Methodology and software for prediction of cogeneration steam turbines performances

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Abstract

Combined Heat and Power generation (known also as cogeneration) represents one of the main ways for increasing the efficiency of primary energy use and consequently for decreasing the CO_2 emissions. Cogeneration steam turbines have a complex configuration and a wide range of possible operation regimes, depending on the level of heat and power demand.
This paper presents a methodology for predetermination of cogeneration units' performances, starting from the heat and power demand. The methodology is used for developing specialized software that can be used during the CHPP operation. A case study is presented for 50 MW units, which are the most encountered in the Romanian CHPP's.

Keywords: Steam Cycles, Cogeneration, Mathematical model.

1. Introduction

Design and off design steam turbines performances, running into a known thermodynamic configuration, can be predetermined by calculus. The main steam turbine elements needed for this determination are: internal isentropic efficiency, exhaust energy losses, labyrinths losses, pressure and temperature variation into characteristic points of the steam turbines.

For steam turbines (with power, steam parameters, flow distributions into design regime known) we propose a method for a rapid computation of any functioning regime. The method is based on the findings given by General Electric [1,2], Samoilovici, Troianovski and Scegliaev [3,4] and Alstom's [5].

2. Mathematical model for steam turbines calculus

Our mathematical model depends on constructive and functional characteristic of cogeneration steam turbines stages. The turbine stages belong to two categories:

- Control stages, with steam flow rate control and power generation purposes. The control stage have adjustable geometry for income flow section;
- Pressure stages, with only power generation purpose. This stage type has fixed geometry.

Internal isentropic efficiencies are calculated with equation 1 for control stages and equation 2 for pressure stages:

$$\eta_i^{TR} = \left(0{,}83 - \frac{0{,}15}{D_1 \cdot v_1}\right) \cdot \left[1 - 0{,}04 \cdot \left(1 - \sqrt{\frac{h_{iz}^n}{h_{iz}}}\right) - 1{,}165 \cdot \left(1 - \sqrt{\frac{h_{iz}^n}{h_{iz}}}\right)^2\right] \qquad (1)$$

$$\eta_i^{TP} = \eta_i^{TP,n} \cdot \sqrt{\frac{h_{iz}^n}{h_{iz}}} \cdot \left(2 - \sqrt{\frac{h_{iz}^n}{h_{iz}}}\right) \qquad (2)$$

Into equations 1 and 2: η_i^{TR} is internal isentropic efficiency for control stages, η_i^{TP} is internal isentropic efficiency for pressure stages, $\eta_i^{TP,n}$ is internal isentropic efficiency for pressure stages on design regime, D_1 is steam flow rate before control stage nozzle into kg/s, v_1 specific volume before control stage nozzle into m3/kg, h_{iz}^n is isentropic stage detent on design regime into kJ/kg and h_{iz} is isentropic stage detent on off design regime into kJ/kg.

Internal isentropic efficiencies are calculated for one stage in the case of control stage. Usually, pressures stages are grouped into characteristic zones for efficiency calculus. These zones are formed between each extraction or between extractions and exhaust of turbines.

If steam detention is into a humid turbine zone, then isentropic efficiency is corrected like in equation 3.

$$\eta_i^{TP,h} = \eta_i^{TP} \cdot \frac{2 - u_i - u_e}{2} \tag{3}$$

Into equation 3: $\eta_i^{TP,h}$ is internal humid isentropic efficiency for design regime, u_i is humidity at zone entrance and u_e is humidity at zone exit.

At turbines cylinder exhausts steam have kinetic energy that, from the process point of view, will be lost. Residual losses are calculated with equation 4.

$$\Delta h_{rez}^h = 0,87 \cdot x_e \cdot (0,65 \cdot x_e + 0,35) \cdot \Delta h_{rez} \tag{4}$$

Into equation 4: Δh_{rez}^h is residual lost in J/kg, x_e is steam title, w_e is steam exhaust speed in m/s and Δh_{rez} is a function of steam speed and turbines construction.

An important step into the method is pressure distribution calculus. Through this we understand calculus on every characteristic point of the thermodynamic circuit. As a basis for steam turbines pressure calculus is design and off design flow distribution. With the assumption that medium temperature variation is ignorable, pressure at one zone entrance is calculated (equation 5) [6].

$$p_i = \sqrt{(p_e)^2 + \left(\frac{D}{D^n}\right)^2 \cdot \left[(p_i^n)^2 - (p_e^n)^2\right]} \tag{5}$$

Into equation 5: p_i is pressure at zone inlet for current regime in bar, p_e is pressure at zone outlet for current regime in bar, p_i^n is pressure at zone inlet for design regime in bar, p_e^n is pressure at zone outlet for design regime in bar, D is current regime flow rate in kg/s and D_n is design regime flow rate in kg/s.

Stodola equation can be applied only to fixed geometry parts, consequently equation 5 can be applied only to pressure stage and starting from steam turbines condensers.

Steam losses from turbines cylinders labyrinth sealing and control valves cause a decreasing of the turbine steam flow and of the generated power. For this reason, labyrinth sealing and control valves steam losses must be precisely known. The losses flow rate can be calculated with equation 5.

$$D_L = C \cdot \sqrt{\frac{p_1}{v_1}} \tag{6}$$

In equation 6, D_L is labyrinth sealing flow rate losses through the pack in kg/s, p_1 is pressure before pack in bar, v_1 is specific volume before pack and C is a constant statistically established.

3. Calculus Algorithm

Cogeneration steam turbines power calculation can be effectuated when heat for thermal consumers demands are known. The iterative algorithm for power calculation when heat demands are known is presented in figure 1.
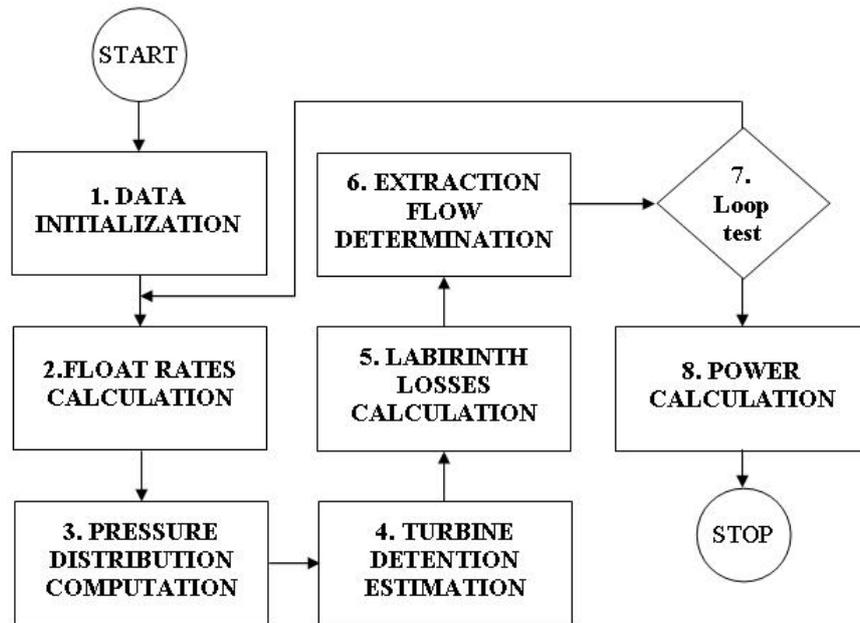

Figure 1. Cogeneration steam turbines calculus algorithm

The presented algorithm has 8 modules. In "data initialization" step, design extraction flow rates are given. For the analyzed regime extraction flow rates are determinate using heat and flow balances equations applied to the steam turbine feed water preheaters. The loop is broken if differences between extraction flow rates from last and actual iteration smaller then an imposed one.

4. Case study

DSL 50 is the most encountered in the Romanian CHPP's. The turbines have 50 MW installed power, three consumer extraction, two for high, respectively intermediate pressure industrial consumers and one for low pressure urban consumers.

The methodology presented before was verified for this installation. Main turbine parameters for characteristic tested regimes are given in table 1.

Table 1. Tested characteristic parameters for steam turbine DSL 50.

Parameter	I	II	III	IV
Live steam pressure, bar	127.4	127.4	127.4	127.4
Live steam temperature, °C	565	565	565	565
Live steam flow rate, t/h	292	230	185	240
Industrial extraction pressure, bar	13	16	13	13
Industrial extraction flow rate, t/h	115	0	0	95
Urban extraction pressure, bar	1.2	0.7	-	1.2
Urban extraction flow rate, t/h	86	160	0	70
Condensing pressure, bar	0.03	0.03	0.053	0.03

Calculated and measured generated powers are presented, together with the relative difference are presented in table 2.

Table 2. Generated power for tested DSL 50 steam turbine

Regime	Measured power, kW	Calculated power, kW	Relative difference, %
I	50,081	49,991	0.18
II	50,390	50,597	0.41
III	49,120	49,267	0.30
IV	40,382	40,044	0.83

From table 2 on observe small relative differences between measured and calculated power.

As seen, on this case study independent energy demands on different levels (three thermal levels and one electrical) are required for DSL 50 steam turbine. Employing of the presented methodology can be important for the optimization of energy generation and assure fuel consumption reduction.

The software has been applied to several CHPP equipped with such steam turbines: Bacău CHPP, Braşov CHPP and Halanga CHPP. Good results are obtained for loads into 50 ÷ 100 % area.

5. Conclusions and future work

The methodology and associated software have the possibility to be customized according to specific characteristic of a given CHPP. Other turbines in the DSL 50 family were modeled using the same methodology. DKUL 50 is a 50 MW installed power with urban backpressure, and two consumer extractions. DKU 50 is a 50 MW installed power with backpressure and one consumer extraction. Also for these models of steam turbines, our methodology and associated software were tested with good results in the same area as our presented case.

The methodology and associated software can be extended to other steam turbine types such condensing steam turbines with or without reheat. This category of steam turbine (that does not have industrial or urban extraction) can be calculated with one simpler algorithm and we predict a higher calculus precision.

Future work will be done for extensive testing and improving of this algorithm in cooperation with power generation companies that own CHPP or condensing steam turbines.

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