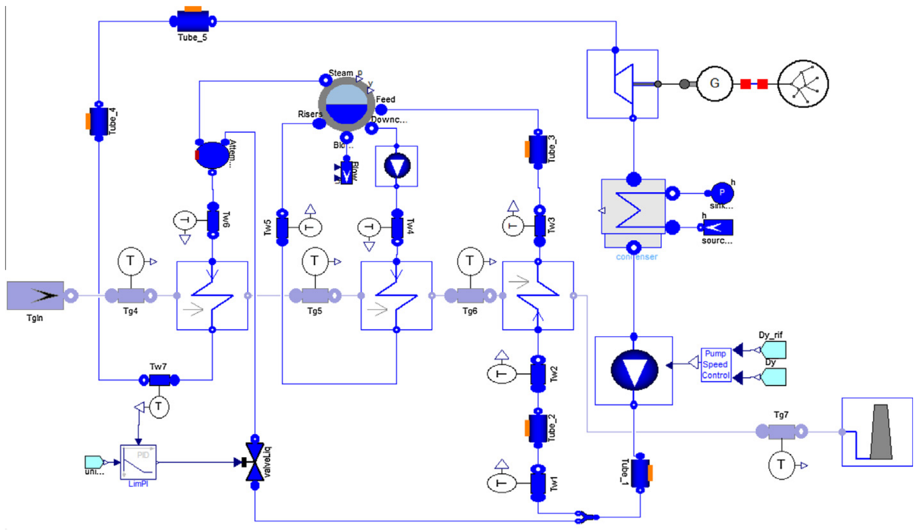

Fig. 2. Modelica object diagram of the single pressure Heat Recovery Steam Generator.

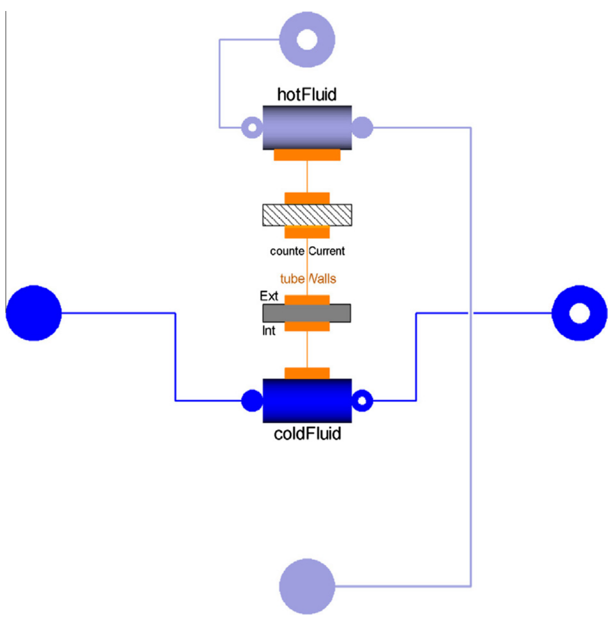

Fig. 3. Modelica object diagram of the heat exchanger model.

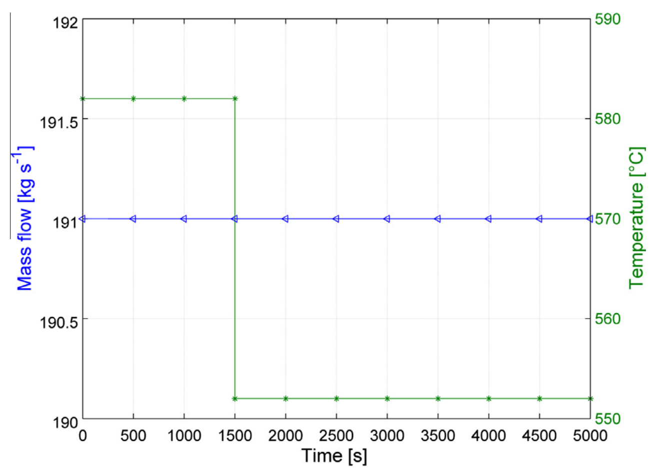

Fig. 4. Mass flow rate and temperature of the exhaust gasses at the HRSG inlet section.

reason, in the present section the three models are compared taking into account some typical transient conditions. At first, the comparison among the three models (the TPL model and the TP model developed respectively in Modelica language coupled with Thermal Power Library and ThermoPower library, and the one in Matlab/Simulink environment, MSM) is performed considering the transient condition shown in Fig. 4.

The selected test case consists in a step variation of the exhaust gas temperature at the HRSG inlet from 582 °C to 552 °C while the exhaust gas mass flow rate is assumed constant. Some interesting results of the simulation processes are reported in the following figures.

The three models have a different response in term of pressure while the trends of the temperature at the exit of the superheater are the same. As shown in Fig. 5, the models developed in Modelica have similar pressure trends despite the different way to model the inertia into the heat exchangers while the MSM model (developed in Simulink) has a faster response mainly due to the different control system. In addition, the feed pump into the TPL model is an ideal pump but is able to take into account the inertia phenomena into the machine; effect which is not implemented into the TP and MSM models. As reported in Fig. 6 the trends of the temperature at the exit of the superheater are more or less the same for the three models; therefore the heat transfer coefficients and the relative heat transfer areas are computed correctly in the three models. Obviously, it is not necessary to introduce the heat transfer areas into the TPL model because the heat exchanger component computes these surfaces starting from the inserted geometry. On the other hand, developing a model where the entire geometry of the components is taken into account is computationally expensive; in particular the simulation time is three times faster for the TP model in comparison of the TPL model. Furthermore, it is important to underline that, to evaluate the thermal stresses and the residual life, it is essential to build a detailed model that permits to determine the pressure and temperature trends into the single part of the device; as a consequence, only the model developed in DYMOLA coupled with Thermal Power Library (TPL) is able to predict thermodynamic variables with this grade of detail. However, in order to better understand the geometry's effects, the dynamic analysis and the lifetime estimation are performed

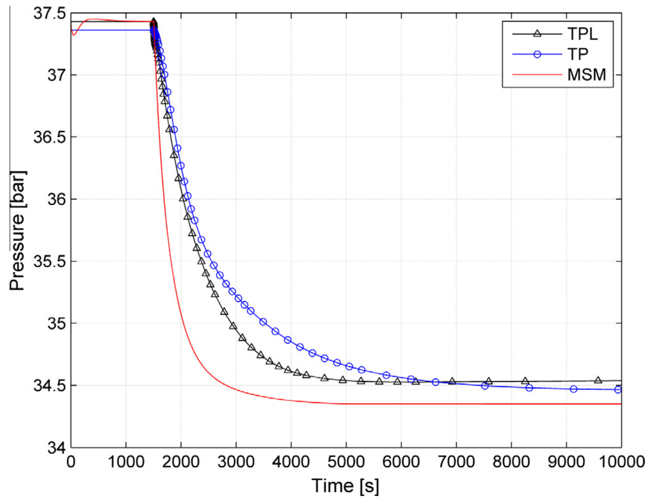

Fig. 5. Steam pressure at the drum inlet.

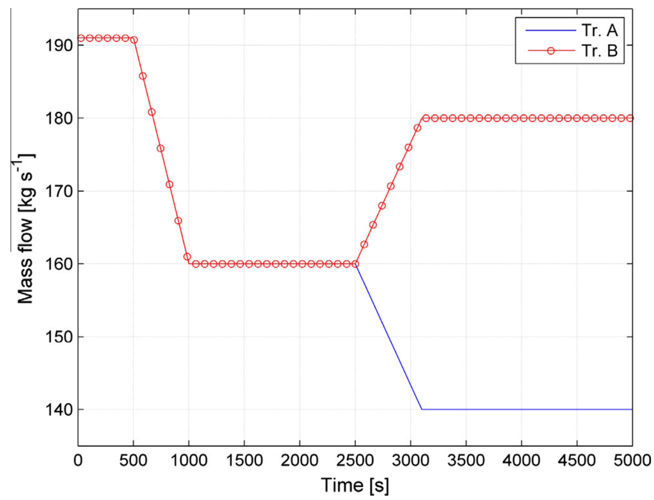

Fig. 7. The exhaust gas mass flow rate at the HRSG inlet for transients "Tr. A" and "Tr. B".

using both the TPL model (components from Thermal Power Library) and the Simulink one. The MSM model is preferred to the one built in DYMOLA with components taking from ThermoPower library (TP) because uses a different approach during the equations' solving process. In this manner, the dynamic behaviour is predicted with two different models based on two different approaches, contrived in two different simulation environments and built from two research groups.

4. Dynamic analysis

In Fig. 7 two typical transient conditions, supplied by an operator of the Italian electricity market and indicated by the acronyms "Tr. A" and "Tr. B", are sketched. The two transients coincide till 2500 s from the simulation start time, when the exhaust gas mass flow rate (and thus the steam turbine power) is equal to 84% (160 kg s^{-1} of the rated value, thereafter they become specular: in "Tr. A" the gas mass flow rate decreases up to 140 kg s^{-1}, whereas in "Tr. B" it increases again up to 180 kg s^{-1}. In both transients, the gas temperature at the inlet of the HRSG remains constant and equal to the rated value (582 °C).

As shown in Figs. 8 and 9, which report respectively the steam drum pressure and the electric power produced by the electric

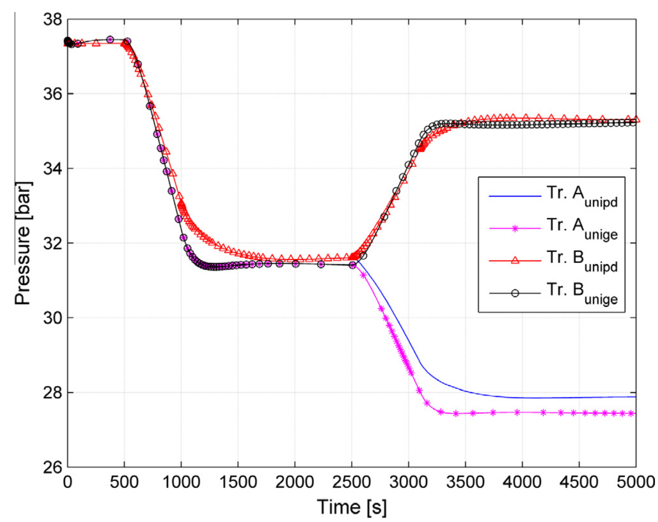

Fig. 8. The steam drum pressure during transients "Tr. A" and "Tr. B".

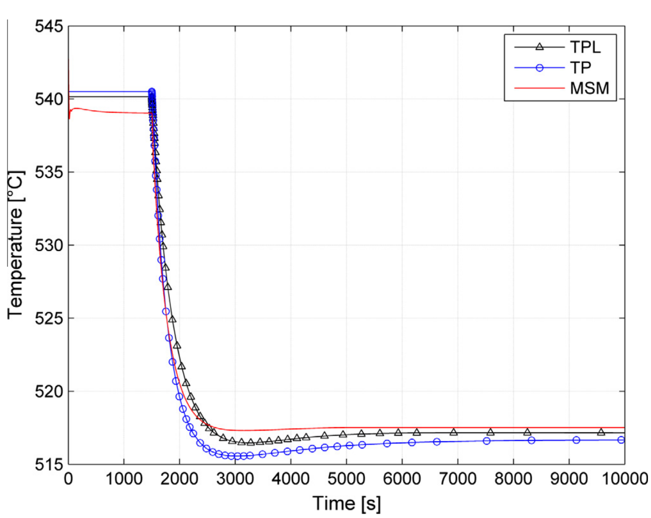

Fig. 6. Steam temperature at the superheater outlet section.

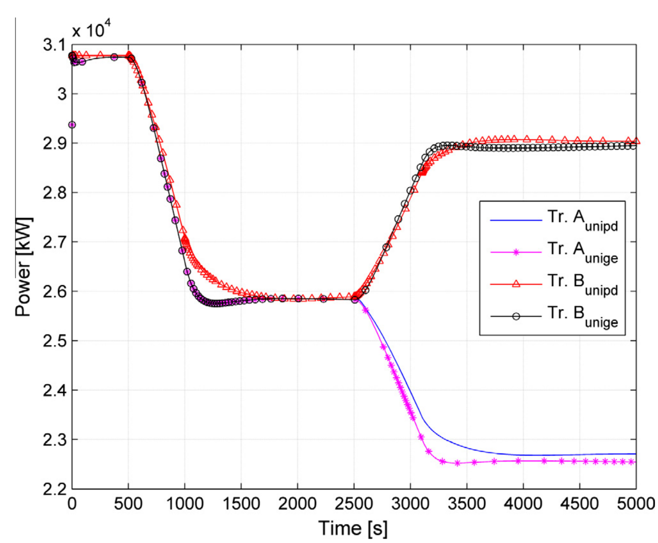

Fig. 9. Electric power produced by the electric generator during transients "Tr. A" and "Tr. B".

generator, there is a complete correspondence between the MSM model and the TPL one. In Figs. 8 and 9, as well as in the following, the results of the Simulink model are plotted in dotted line, whereas those coming from the TPL model are represented by a solid line.

5. Lifetime calculation procedure and results

In the selected test case, the analysis of the thermodynamic variables shows that the drum is the most stressed component. For this reason, the output values of the dynamic simulation models are used as input for the mathematical tool able to evaluate the steam drum's life reduction due to low-cycle fatigue.

The fatigue life calculation tool is implemented in Matlab/Simulink and considers a simple cylindrical geometry for the steam drum: an inner metal cylinder and an outer insulation cylinder. The presence of discontinuities in the metal structure of the component (due to the connection of downcomers and risers and to the drum end plates) is taken into account by assuming suitable stress concentration factors. The tool is developed according to the EN 13345 Standard [60]. The fatigue life calculation procedure can be summarized as follows:

1. Calculation of temperature and thermo-mechanical stresses (radial, circumferential and axial components), on the basis of the trend of steam pressure and temperature inside the drum, for each coaxial metal layer in which the cylinder has been divided.
2. Calculation of the three principal structural stresses for each metal layer.
3. Calculation of the signed Tresca equivalent stress for each metal layer.
4. Application of the "Rainflow Counting Method" to the signed Tresca equivalent stress, in accordance with the ASTM E 1049 Standard [61], to identify fatigue cycles for the more stressed metal layer.
5. Analysis of the results of Step 4 for each i_{th} fatigue cycle (the stress range $\Delta\sigma_l$, the mean stress $\Delta\sigma_{m,i}$ the maximum stress $\Delta\sigma_{max,i}$, the value n_i that is equal to 0.5 if the stress range $\Delta\sigma_l$ is counted as a half cycle or to 1 if it is counted as one cycle, the starting time $t_{0,i}$, the period τ_l and the average temperature T_i^*).
6. Application of the life reduction calculation procedure, developed in accordance with the EN 13445 Standard, to each fatigue cycle (identified by Steps 4 and 5) in order to compute the allowable number of fatigue cycles N_i and, as a consequence, the "cumulative fatigue damage index" D for the examined transient by applying the Palmgren–Miner rule which indicates to sum the fatigue damage indexes $LC_i = n_i/N_i$ of each cycle.

For this analysis, other three more strict transient operating conditions are selected (see Fig. 10). All of them consider a plant load reduction from the rated value to about 55%, but they differ for the decreasing time values. In particular, transient "Tr. 1" is selected as reference case (the flow rate decreases from 191 to 130 kg s^{-1} in 37.5 min while the temperature from 582 °C to 515 °C in 6.25 min), whereas transients "Tr. 2" and "Tr. 3" can be considered respectively as a "less" and a "more" stressed operating condition. They are characterized by decrease time values 25% higher and lower than that of "Tr. 1", respectively. As shown in Fig. 10, it is assumed that in all the three transients the gas temperature starts diminishing when the mass flow rate attains the value of 140 kg s^{-1}; this hypothesis is in accordance with data supplied by combined cycle operators. In particular, the selected test cases have been chosen in order to investigate the dependence of the residual life reduction on the slope of the load reduction curve.

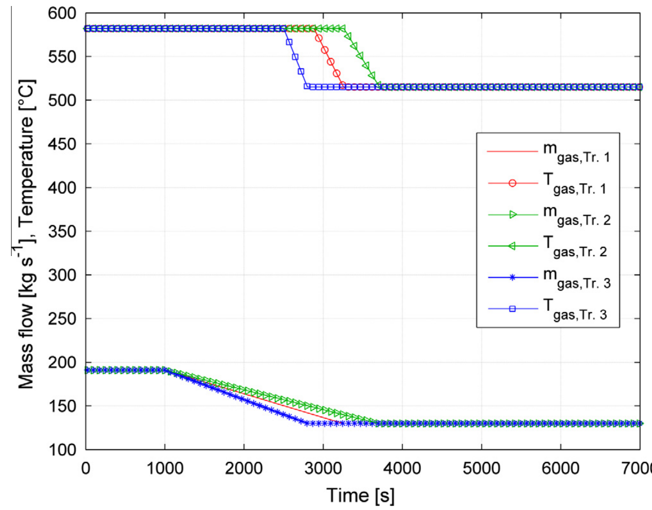

Fig. 10. Mass flow rate and temperature of exhaust gases at the HRSG inlet during transients Tr. 1, Tr. 2 and Tr. 3.

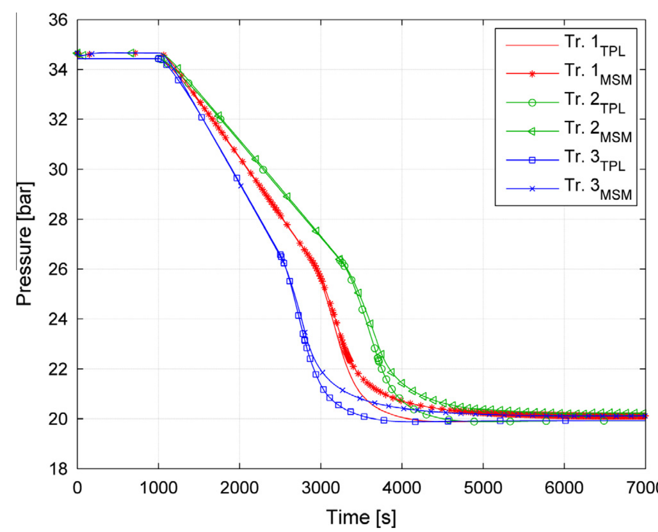

Fig. 11. Steam pressure at the superheater exit during transients Tr. 1, Tr. 2 and Tr. 3.

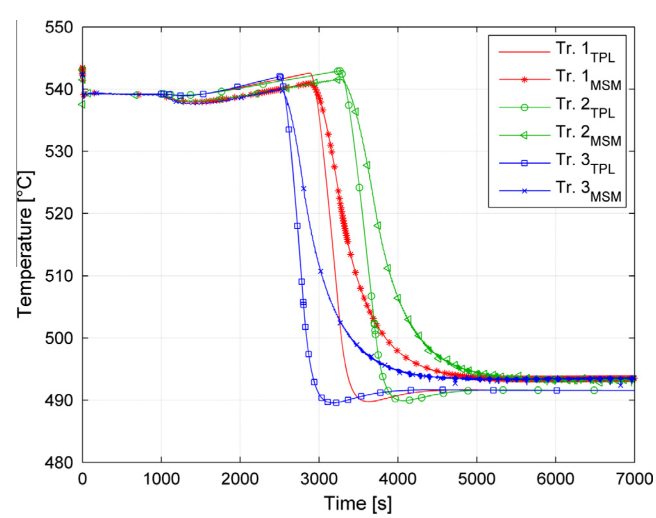

Fig. 12. Steam temperature at the superheater exit during transients Tr. 1, Tr. 2 and Tr. 3.

The good accordance between the two models, MSM and TPL, is proved also by analyzing the simulation results related to the three aforesaid transients; to this end, Figs. 11 and 12 plot the thermodynamic conditions of steam at the superheater exit as a function of time. Remember that, the three different colours (red, green and blue) indicate the three transients.

Being one of the scopes of the present work to estimate the steam drum residual life, in Fig. 13, the trend of the steam drum pressure is reported. This signal, together with that of the steam saturation temperature, is the main input of the fatigue life calculation tool. The trend of pressure is very similar to that of the signed Tresca equivalent stress, the latter shown in Fig. 14 for the most stressed metal layer: this is mainly due to the fact that, in the three examined transients, the mechanical stress (due to pressure) is predominant on the thermal stress (due to thermal gradients) since the exhaust gas temperature gradient, and so that of steam, is not so high; only near the time instant of 3000 s it can be noticed a certain deviation of Tresca signal (see Fig. 14) compared to the pressure signal (see Fig. 13) because at that time the maximum thermal stress occurs.

From the life reduction calculation, it derives that, in comparison with the transient "Tr. 1" (that is the reference case) the transient "Tr. 2" is very severe since it determines a decrease in the life reduction of about 9.4%, whereas the transient "Tr. 3" leads to an increase of about 5.3%.

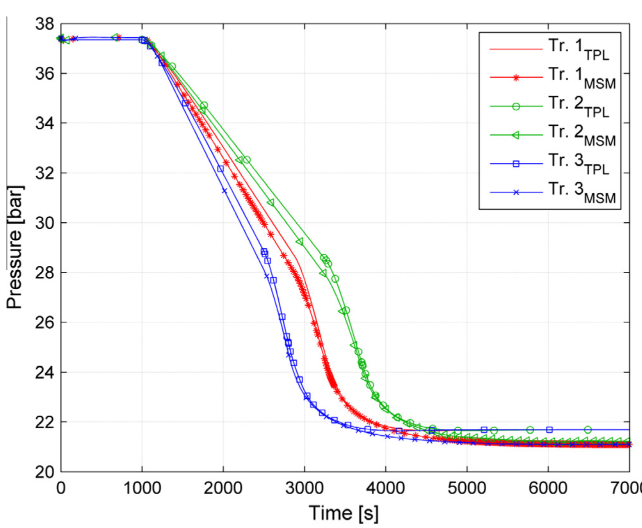

Fig. 13. Steam pressure into the drum during transients Tr. 1, Tr. 2 and Tr. 3.

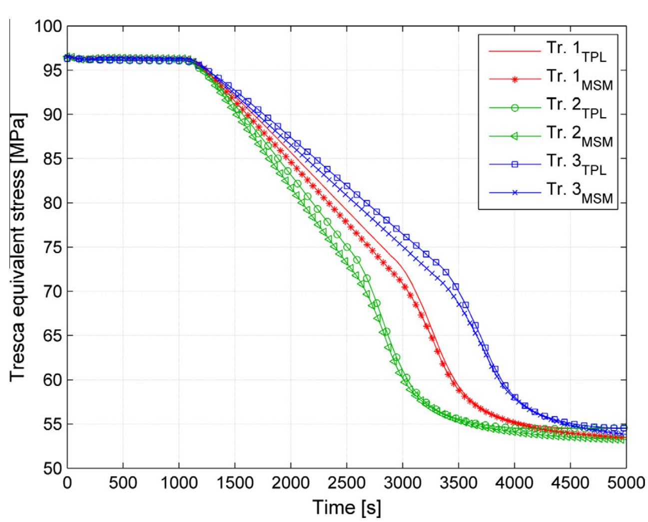

Fig. 14. The signed Tresca equivalent stress during transients Tr. 1, Tr. 2 and Tr. 3.

6. Conclusions

The role played by combined cycle power plants in the scenario of the liberalized electricity market is a fundamental aspect to take into account. The issue of the plant's lifetime reduction has been pointed out as a consequence of more and more frequent cycling operations. In this context, the availability of dynamic simulation models of power plants, able to test different operating conditions and evaluate their impact on the residual life of plant devices, is paramount. At this purpose, three different flexible simulation models for the same single pressure level combined cycle unit have been developed, tested and successfully validated.

Then, the dynamic analysis results have been used in a procedure able to estimate thermo-mechanical stresses and the associated devices' lifetime reduction. The proposed tools can be considered as valuable innovative instruments to assist power plant designers and operators in order to improve the plant's flexibility without excessively compromising the integrity of devices subjected to high thermo-mechanical stresses. Furthermore, another key factor is the user-friendly interface of the life reduction calculation tools proposed in this dissertation and their short run-time in comparison with finite elements analysis tools.

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