The Specificity of Steam Turbine Stages Working in the Low Mass Flow Rate Regimes and Keeping them in the Simulation of Hydrodynamic Processes in the Turbine in Steamless and Motor Modes

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***Abstract*— Operating peculiarities of the stage, the group of stages and the flow part of the steam turbine in low-flow modes, in particular, in the steamless and motor modes, and methodological approaches to their registration in the simulation of hydrodynamic processes are considered. Particular attention is paid to determining the power losses for friction and ventilation. The technique and algorithm for calculating the temperature state of the flow part of a steam turbine are given. Based on the conducted studies, the regularities characteristic of the motor regime of most steam turbines are revealed.**

***Keywords*— steam turbine; small steam mode; distinctive features; steamless and motor modes; mathematical model; power losses; flow separation; temperature graph; algorithm; model studies; typical patterns**

I. Introduction

The growth of power consumption is currently accompanied by a shortage of electricity production and an increase in the unevenness of load schedules. The increase in the share of highly efficient power units, including on the basis of steam-gas technologies in the total capacity of power systems inevitably leads to a need to attract them to the regulation of the graphs of electrical loads and increasing the duration of their work at partial loads and redundancy modes, including with the use of stopping-starting modes with the passage of the failures of the graphs of power consumption. It is obvious that the operation of the equipment under variable loads and frequent starts and stops leads to its increased wear, causing a decrease in efficiency and reliability. It is particularly difficult to prevent a decrease in the reliability of turbine units at fast start required to regulate the load of the power system. At the same time, earlier large-scale studies of steam turbines operation in the motor mode have shown a real possibility of a significant increase in the reliability of their operation in the above modes [1, 2].

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II. Features of Steam Turbine Stage Working in Low Mass Flow Modes

Under the low-steam (steam-free, motor) mode, such a mode of operation of the stage or group of stages of the turbine is adopted, when the steam is sucked into the flow part of the turbine through the turbine seals, which in these modes operate in the opposite direction of steam flow to the nominal mode, or a small amount of steam is supplied to the flow part of the corresponding parameters for cooling the stages in the mode of forced idle rotation. At the same time, the steam moving along the flow part of the turbine at a low speed, performs only the function of cooling the stages and guide blades.

The work of the group of stages in low steam mode is a deeply non-calculated mode, which requires taking this factor into account when modeling hydrodynamic processes occurring in the turbine stages. Features of the stage and group of stages in this mode are due to the following factors:

- due to the low flow rate of steam and a slight drop in its pressure, there is no expansion of the steam in the stage and it is in the idle rotation mode;
- the flow part of the turbines (except for turbines with intermediate steam overheating) is under steam pressure in the condenser, which leads to the fact that the condenser receives superheated steam;
- unavoidable power losses during idle stage rotation (friction, ventilation, etc.) in the absence of cooling steam lead to heating of steam and metal stage;
- heat loss to the outside environment via the external surface of the turbine housing, usually ignored in calculations of turbine stages in the design modes because of their smallness (in comparison with the amount of heat flow of steam through the stage), low mass flow modes is comparable to the heat loss due to friction and ventilation (especially for steps CHP) and can affect the temperature state of the metal of the guide vanes and the housing;

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- in part of the turbine stages, steam flow is forced into the peripheral part of the blade to form vortex flows at the root of the blades, while at high vacuum in the condenser, the reverse flow is sucked up to the guide blade of the last stage of the turbine and discharged with the main steam flow into the condenser without the formation of vortex flows [1, 3, 6].

In the absence of special highly maneuverable power units, the motor mode is used to improve the maneuverability, reliability and efficiency of the equipment of thermal power plants, originally designed for operation in basic modes [1]. Large-scale experimental studies on the introduction of the motor mode on condensation and heat-generating units were carried out by a number of scientific, commissioning and operational organizations. As a result, the motor mode was implemented on many steam turbines with a capacity of 50-200 MW [1, 6, 8], on a wet-steam turbine K-220-44 type (Kolskya NPP) [9]. The calculated studies have shown the possibility and efficiency of the application of the motor mode also in the turbine units with supercritical steam parameters with a capacity of 300 and 800 MW. Work on the transfer of turbine units to the motor mode was also carried out abroad on turbines with a capacity of 50-200 MW (Cuba, Bulgaria, Japan) and on a turbine unit with a capacity of 500 MW to critical parameters (TPP in the US) [5]. In modern conditions for power systems, in which the main power input is made at the expense of high-efficient, but low-maneuver power units of nuclear power plants, CCGT, etc., the relevance of the application of the motor mode increases. Currently, model studies are underway to determine the feasibility of reserving the power of CCGT in condensation and heating modes by converting the steam turbine into a motor mode [11, 14].

The first attempts of developing a mathematical model of stages and groups of stages of the steam turbine when it is working in low mass flow modes done in [6, 10], however, due to the adopted by the authors of the assumptions and simplifications they have developed models give satisfactory results only in steamless modes. On the basis of the general idea of the process in the turbine stage in low-steam modes in [1] a mathematical model of the stage and a group of stages in their operation in steam and motor modes is created. This report presents the main provisions of this model and the calculated dependences for the calculation of the total power losses for friction and ventilation, the algorithm for calculating the temperature state of steam and blades in the turbine flow and some results of the generalization of the calculations.

III. Common Approaches to Drawing Up Mathematical Models of the Stages of the Steam Turbine in Low Mass Flow Modes

The mathematical model of the turbine stage is based on the well-known equations of one-dimensional flow:

- mass balance, which links the change in the mass of steam at the exit of the stage in transient modes with the change in its density in the volume of the stage;
- the equation of the amount of motion for a section of a straight channel in a one-dimensional flow;
- the equation of state for superheated steam (at low steam operation at all stages high temperatures at low pressure):
- the energy balance equation is expressed by the sum of all types of energy supplied to and from the system.

The amount of heat supplied to the pair, in general, can be written in the form:

$$dq = dq_{FV} \pm dq_M - dq_S + dq_{SH}, \tag{1}$$

where dq_{FV} – heat release in the stage due to energy losses for friction, ventilation and other losses; dq_M – the heat coming to the metal heating if the steam temperature is higher than the metal temperature (sign -) or the steam transmitted by the metal in cases when its temperature is higher than the steam temperature; dq_S – heat loss to the environment through the outer surface of the housing; dq_{SH} – heat leaks through the turbine rotor shaft.

When transition a steam turbine into a motor mode in the period before the stabilization of the temperature state of steam, several modes of operation of the stages are possible (it is assumed that the temperature of the metal of the stage is equal to the temperature of the steam):

1. The steam temperature in the stage is lower than the metal temperature of the stage and the body. This mode is typical for the first stages of all types of CHP turbines and CMP steam turbines with intermediate steam overheating, where the steam comes from the front end seals and due to significant throttling has a lower temperature than the temperature of the metal stage and housing. The heat is transferred from metal to steam, and the heat loss to the environment is due to the accumulated heat in the metal housing. The expression (1) for this mode is written as

$$dq = dq_{FV} + dq_{MB} - dq_{MH}, \tag{2}$$

where

dq_{MB} – heat transferred to the metal of the blade device to increase its temperature; dq_{MH} – heat transferred by steam from body parts.

2. The steam temperature in the stage is higher than the metal temperature of the body. This mode is typical for turbine stages with relatively long blades, for which significant losses of power for friction and ventilation lead to a rapid increase in steam temperature. For this mode, the balance equation will take the form:

$$dq = dq_{FV} - dq_{MB} - dq_{MH} \tag{3}$$

3. Mode with changing the direction of heat transfer in time. This mode is typical for the steps of the average height of the blades, when in the initial period of time the temperature of the steam is lower than the temperature of the metal, but in the future there is a slow increase in the temperature of the steam due to the loss of power for friction and ventilation and over

time it becomes higher than the temperature For this mode of stages, the temperature condition of the turbine blades must be calculated by the appropriate expressions (2) or (3), which requires monitoring the dynamics of the steam temperature in the stage.

From a thermodynamic point of view, it is also possible a number of modes of operation of a stage depending on the flow rate and the steam parameters at the stage:

4. Operation mode of the stage in the absence of steam expansion, which is typical for all stages of the turbine in steamless mode and for stages with relatively long blades in low steam modes. The mode of operation of the stage with steam expansion, i.e. with the performance of work is typical for stages with an average length of the blades at a certain steam flow through the stage.

5. As mentioned above, the steam pressure in the turbine flow is mainly dependent on the pressure in the condenser. For the CMP and CLP of turbines with intermediate steam overheating and the entire flow part of the turbines without it, this connection is obvious, since they have a direct connection with the condenser and in low-flow modes, all cylinder stages are under vacuum. The supply of cooling steam to the flow part of the turbine slightly increases the steam pressure in the 2 stages closest to the steam supply point in the course of steam movement, but this increase slightly affects the operating mode of the stage. Based on this, when compiling a mathematical model of the stage, it is assumed that the vapor pressure in the stage is constant.

One of the main values that determine the temperature state of the steam turbine flow part is a set of energy losses for friction and ventilation. Despite the wide application of low-steam modes in the operation of heat-generating turbines, the techniques available in the technical literature for determining these losses give disparate results [1, 5, 7, 8].

The heat released in the stage due to the total loss of power for friction and ventilation, due to vortex flows, etc., is presented in the form of:

$$dq_{FV} = \frac{N_{FV}}{G}\,d\tau = \frac{\Delta N_F + N_V}{G}\,d\tau, \tag{4}$$

where N_{FV} – total power losses for friction and ventilation; G – steam flow through the stage.

The following expression was used to calculate the friction power loss of the disk (ΔN_F) [1]:

$$\Delta N_F = 0{,}05({}^{2S_g}\!/_{d_s})\,\mathrm{Re}^{-0{,}2}\,u_c^3\rho_s, \tag{5}$$

where d_s – the average diameter of the stage; u_c – the circumferential velocity at the middle diameter; $2S_g$ – the average gap between the disk and the diaphragm; Re – Reynolds number.

There are many empirical formulas available in the technical literature to determine ventilation power losses (T_V). In [6] on the basis of generalization of a large number of calculated and experimental data for rough estimation the dependence is proposed:

$$N_V = 11500\, l d_{av}^3 P_c, \tag{6}$$

where P_c – condenser pressure.

The ventilation losses in this study are determined by the expression, taking into account the specific features of the various stages of the turbine [5]:

$$N_V = 0{,}0068\, d_{av}^4 l_2^{1,5} n\rho_s, \tag{7}$$

where l_2 – the rotor blade length; n – rotational speed.

For blades with separation of steam flow in the root zones of the blade, the expression given in [1] was used to calculate the ventilation power losses:

$$N_V = 1{,}57.10^3 C d_s l_{GCD} ({}^{u}\!/_{100})^3 \rho_s, \tag{8}$$

where C – the coefficient depending on the group of regime and geometric factors is determined by experimental dependences taking into account the actual combination of gratings, quotient $({}^{1}\!/_{d_s})$ and the relative width of the blade [1]; l_{GCD} – the height of the separation zone, which is determined in [1] on the basis of a generalization of the calculated and experimental studies, is given an empirical dependence.

The heat coming to the heating of the metal blades (dqM_B) and the metal given to the housing parts pair (dqM_H) is calculated by the well-known heat transfer formulas taking into account the above assumptions.

Consideration and calculation of the mathematical model in this paper is carried out in the absence of steam (dl = 0) and at a constant time pressure of steam in the stage under the following assumptions: 1) the temperature of the metal blades and its change in time are equal to the temperature of steam and its change (due to the high values of the heat transfer coefficient of steam); 2) the temperature of the steam in the stage changes from the initial value T0 at the entrance to the stage to the value T at the exit from it;

Taking into account these assumptions, the following calculated dependence is obtained on the basis of the described mathematical model of the steam turbine stage [9, 10]:

$$T = T_0 + \left(\sqrt{\frac{\beta \cdot p}{c_p GR} + \left(\frac{T_0}{2}\right)^2} - \frac{T_0}{2}\right)\left[1 - \exp\left(-\frac{c_p G}{c_m m}\tau\right)\right], \tag{9}$$

where

T_0 – steam temperature in the stage at the time of transfer of the steam turbine in steam or motor mode; c_p, c_m – heat capacity of steam and metal; m – mass of metal blade; P – steam pressure in stage; R – gas constant; β – coefficient depending on the geometric characteristics of the stage, the steam flow rate and its parameters and constant for the given stage.

$$\beta = \left[0{,}1 \cdot \left(\frac{2S_G}{d_s} \right) \mathrm{Re}^{-0{,}2} \, \frac{u^3 d_s^2}{2} + 6{,}8 \cdot 10^{-3} d_s^4 l_2^{1{,}5} n \right], \qquad (10)$$

To calculate the steam temperature by steps, it is assumed that its value at the entrance to the next stage is equal to the steam temperature at the exit from the previous stage, i.e.:

$$T_{i+1} = T_{i0} + \Delta T_{i+1}$$

$$\Delta T_{i+1} = \left(\sqrt{\frac{\beta_{i+1} p_{i+1}}{c_{pi+1} GR} + \left(\frac{T_{0i}}{2} \right)^2} - \frac{T_{0i}}{2} \right) \left[1 - \exp\left(-\frac{c_{p_{i+1}} G}{c_{mi+1}} \tau \right) \right], \qquad (11)$$

The initial data for the calculation of the temperature state of the turbine flow part in steam-free and motor mode are: geometric characteristics of the turbine stages; steam consumption for turbine seals; thermal scheme of the steam turbine in the motor mode; the steam consumption and the parameters of the cooling steam flows; steam pressure in the condenser; initial temperature condition of the turbine stages.

The algorithm of calculation in the unknown costs of steam through turbine stage following [9, 10]:

1. We set the steam temperatures on the flow part of the turbine, based on the conditions of the regime nature or other considerations (maintaining a certain temperature state of the turbine stages before starting from different thermal conditions, etc.).
2. We set the steam consumption for sealing and for cooling of turbine cylinders; based on the calculated steam consumption steam pressure into the compartments, taking into account the temperature correction. The reverse approach is also possible-the steam pressure in the compartments and in the stages is set on the basis of the operating characteristics or for technological reasons and the required steam consumption values are determined.
3. On the basis of the obtained pressure distribution, we calculate their ratio by steps and the critical value of these relations.
4. On the basis of the obtained relations, we determine the mode of operation of the stage by the method given in [1] and choose a mathematical model corresponding to this stage.
5. Using the mathematical model, we calculate the temperature of the steam throughout the flow part of the turbine, while the steam parameters at each stage are equal to their input parameters to the next stage. Special attention is paid to the calculation of friction and ventilation losses, taking into account the degree of partial character of the considered stage.
6. According to the obtained distribution of temperatures, the values of steam pressures in stages are specified taking into account the temperature correction.
7. After the end of these calculations, the temperature condition of the turbine is compared with the accepted state. If necessary, set new costs or new temperature distributions and the entire calculation is repeated.

The adequacy of the model was checked by comparing the results of model calculations carried out for steam turbines K-200 and T-100 with the experimental data given in [1, 7]. The model was considered adequate if the results of the calculations fall into the range of experiments (fall into the confidence corridor). It is obvious that the adequacy of the model depends to a greater extent on what dependencies are used in the calculation of power losses for friction and ventilation. According to the results of the studies, it is recommended to use the above dependencies (5), (7) and (8) for their calculation.

An analysis of the results of computational studies carried out according to the developed algorithm for steam turbines of various capacities made it possible to distinguish certain regularities characteristic of the motor regime of most steam turbines, including:

- a unique dependence of the temperature of the turbine blades on the flow and parameters of the cooling steam at a given pressure in the condenser, which makes it possible to control the temperature of the flow part of the turbine in order to create the necessary conditions for rapid loading of the turbine;
- for the duration of the temperature stabilization in the flow part of the turbines after the supply of cooling steam – for all types of turbines is 15-20 minutes, after which the turbine can operate in the motor mode for a long time;
- at the optimum value of the pressure in the condenser – which makes 0.004-0.005 MPa (at higher pressure, friction and ventilation losses increase, which leads to increase in the generator power, and at lower pressure, the air suction in the condenser and the power consumption of the circulating pump increase);
- according to the optimum configuration for supplying cooling steam to the turbine – for single-cylinder turbines to the head of the turbine, for two-cylinders with intermediate superheating of steam – to IPC and to the input of LPC, etc.

IV. CONCLUSION

1. The review of the application of the motor mode at the TPP showed the relevance of continuing research aimed at modeling the operation of steam turbine stages in low steam modes;
2. Taking into account the specifics of the steam turbine stages in low-flow modes, a model and an algorithm based on the calculation of the steam temperature in the stage and the group of stages are compiled.
3. It is shown that the adequacy of the model largely depends on the dependences used in the calculation of power losses for friction and ventilation and the results

of the studies recommended dependences that provide the necessary model studies for the accuracy of calculations.

REFERENCES

[1] Arakelyan E.K., Starshinov V.A. Increase of efficiency and maneuverability of the equipment of thermal power plants. Moscow. Publ. MEI. 1993, 328 p. (in Russian)

[2] Radin Y.A. Research and improve maneuverability combined-cycle plants. Author's abstract on competition of a scientific degree academic article doctor. Techn. sciences'. Moscow. 2013, 42 p. (in Russian)

[3] Bauman K. Der Anvarmvorgung in Dampfturbinen VIEN. Frankfurt am Nein. 1962.

[4] Bjoorgard A.B. Economic problems in connection with production reactiw power. CIGRE, Report and Power. 1962. No. 109.

[5] Powell E.B. Sunehernenos, Condensos operation of turbogenerators CIGRE, Report and Power. 1960. No. 16.

[6] Machovka Y.M. Research steamless mode HPC powerful steam turbines with the purpose of increasing maneuverability in terms of two-bypass start-up and reset system of the power unit. Astract of Dis. PhD Techn. Sciences. L., LMZ. 1978, 20 p. (in Russian)

[7] Madoyan A A., Levchenko B.L., Arakelyan E.K. and oth. Application of the motor mode at thermal power plants. Moscow. Publ. Energy. 1980, 256 p. (in Russian)

[8] Kulichikhin V.V., Gutorov V.F. Use of a motor mode of the turbine units at thermal power plants. Moscow. Publ. SPO ORGRES. 1977. (in Russian)

[9] Madoyan A.A., Kobzarenko L.N. About expediency of the motor mode on turbine units of the NPP. Thermal engineering. 1986, No. 3. (in Russian)

[10] Shapiro G.A. Improving the efficiency of CHP. Moscow. Publ. Energoizdat. 1981. (in Russian)

[11] Arakelyan E.K., Andryushin A.V., Andryushin K.A. Increased reliability, manoeuvrability and durability of steam turbines through the implementation of the generator driving mode. WIT Transactions on Ecology and the Environment. WIT Press. 2016.Vol. 205, p. 95-105.

[12] Neuimin V.M. Improving the efficiency of CHP by optimizing the low-potential part of the heat turbines. Abstract PhD Techn. Sciences. Minsk, 1995, 20 p. (in Russian)

[13] Samoylovich G.S., Troyanovsky B.M. Variables and transition modes in steam turbines.Moscow. Publ. Energoizdat. 1982. (in Russian)

[14] Arakelyan E.K., Andryushin A.V., Burtsev S.Y., Andryushin K.A. Technical and economic expediency of conversion of steam turbine PGU-450 into motor mode. Power stations. 2017, no. 6, pp. 25-29. (in Russian)