EXPERIMENTAL AND NUMERICAL PERFORMANCE TESTS OF THE NEW 150 MW UNIT OF CHP PAROŞENI

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The purpose of this paper is to study by both experimental and modelisation techniques the behavior of a new cogeneration unit started in commercial operation in 2007. The experimental results are compared with data obtained after running two separate original numerical models (one for the steam generator and the other for the turbine), built-up on several modules: combustion and heat transfer equations, generation of pollutant emissions, mass and energy balance of heat exchangers, pressure distribution and turbine stage efficiency, etc.

The numerical models of the steam cogeneration unit will be validated both by experimental data measured in a test performance campaign and data revealed from the warranty operational diagram delivered by the equipment supplier.

Keywords: performance tests, cogeneration steam turbine unit.

1. Introduction

The authors were invited as an independent authority to supervise the warranty performance test campaign carried out in the period 2007-2008 by a specialized company [1], together with the equipment suppliers and plant operational staff. The unit 4 is composed of one brown coal fired steam generator with flow-rate of 540 t/h coupled with one condensing steam turbine (P_b=150 MW_e) provided with a controlled extraction of heat (Q=134 MW_t) for district heating. In order to be sure that all operational regimes (P and Q) of interest for the plant management will be replicated, the tests were carried out during two seasons (late autumn-winter, spring).

After tests, we have developed a complex theoretical study of this plant, containing the verification calculus of nominal regime and a model able to validate the figures obtained during measurements and to explain its part load behavior particularities.

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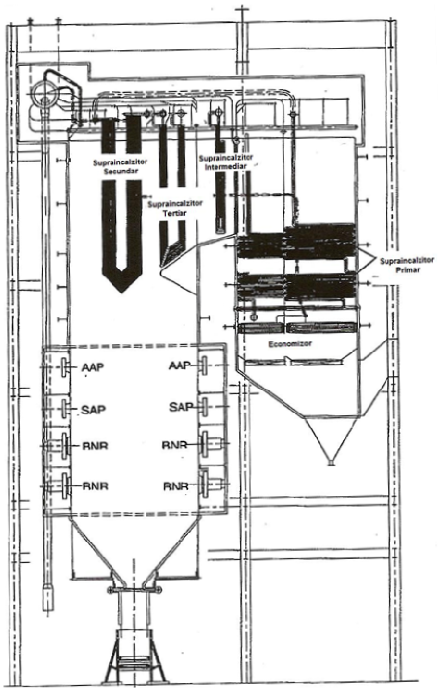
2. Rated figures of the plant

The steam boiler was designed and build-up by Babcock-Hitachi, based-on the natural circulation principle, using as fuel the pulverized hard coal in association with natural gas at startup and backup. The cross-section and the main parameters of the boiler are shown in Figure 1.



Main operating parameters

Live steam	
Mass flow-rate:	150 kg/s
Temperature	541 °C
Relative pressure	13.85 MPa
Reheated steam	
Mass flow-rate:	130.94 kg/s
Intake temperature	350.7 °C
Outlet temperature	541 °C
Intake relative pressure	3,12 MPa
Outlet relative pressure	2.96 MPa
Feedwater	
Intake temperature	235.4 °C
Hot gasses	
Exhaust temperature	143 °C
Exhaust air excess coefficient	1.30
Overall efficiency	90.7 %

Fig. 1. The cross-section and the main parameters of the boiler

In order to ensure the fuel supply for the 16 burners, there are in operation three coal mills (with a rated load of 12 kg/s) while one is kept as backup. Design coal characteristics are the following:

C^{mc*} = 75-78 % (78 % warranted); H^{mc} =3.5-4.7 % (4.7 % warranted)
S^{mc} = 1.6-2.1 % (1.9 % warranted); N^{mc} = 0.7- 1.5 % (1,2 % warranted)
O^{mc} = 12 % (12 % warranted); W_t^{i**} = 11.2-6.0 % (10.7 % warranted)
A_i =46-33.7 % (38.0 % warranted); LCV= 13.8-18.9 MJ/kg (16.4 MJ/kg warranted)

Ash analysis is provided by Table 1:

* mc – combustible mass state
** i – initial state

Table 1

Ash analysis

Melting temperature	1410 °C
SiO_2	46,43 %
Al_2O_3	17.5 %
Fe_2O_3	10.3 %
CaO	6.7 %
MgO	1.2 %
Na_2O	3.8 %
K_2O	0.74 %
SO_2	3.1 %

The steam turbine named K-160-130-2PR2, was design and built by HMZ (Ukraine) and operates at live steam parameters of 12.8 MPa / 535 °C and a reheat temperature of 535 °C. The rated cooling water temperature is 12 °C and allows a design value of the electrical power of 170 MW, larger than the actual rated value of 150 MW. That means that at higher cooling water temperatures it is still possible to reach actual rated power.

In figure 2 is presented the configuration of the thermal circuit, which is a little bit different of the real circuit of the plant, based-on the old configuration existing on the site. You can see the condenser, two condensing pumps and a water quality treatment unit, three low pressure water heaters, a deaerator, the feedwater pump and two high pressure water heaters.

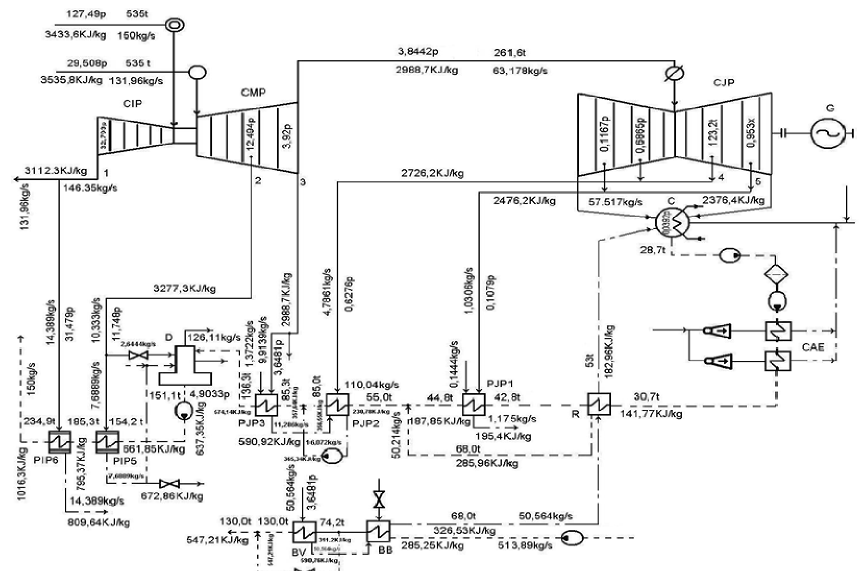


Fig. 2. Thermal circuit rated parameters

Using the geometrical and operational data of the plant equipments, a verification calculus at rated regime have been made, using the algorithms presented in [2], [3]. Values of the main parameters are also presented in figure 2.

3. Part load model

The ***steam boiler*** model is based on few hypotheses [3], such as:

- All part load regimes are stabilized;
- Live steam pressure and temperature remain constant during boiler charges between 70-100 %.
- Feedwater temperature represents an input value exported from the heaters model.

When the diagram of hot gasses enthalpy $I_g(\lambda,t)$ is not available, the model of the boiler is complex. Here there are presented the main equation:

$$t_{exh}^{'} = \left(t_{exh} + 273.15\right) \cdot \left(\frac{S}{100}\right)^{0.2} - 273.15 \tag{1}$$

where $t_{exh}^{'}$ - hot gasses temperature at part load, [°C];

t_{exh} - hot gasses temperature at nominal regime, [°C];

S - boiler's charge, [%].

$$Q_u^{'} = \left(0.009 \cdot S + 0.1\right) \cdot Q_u \tag{2}$$

where $Q_u^{'}$ - useful heat at part load, [kW];

Q_u - useful heat at nominal regime, [°C];

$$B^{'} = \frac{Q_u^{'}}{\eta^{'} \cdot Q_d} \tag{2}$$

where $\eta^{'}$ - boiler's efficiency at part load, [°C];

Q_d - available heat at nominal regime, [°C];

The steam turbine model is in fact the "hybrid method" described in [2], which is an iterative method with main following rules:

- Stodola's equation for pressure distribution at part loads, using the relative ratios for mass-flow rate (R_D) and absolute temperature (R_T):

$$p_0 = \sqrt{p_2 + R_D^2 \cdot R_T \cdot \left(p_{0n}^2 - p_{2n}^2\right)} \tag{3}$$

- Relative internal efficiency of a stage or group of stages $\overline{\eta}$:

$$\overline{\eta}_i \ = 2{,}1 \cdot \overline{r}_v - 1{,}19 \cdot \overline{r}_v^2 + 0{,}09 \cdot \overline{r}_v^3 \tag{4}$$

where

$$\overline{r}_v \cong \sqrt{\frac{H_{tn}}{H_t}} \tag{5}$$

that means the square root of isentropic drop in nominal, respectively part load regime.

- Energy balances for each heater are performed every iteration, in alternation with turbine line expansion calculation.

4. Results of the tests

In the period November 2007 - April 2008, 11 tests have been performed at unit 4 of CHP Paroşeni, but only 8 have been used for comparison with the model results and are presented in table 2 [1].

Table 2

Experimental tests on unit 4 of CHP Paroseni

Param./Nr	1CHP	2CHP	3CHP	4CHP	6CHP	9C	10C	11C
P [MW_e]	120.2	125.0	113.1	149.8	90.1	149.9	126.9	113.2
Q [MW_t]	91.6	94.9	86.6	19.5	19	0	0	0

In order to evaluate the boiler performances, we decided to keep only the first 4 tests from table 2 and to compare some operational parameters (*mes*-measured, *calc*-calculated). In Figure 3 is shown the fuel consumption versus steam mass flow-rate, where the measured values are much lower than the calculated ones. The same trend shows the air excess coefficient at stack (λ_{exh}), which is lower in tests than in calculus.

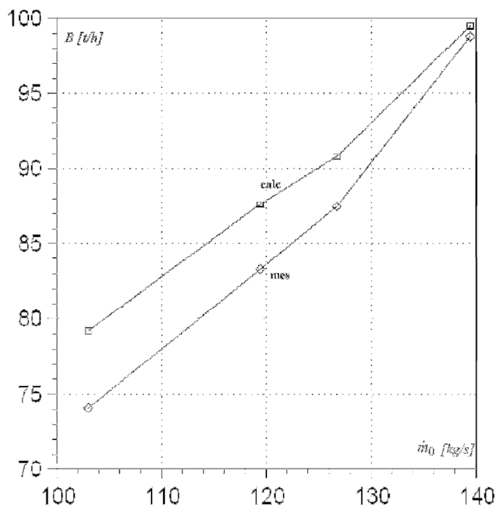


Fig. 3. Fuel flow-rate

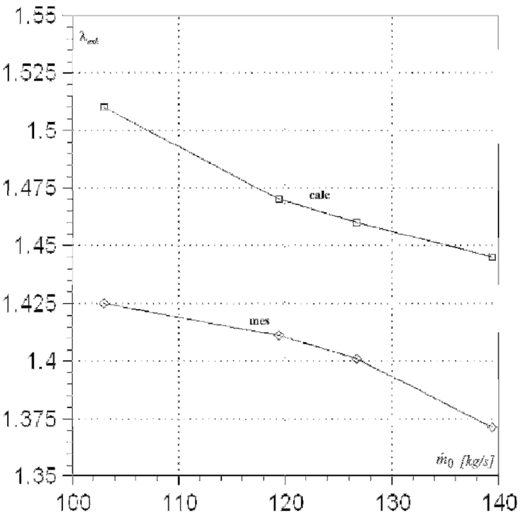


Fig. 4. Air excess coefficient

Other proves that the real behavior of the boiler in operation is much better than the figures resulted from the part load model are given by Figures 5 and 6. Thus, in Figure 5, the calculated exhaust temperature of hot gases increases with the charge diminution, while the same measured parameter is quite constant.

But the strongest evidence of this statement is represented by the comparative variation of the boiler efficiency, presented in figure 6. It is obvious that the measured values are higher than those resulted from the model.

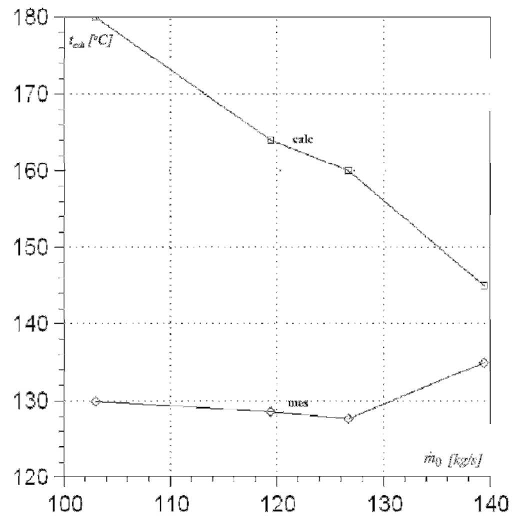


Fig. 5. Flue gasses exhaust temperature

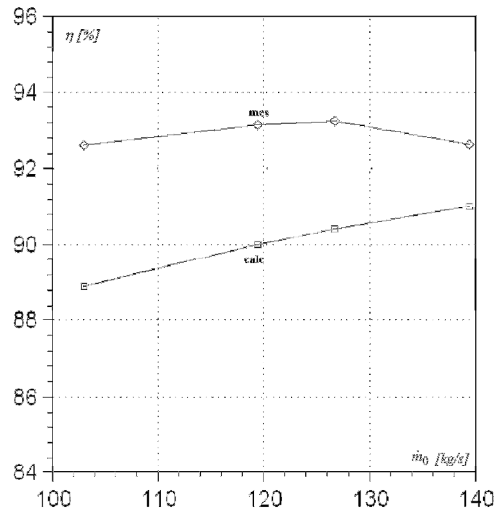


Fig. 6. Boiler efficiency

For the steam turbine evaluation, we wish to present two Figures which express some particularities related to the part-load operation. The first one is Figure 7, where is shown the calculate evolution of the internal power of the control stage during load decrease.

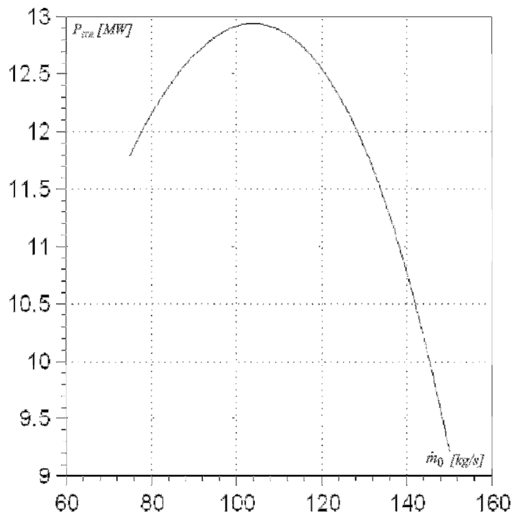


Fig. 7. Control-stage internal power

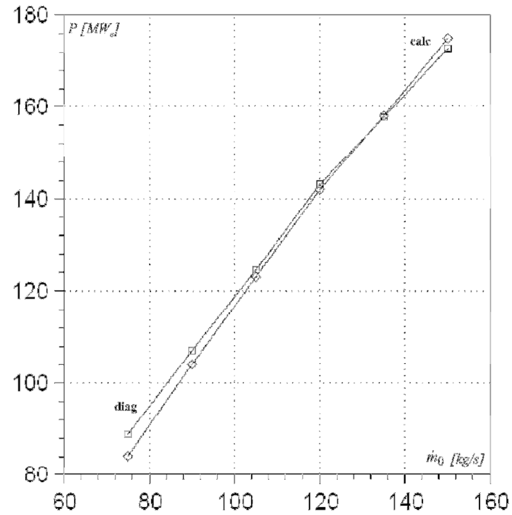


Fig. 8. Electrical power versus steam flow-rate

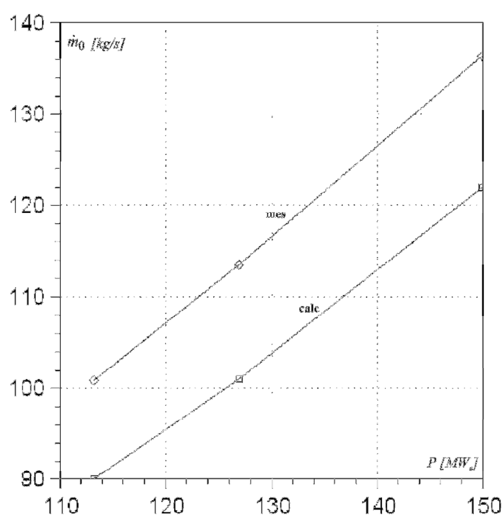


Fig. 9. Electrical power in "C" regimes

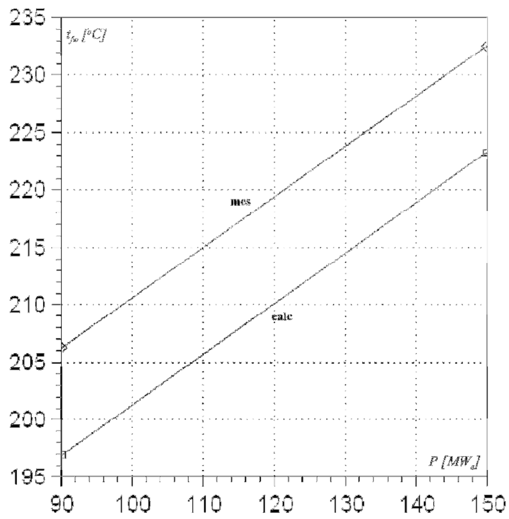


Fig. 10. Feedwater temperature in "CHP" regimes

Thus, when the steam flow is reduced, the pressure after control stage diminishes too, contributing to the isentropic drop augmentation. In this manner, the stage power starts to rise until a maximum, after that, the influence of steam flow decrease is more important than isentropic drop increase. Figure 8 shows the dependence of steam flow versus electrical power, comparing the results of the calculus with those read from the designer operational diagram. Because both are theoretical approaches, the values are very close. Not the same situation we find in Figures 9 and 10, where measured values of the steam flow and feedwater temperature are higher then the calculated ones, indicating a bad internal efficiency of the turbine, increased losses in the internal and external labyrinth seals,

6. Conclusions

The tests have demonstrated that the boiler performance was better than its design values, including part-load behavior, while the turbine has proven poorer efficiencies than predicted values.

Due to the fact that the (new generation) boiler measured parameters were better than those calculated, a process of recalibration of the model should be performed. The model is useful for the operational staff in order to evaluate the plant behavior in futures regimes and to select the right ones.

R E F E R E N C E S

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