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ASSESSMENT OF THE EFFECTS OF THE OPERATION OF POWER UNITS ON SLIDING-PRESSURE

In the article the results of an analysis of the performance of a unit operated with incomplete sliding-pressure and with full sliding-pressure adjusted each time to a specific load are shown. The article also presents the gain obtained from the use of full sliding-pressure resulting from the reduction of thermodynamic loss in the system and the reduction of the unit's own needs. The measurements of the condensation-heating turbo set operated with incomplete sliding-pressure was used in the analysis. The method of choosing the reference steam pressure before the turbine was worked out to ensure minimal loss.

Keywords: *steam turbines, adjustment, sliding-pressure.*

1. Introduction

The control of power units designed to operate with sliding-pressure should for partial loads keep up the pressure after the boiler at slightly higher values (to make up for the loss in the piping, the control valve fully open) than the pressure before the first stage of the turbine (made for this type of control without regulation stage). The pressure after the boiler for partial loads results from the equation of the turbine flow capacity and the hydraulic loss in the piping. For operational reasons, i.e. to provide a possibility of quick changes of the unit power, a combination of sliding-pressure control with throttle control is used to retain higher pressure after the boiler than before the flow system of the turbine. Such unit operation results in a reduction of the cycle efficiency due to increased loss of the throttling at the control valve. Additional loss results from increased operation of the feed pump.

The analysis used an algorithm of calculations for the heating system of a unit formulated on the grounds of the equations of performance of the elements determined for this system, the equations of the turbine flow capacity and efficiency characteristics of the turbine, and the characteristics of heat exchangers. In order to determine these characteristics operational measurements of the unit were used.

To analyse the loss resulting from the unit operation with incomplete sliding-pressure an algorithm and a computer programme for balance calculations of the heating system were worked out to make it possible to determine the working medium parameters at individual cycle points for any value of the electric and heating load, and for different from the nominal values of: live steam pressure and temperature, the temperature of reheated steam, the temperature of feed water, the pressure in the condenser.

2. Turbine sliding-pressure control

Up till now basically two methods of control have been used to change the power of the turbine: throttle control and group (filling) control. In the former steam flows through one or two valves which open simultaneously. In the latter there are several valves which open successively. In both cases steam pressure in the boiler is steady and kept so with the pressure control.

In block systems sliding-pressure control also can be used by means of reducing appropriately the pressure after the boiler without the throttle valve intervention. The difference between a unit operating in the steady-pressure and sliding-pressure systems results from the scheme shown in Fig. 1 [1].

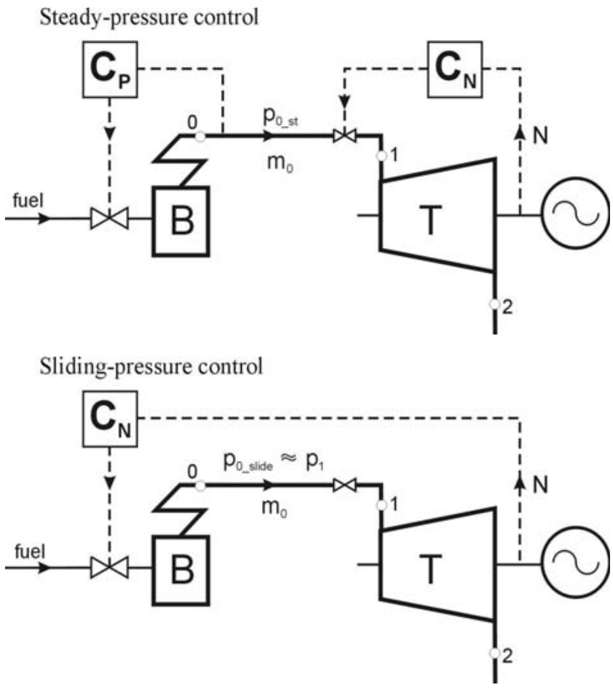

Fig. 1. The principle of operation of steady-pressure and sliding-pressure control of the boiler-turbine unit

In steady-pressure control the steam unit has two controls: the power of the turbine is adjusted with control C_N, which has an effect on the turbine valves and changes the steam jet according to demand. Control C_P maintains steady pressure after the boiler p_{o_st} (before the turbine valves) adjusting the steam jet generated in the boiler to the current turbine demand for steam. Control C_P acts on the steam jet delivered to the boiler.

The steam pressure before the blade system of turbine p_1 changes according to the principle of flow capacity. In sliding-pressure control the unit has only one control – power control C_N, with no pressure control. In consequence, the steam pressure after the boiler p_{0_slide} is the same as the pressure before the blade system (except for the loss caused by the hampering in the piping). Power control C_N measures the power of the turbine and acts directly on the stream of fuel adjusting the amount of steam to demand. The turbine consumes as much steam as is currently being produced by the boiler.

The pressure of the steam in the boiler varies in proportion with the steam efficiency of the boiler. At the same time the temperature control maintains a steady temperature of live steam.

Fig. 2 shows the variation of the parameters before the turbine with throttle and sliding-pressue controls.

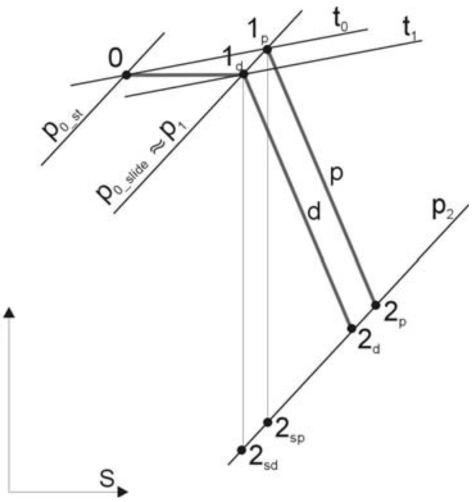

Fig. 2. Expansion in the turbine with sliding-pressure (p) and throttle (d) control

As the pressure of live steam diminishes, the efficiency of the cycle comes down as well. However, the efficiency fall is smaller than with throttle control because there is no loss of pressure in the turbine valves (p_{o_st}-p_1), and, resulting from it, the decrease of the steam temperature before the first stage of the turbine (T_0-T_1). Also, with throttle control the feed pump forces the condensate for the full nominal pressure of the boiler whereas with sliding-pressure control, in the conditions of partial load, the pressure in the boiler – and the force pressure of the pump – is lower, the power needed to drive the feed pump is lower (Fig. 3).

The main advantage of the sliding-pressure control in comparison with the throttle control is the labour saving of the feed pump under partial loads. The saving rises in proportion to the nominal pressure of live steam.

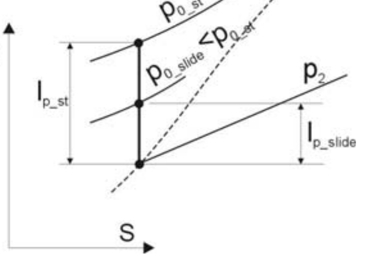

Fig. 3 Operation of the feed pump under partial load with steady-pressure (l_{p_st}) and sliding-pressure (l_{p_slide}) control

3. Scope of research and calculations

The aim of the research is to assess the effects of the change of the turbine control from incomplete sliding-pressure control to full sliding-pressure control. For the analysis a condensation-heating unit of the electric power of 145MW was assumed. The measurement scheme of the turbine is shown in Fig. 4. For the calculations of the turbine heating system the data from guarantee measurements were used. The measurements were within the full range of the turbine power (55-155MW) [2].

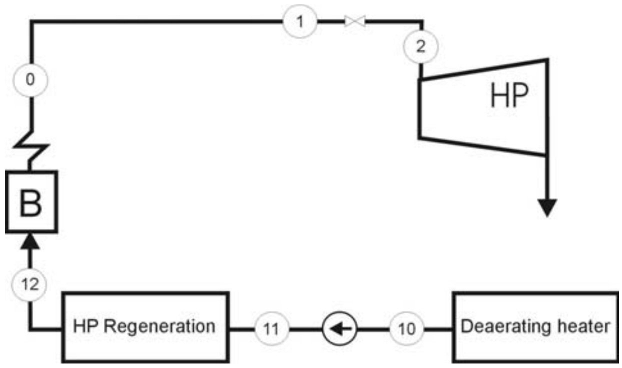

Fig. 4. Simplified scheme of the turbine heating system

4. Determination of nominal sliding-pressure

The condensation-heating turbine operates usually, under partial loads, with partly closed control valves of the HP part. This leads to the throttling of the steam jet and losses involved with it. These losses can be divided into thermodynamic and those caused by the excess power of the feed water pumps. The losses can be avoided by switching to sliding-pressure operation only. In this case the turbine control valves should be fully open. The pressure adjustment is then made with the boiler. The value of this particular pressure results from the current load of the turbo set. A method to determine it is shown below [3].

The equation of flow capacity for the HP part determines the pressure at the steam admission into the flow system (1) – i.e. before the first stage of the turbine.

$$\frac{m_2}{m_{02}} = \sqrt{\frac{T_{02}}{T_2}}\sqrt{\frac{p_2^2 - p_{10}^2}{p_{02}^2 - p_{010}^2}} \Rightarrow p_2 = f\left(m_2, p_{10}, T_2\right) \tag{1}$$

where: m_2 – steam jet passing through the HP part of the turbine, m_{02} – steam jet passing through the HP part of the turbine for reference conditions, T_2 – temperature before the HP part of the turbine, T_{02} – temperature before the HP part of the turbine for reference conditions, p_2, p_{10} – pressure before and after the HP part of the engine (p_{10} determined by measurement), p_{02}, p_{010} – pressure before and after the HP part of the engine for reference conditions.

1. Allowing for the loss of pressure in the piping from the boiler and losses in the isolation and control valves, the pressure after the boiler is:

$$\zeta_{kot-2} = \frac{p_0 - p_2}{p_0} \tag{2}$$

where: p_0 – pressure after the boiler, p_2 – pressure before the turbine

For the analysis the value of the loss (2) determined by the measurement of the turbine for full power (control valves fully open) was assumed. It amounts to:

$$\zeta_{kot\_2} = 0.037$$

2. The loss of pressure in the boiler (3) is determined by the measurements of the unit.

$$\zeta_{kot} = \frac{p_{wz} - p_0}{p_{wz}} \qquad (3)$$

where: p_{wz} – pressure of feed water before the boiler (Point 65 in the heating scheme)

The characteristics of the resistance in the boiler in the function of the steam jet are shown in Fig. 5, and a straight line was approximated with the following equation (4):

$$\zeta_{kot} = 0.000808645 * m_0 + 0.00902612 \qquad (4)$$

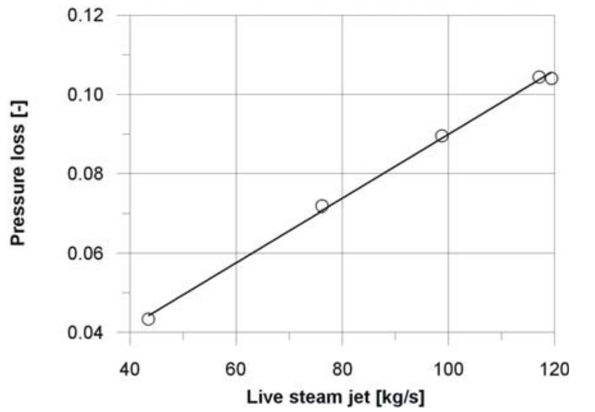

Fig. 5. Relative pressure loss in the boiler in the function of live steam jet

3. Determination of the feed water pump pressure on the grounds of the losses derived from formula (5):

$$\zeta_{pom\_kot} = \frac{p_{pom} - p_{wz}}{p_{pom}} \qquad (5)$$

where: p_{pom} – pressure at feed water forcing

The average value of the loss determined by the measurements for various loads amounts to:

$$\zeta_{pom\_kot} = 0.047$$

5. Thermodynamic losses in cycle

The pressure of live steam after the boiler for various conditions of the turbine load was maintained in accordance with curve (o) shown in Fig. 6. In order to adjust the steam pressure before the first stage of the HP part of the turbine, the steam jet had to be throttled in the control valves. The decrease of the pressure of live steam in the valves is described as the difference between the pressure before the valves (o) and the pressure before the turbine (x). The difference between the two is shown as the marked area between the course of the pressures in question. The loss is illustrated as the curve in the bottom part of the chart (Fig.7). The lower the power of the turbine, the bigger the throttle waste, which involves significant losses connected with the change of the thermodynamic parameters of the steam jet.

The throttling of the heating medium together with the decrease of the pressure cause a fall in its temperature. This change is shown in the chart as the area which makes up the difference between the course of the steam temperature before the valves and before the turbine. The difference is also shown as the curve in the bottom part of the chart (Fig. 7). The lower the power of the turbine, the bigger the loss caused by the decrease of the parameters of steam conducted to the turbine. In such cases the temperature of live steam differs considerably from nominal temperature (cf Fig. 7).

Adjusting steam pressure before the turbine to individual load conditions through the throttling process generates thermodynamic losses in the system. The analysis which was carried out makes it possible to present the losses in the shape of the function shown in Fig. 8. Operation under maximum load does not generate thermodynamic losses in the system, but the lower the load, the bigger the loss. For minimal loads of the turbine in question it amounts to 50 kJ/kWh. This constitutes about 0.55% of the heat consumption per unit of the turbo set (Fig. 9).

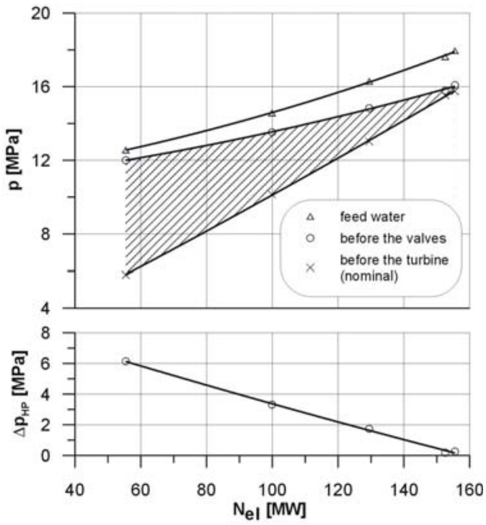

Fig.6. Pressure loss in the valves

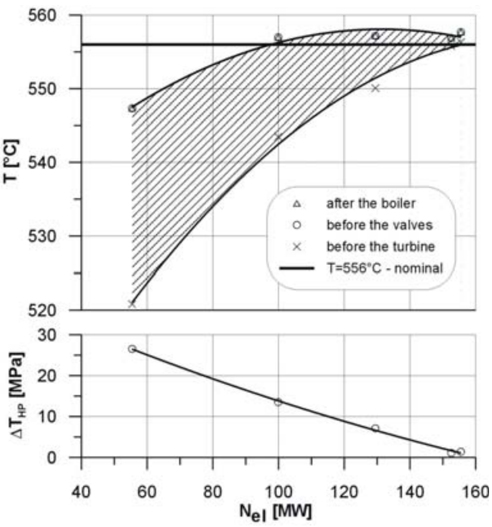

Fig. 7. Temperature loss in the valves

6. Assessment of the effects of the change in the type of control

The second stage of the analysis comprised the determination of the effect of the change in the type of control, from incomplete sliding-pressure to full sliding-pressure, on the power of the feed pump. The power of the pumps for the current operation conditions of the unit was shown in Fig. 10 as a curve marked in the chart as (Δ). After assuming full sliding-pressure (curve "x" in Fig. 6), the power needed to drive the pumps for partial loads of the unit will diminish. The value of this power for particular turbine loads is shown in Fig. 10 as curve (□). The difference between the power of the pump for current operation conditions and the power for full sliding-pressure is illustrated as the area marked in Fig. 10. The loss resulting from excessive power of the feed water pump for altered load conditions is shown in the bottom part of the chart.

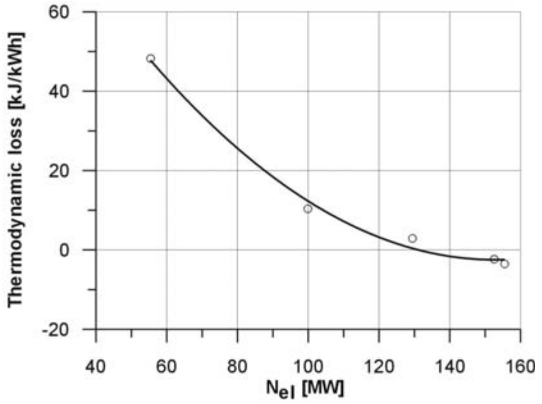

Fig. 8. Thermodynamic loss resulting from failure to keep the parameters of live steam

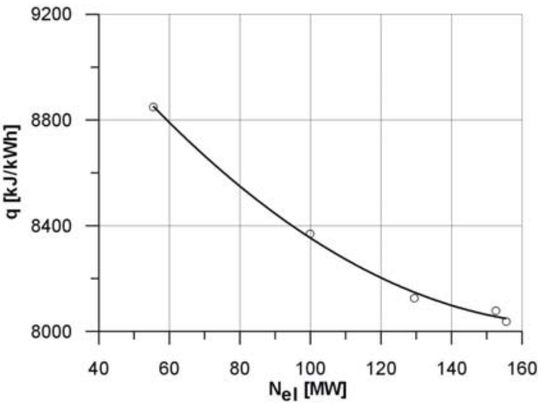

Fig. 9. Heat consumption per unit

7. Assessment of the reduction of own needs of the power plant after the change of the type of control

The assessment of the unit's own needs reduction which results from the decrease in the feed water pump labour was carried out for measurement data from a period of a two-week running of the turbo set. The data collection was done at one-minute intervals for the whole sampling period. In this way, the course of the turbine power shown in Fig. 11 was achieved. According to the course, the turbine operated usually in two scopes of power: the first comprised maximum powers (approx. 11800 min of operation – unit load in peak hours); the second – minimal powers (approx. 4000 min of operation – unit load during the night off-peak hours). The percentage of the turbine operation in individual scopes of power is shown in Fig. 12.

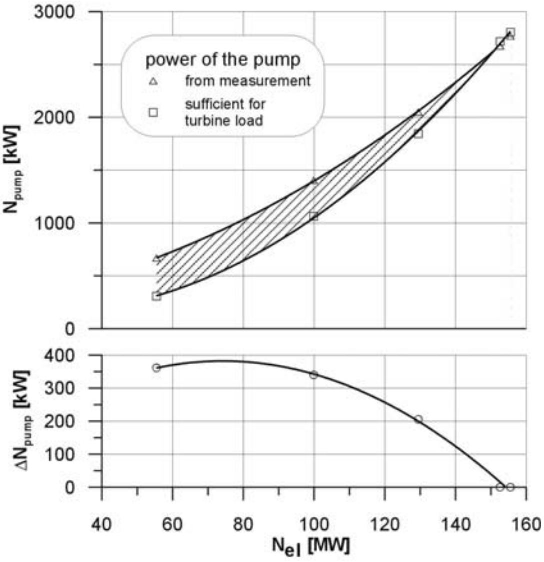

Fig. 10 Power change of the feed water pump

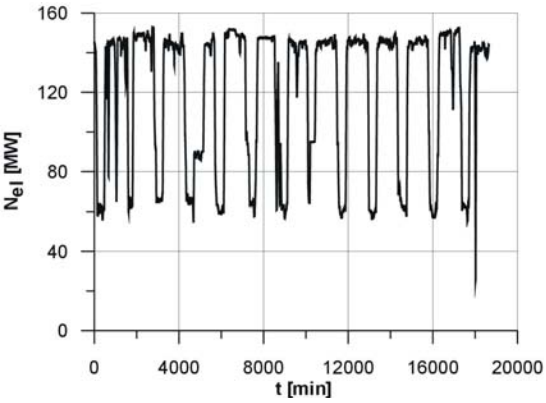

Fig. 11 Course of the turbine power

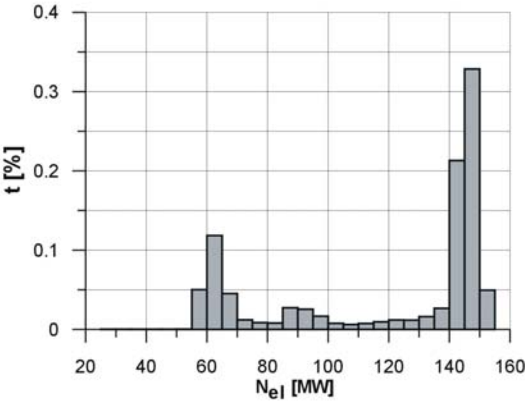

Fig. 12 Time of operation in particular scopes of power

For the first kind of operation (maximum power) the turbine does not generate any losses connected with type of control. However, with partial load the losses, as has been shown above, reach significant values (Fig. 8).

In the lower range of the turbo set load, with the operation of the turbine with full sliding-pressure, the estimated fall in the power of the feed water pumps is about 360 kWh. For the considered time of operation of t = 3985 min the gain amounts to about 24000 kWh. Assuming that the turbine operates like this for a whole year (allowing for maintenance works) the gain totals about 529000 kWh.

The longer the time when the turbine operates with loads smaller than nominal, the bigger the effect.

8. Conclusion

The operation of units running with sliding-pressure for partial loads is more advantageous in comparison with other types of control. This results from the fact that in such conditions the unit does not generate thermodynamic losses being a consequence of steam throttling in control valves, which in turn causes a fall in the temperature of steam before the first stage of the turbine. Also, the own needs of the power plant are smaller as the feed water pressure is lower.

In the case of turbine operation with incomplete sliding-pressure, where steam pressure before the turbine is partially adjusted to loads, a substantial gain can be achieved due to a change of the type of control in the shape of a decrease in the heat consumption per unit and a reduction of the own needs of a power unit.

9. References

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