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Comparative Analysis between Single - and Two- Stage District Water Heating by Backpressure Steam Turbine

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Abstract

In the article is estimated the profit of the transition from single to two-stage district water heating by back pressure steam turbine. It is obtained theoretical expressions for extra produced and consumed power in a two-stage heating system compared to one single-stage. An innovation in the expressions is presentation of extra powers as ratio to the turbine heat load. By using the expressions is obtained results for these powers under certain assumptions. The results of extra produced and consumed powers from the formulas are compared with results from models of a concrete steam turbine installation.

Keywords: backpressure steam turbine; single-stage district water heating; two-stage district water heating.

1. Introduction

The backpressure steam turbines (BPST) for power generation and district heating (CHPDH) are characterized by high energy efficiency [1, 2]. The only way to increase their effectiveness is changing from one stage to multi-stage district water heating. In case of multi-stage district water heating the efficiency is increased due to additional production of electricity. At the same time increases the consumption of thermal energy from the steam turbine installation, and the efficiency decreases. The end effect is obtained as a resultant of these two effects.

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On the end effect do influence some factors, the most significant of which are the temperature regime of district heating system (DHS) - supply and returned temperature, and the heat load of the turbine. The research to increase the efficiency of multi-stage district water heating are focused on condensing steam turbines with controlled steam extraction because, they have a lower overall energy efficiency beside the backpressure turbines on the one hand and on the other - a lack of connectivity between thermal and electrical load that connectivity is a major disadvantage of backpressure steam turbines [3]. Therefore the aim of this study is to compare the effectiveness of BPST in case of single-stage and two-stage district water heating.

The two-stage district water heating is the optimum in terms of the relation between increased efficiency and investment. As the heating of the district water is usually 30-50 °C, for such range is suitably a two-stage district water heating. The implementation of the three-stage heating is unjustified because increased investments much more in comparison with two-stage heating whereas the efficiency is increased about 2-3%.

2. Single- and two-stage heating of district water

Figure 1 and figure 2 show schematic diagrams respectively of single- and two-stage district water heating by backpressure steam turbine. It is assumed that the pressure in each heater is equal to that in the steam extraction, i.e. neglected losses of pressure in steam lines between the turbine and heaters which losses normally are 3 %. The reason for this assuming is as follows. In the range of low pressures 0.3-1.0 bar if the difference between two pressures is 3 %, the difference between theirs saturation temperatures is less than 0.8 ° C.

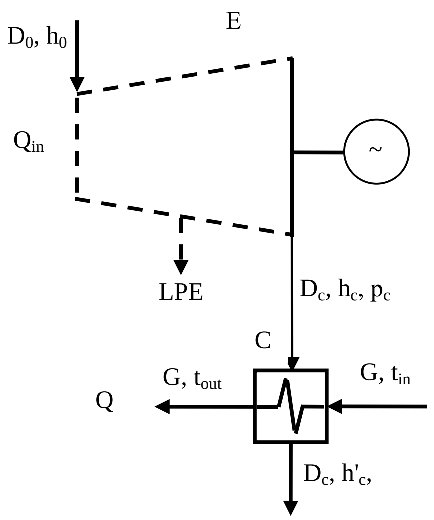

Figure 1: Single-stage district water heating

Basic condition for the comparison of the two schemes is to give the same heat output Q at the same district water temperature regime. In the single-stage scheme that is the power of the DH-condenser (C):

$$Q = c.G(t_{out} - t_{in}) \tag{1}$$

and in the two-stage scheme is the sum of the thermal power of both DH-condensers (C1 and C2).

$$Q = c.G(t_1 - t_{in}) + c.G(t_{out} - t_1) = c.G(t_{out} - t_{in}) \quad (2)$$

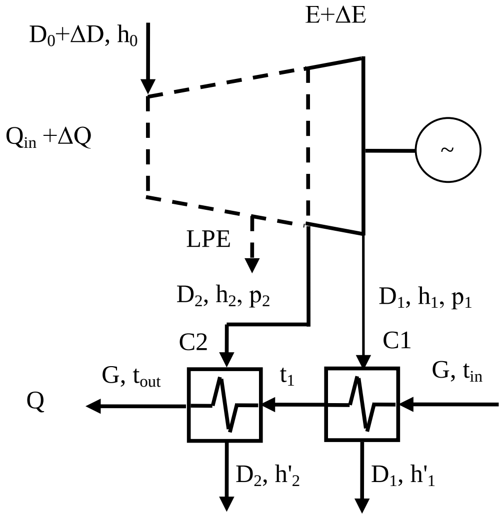

Figure 2: Two-stage district water heating

where:

- c - specific heat capacity of water;
- G - mass flow rate of the district heating water;
- t_{out} and t_{in} are the temperatures respectively of forward and return water;
- t_1 is the water temperature after first stage DH-condenser C1 (fig. 2).

In the two-stage scheme to heat the water after the DH-condenser C2 to the same temperature as in single-stage scheme is necessary the parameters in steam extraction, powered DH-condenser C2, be the same as those in backpressure of DH-condenser C, i.e. $p_2 = p_c$ and $h_2 = h_c$. Of course, an additional condition is the same terminal temperature difference (TTD) in C and C2.

The steam flow rate D_1 in its expansion between extraction to C2 and the turbine output will produce additional electric power ΔE'. The condensate from this stream flow (after C1) has a lower temperature and enthalpy then the condensate after DH-condenser C2 which parameters correspond to the parameters of the condensate after C at single-stage scheme. The raise of the parameter values (temperature and enthalpy) of the condensate outgoing from C1 to the same values of outgoing from C2 condensate will require an additional heat power ΔQ. This additional power is covered by increasing the mass flow rate of ingoing steam flow to the turbine with ΔD. The steam flow ΔD will produce through the turbine an additional electric power ΔE''.

Therefore, when switching from one-stage to two-stage scheme of district water heating is necessary to evaluate the following three effects:

- increasing the turbine electric power with ΔE' due to further steam expansion of mass flow D_1 from pressure p_2 to p_1;
- increasing the heat consumption of the turbine with ΔQ for extra heating of the condensate after C1 with mass flow D_1 to the parameters of the condensate after C2;
- increasing the turbine electric power with ΔE" as a result of increased steam flow through turbine with ΔD to cover the increased heat consumption of the turbine with ΔQ.

3. An estimation of the effects of a two-stage district water heating

The benefit in efficiency of two-stage district water heating depends on the interrelation between extra electric power generated (ΔE'+ ΔE") and extra heat power consumed (ΔQ).

The extra generated and consumed powers depend mainly on the steam parameters at the extractions supplied with steam the two DH-condensers, respectively on water temperature regime and the heat load allocation between the first and second stage.

The absolute value of the additional generated and consumed powers depends on the above factors and the steam turbine power. To do a comparison between different type of turbines it is estimated the relative value of the additional powers referred to an adopted base. For steam turbines with district heating controlled extraction the adopted base is the maximum possible additional generated power [4] and on this basis has been shown that the maximum effect by multi-stage district water heating is obtained with equal heat load of each stage [1, 5].

In the present study are evaluated the individual effects and the total effect of the two-stage district water heating on base heat load of backpressure turbine (Q).

3.1. Extra generated power ΔE'

The extra generated power ΔE' from the steam flow D_1 in two-stage scheme is:

$$\Delta E' = D_1(h_2 - h_1) \tag{3}$$

where:

- D_1 - outgoing steam mass flow from the turbine;
- h_1 - enthalpy of turbine outgoing steam flow;
- h_2 - enthalpy of steam in the extraction, powered second stage heater.

If multiply the right side of expression (3) by electro-mechanical efficiency of the turbo-generator η_{TG} can obtain the extra generated electric power on generator terminals:

$$\Delta E'_e = D_1(h_2 - h_1).\eta_{TG} \tag{3e}$$

The heat balance of heater C1 without giving an account its heat losses is:

$$D_1(h_1 - h_1') = c.G.(t_1 - t_{in}) \tag{4}$$

from where can determinate the steam mass flow to the first stage heater:

$$D_1 = \frac{c.G.(t_1 - t_{in})}{h_1 - h_1'} \tag{5}$$

where h'$_1$ is the enthalpy of the condensate in first stage heater.

After substitution of (5) into (3):

$$\Delta \mathrm{E}' = c.G.(t_1 - t_{in})\frac{(h_2 - h_1)}{(h_1 - h_1')} \tag{6}$$

Dividing expression (6) of the heat load (Q) - expression (1) is obtained:

$$\frac{\Delta \mathrm{E}'}{Q} = \frac{t_1 - t_{in}}{t_{out} - t_{in}} \cdot \frac{h_2 - h_1}{h_1 - h_1'} \tag{7}$$

The first multiplier on the right side of the last expression represents the relative heat power of the first heating stage respectively it represents the heat load allocation between the two stages.

Fig. 3 shows a graphic form of expression (7) using the typical for Bulgaria district water temperature regime 90/60 (t_{out}/t_{in}) °C and argument the water temperature after the first heating stage - t_1. It was obtained under the condition that incoming into the C1 and C2 steam is in saturation state (x=1) and TTD is 3 °C in each DH-condenser.

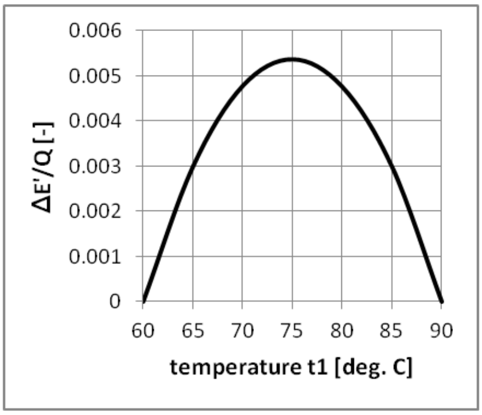

Figure 3: Graphic form of expression (7) if the steam is dry saturated (x=1)

The maximum of the ratio ΔE'/Q is at 75 °C, which corresponds to the same with 15 °C heating in each stage and confirms the known fact that the maximum effect of stepwise water heating is obtained with equal heat load for each of the stages [3]. Fig. 3 shows also that the maximum additional generated power ΔE' is 5.3 kW per 1 MW heat power to the district heating system.

In fact, the incoming to the C1 and C2 steam is not in dry saturation state [3]. The incoming to C1 steam has quality x = 0.95 to 0.99 (wet steam) and this to C2 may be wet as well slightly superheated, i.e. the enthalpy $h_1<h_1''$ and the enthalpy $h_2<h_2''$ $h_2>h_2''$. As a result, the second multiplier in the expression (7) has about 5 times greater value compared to its value in case of dry saturated steam (x=1), $h_1= h_1''$ and $h_2=h_2''$. The graphic form of expression (7) if the steam to C1 has quality x = 0.97, and that to C2 has quality x = 1 is shown on fig. 4. In this case and equal allocation of the heat load between C1 and C2 the additional generated power ΔE' is 21 kW per 1 MW heat power to the district heating system.

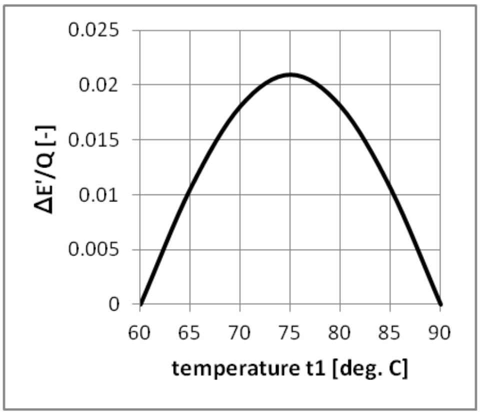

Figure 4: Graphic form of expression (7) if the steam is in real conditions

3.2. Extra consumed from the turbine power ΔQ

The heating of the condensate outgoing from heater C1 to the parameters of outgoing from heater C2 condensate will consume an additional heat power:

$$\Delta Q = D_1(h_2' - h_1') \tag{8}$$

After replacing D_1 from expression (5) in expression (8) for additional consumed power is obtained:

$$\Delta Q = \frac{c.G.(t_1 - t_{in})}{h_1 - h_1'}(h_2' - h_1') \tag{9}$$

where h'_2 is the enthalpy of condensate in second stage heater C2.

By division of expression (9) to expression (1) it is obtained:

$$\frac{\Delta Q}{Q} = \frac{t_1 - t_{in}}{t_{out} - t_{in}} \cdot \frac{h_2' - h_1'}{h_1 - h_1'} \tag{10}$$

The relative extra consumed power - expression (10) also depends on the load allocation between the first and second heating stage - the first multiplier on the right side of expression (10) and the corresponding of this allocation parameters of heaters C1 and C2.

3.3. Extra generated power $\Delta E''$

Covering the extra consumed power ΔQ requires an increasing of steam flow through turbine with ΔD. The value of ΔD depends on the turbine and auxiliary design, in particular where is going the condensate with flow rate D_1 at the heat diagram of the turbine. As the additional steam flow ΔD will produce extra power through the turbine, the theoretical maximum value of this power will occur if the condensate after C1 is heated to the parameters of condensate after C2 with steam from turbine extraction powered with steam the heater C2.

Then the increase of steam flow through the turbine is:

$$\Delta D = D_1 . \frac{h_2' - h_1'}{h_2 - h_2'} \tag{11}$$

In practice, heating of the condensate after C1 is carried out in the heater fed with steam from extraction upstream then extraction for heater C2. Assuming this extraction with conditional number 3, the additional steam flow through the turbine will be:

$$\Delta D = D_1 \frac{h_2' - h_1'}{\eta (h_3 - h_3')} \tag{12}$$

where:

- h_3 is the enthalpy of steam powered the heater;
- h'_3 is the enthalpy of condensate in the heater;
- η is coefficient taking into account the heat loss of the heater.

The extra generated power from steam flow ΔD through the turbine is:

$$\Delta \mathrm{E}'' = \Delta D . (h_0 - h_3) \tag{13}$$

where h_0 is the enthalpy of live steam to the turbine input.

If multiply the right side of expression (13) by electro-mechanical efficiency of the turbo-generator η_{TG} can obtain the extra generated electric power on generator terminals:

$$\Delta E_e'' = \Delta D . (h_0 - h_3) . \eta_{TG} .$$
(13e)

Maximum extra power will be generate if the additional steam flow is determined by expression (11). After substitution of (11) to (13) and dividing the result to the heat power Q from expression (1) is obtained:

$$\frac{\Delta E''}{Q} = \frac{t_1 - t_{in}}{t_{out} - t_{in}} \cdot \frac{h_2' - h_1'}{h_1 - h_1'} \cdot \frac{h_0 - h_2}{h_2 - h_2'} \tag{14}$$

Again, the first multiplier on the right side of the last expression represents the relative heat load of the first stage, the second one is connected with load allocation between the two stages and the third take into account the impact of the turbine input parameters.

4. Results

Figure 5 shows in graphic form expressions (7), (10) and (14), the sum of (7) and (14) - ΔE/Q and the profit ΔE/Q-ΔQ/Q by water temperature regime 90/60 (t_{out}/t_{in}) °C depending on the water temperature between first and second stage (t_1), characterized the load allocation Q between them.

The extremes of these functions is at t_1=75 °C, i.e. in the same way with 15 °C heating the water in each stage. The maximum values are:

- maximum extra generated power ΔE'/Q (as indicated above) is 0.021, i.e., 21 kW per 1 MW of heat power to DHS;
- maximum extra generated power ΔE"/Q is 0,005, i.e. 5 kW per 1 MW of heat power to DHS;
- maximum extra consumed power ΔQ/Q is 0.014, i.e. 14 kW per 1 MW of heat power to DHS.

Therefore, in the two-stage district water heating it will generated extra power 26 kW per 1 MW heat power, but will consumed 14 kW per 1 MW heat power. The end result - profit from two-stage compared single-stage district water heating is 12 kW per 1 MW of heat power to DHS.

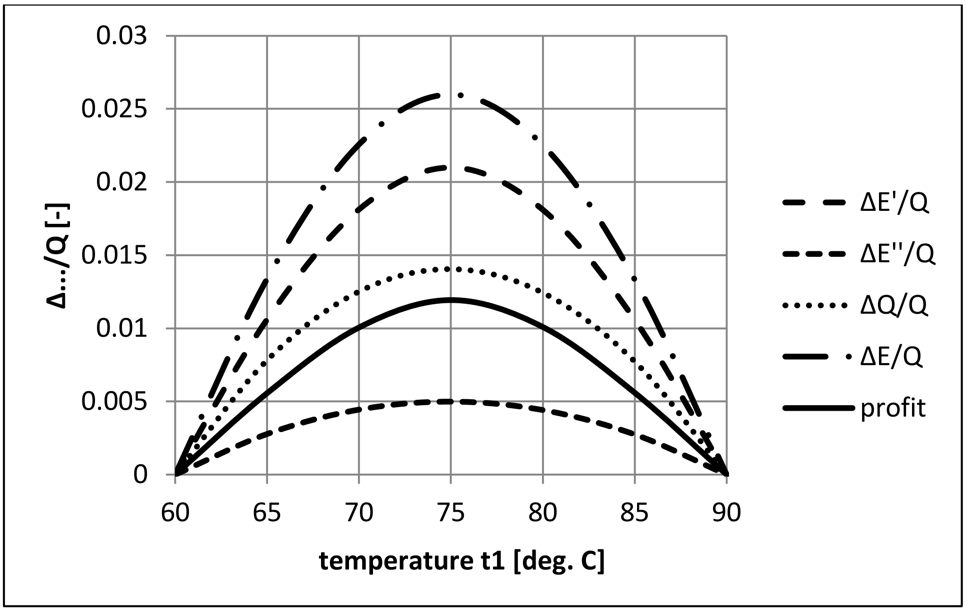

Figure 5: Relations of ΔE'/Q, ΔE"/Q, ΔQ/Q, ΔE/Q and profit from heat load allocation between the two stages in case of water temperature regime 90/60 °C

5. Results verification

The obtained results in an analytical way for efficiency of two-stage district water heating by backpressure turbine were compared with results from GateCycle models of steam turbine plant with a steam turbine type PR-35-8.8/1.0. The turbine is made in Russia and is used in DHS of Sofia to cover the base heat load. It is backpressure turbine with nominal electric capacity 35 MW and 80 MW heat power to DHS. The live steam parameters are: pressure 90 bar, temperature 535 °C. The pressure at controlled extraction (powered the deaerator) is 8-13 bar, the backpressure (at end of turbine) can change from 0.33 to 0.98 bar depending on returned water temperature (t_{in}). The steam turbine plant layout is DHC+LPE+D+2*HPE ("DH-condenser"+"low pressure exchanger"+"deaerator"+"two high pressure exchangers"). The DH-condenser is horizontal type PSG-1800 with 1800 m^2 heat transfer area.

They were created two models of steam turbine installation using GateCycle software [6]. In the first model, shown on fig. 6, the district water heating is by single-stage scheme. This model has been validated with the results of warranty tests by acceptance in operation of the turbine. The second model, shown on fig. 7, was created using the first model to which is added a second stage water heater powered with steam from uncontrolled extraction between turbine stages 15 and 16. For even distribution of heat load between the two stages the two heaters are identical with heat exchange surface 1000 m^2 of each.

The heat load in both models is identical Q=80 MW, which by 90/60 °C water temperature regime is exported with water flow rate 2290 t/h (636 kg/s).

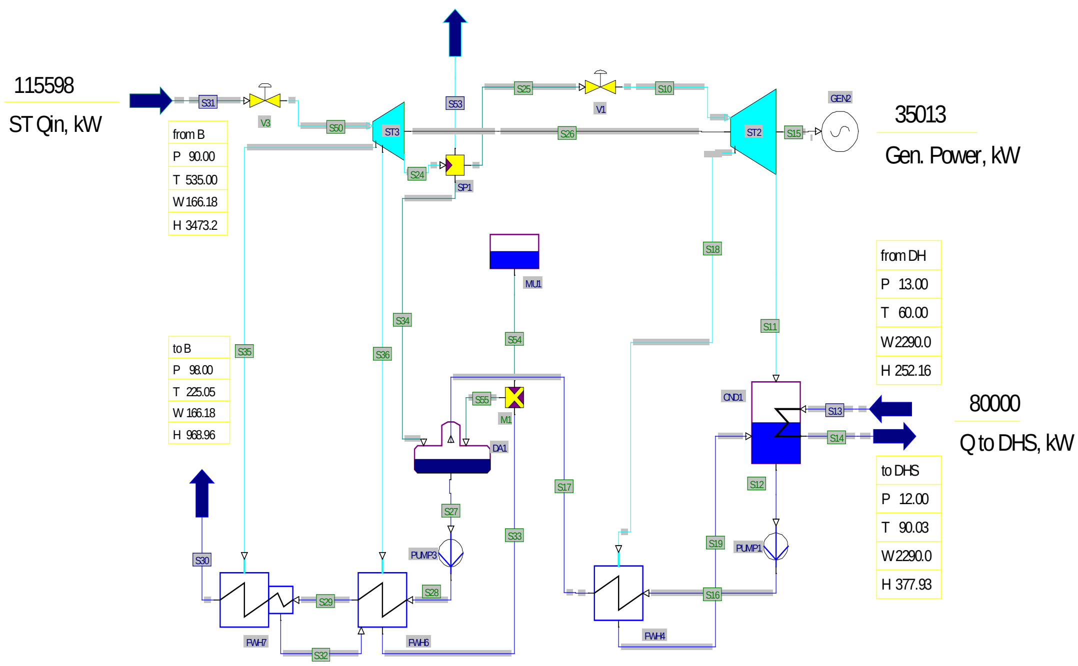

Figure 6: GateCycle model of steam turbine PR-35-8.8/1.0 in case of single-stage district water heating

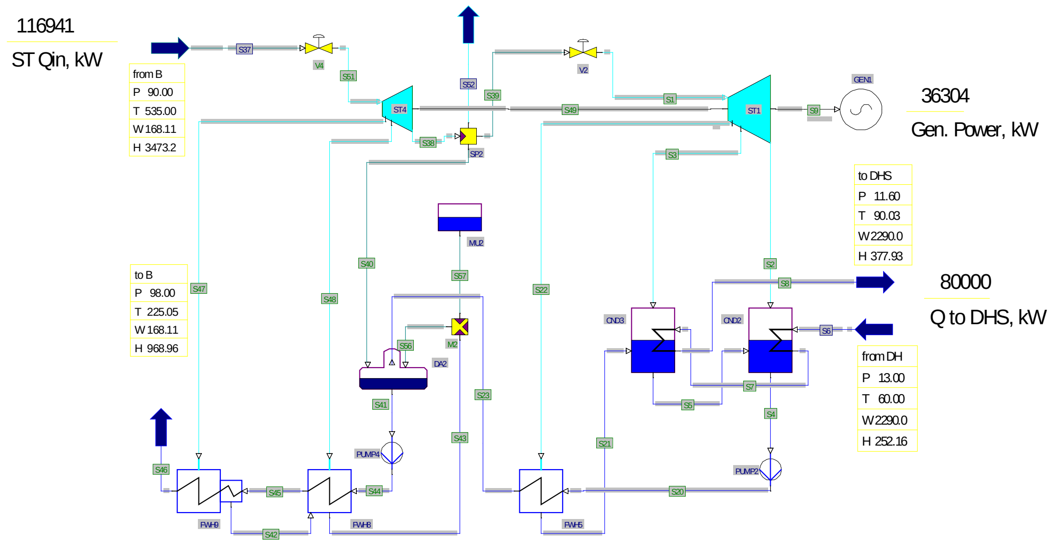

Figure 7: GateCycle model of steam turbine PR-35-8.8/1.0 in case of two-stage district water heating

Table 1 shows the results of simulation models and results expected for the two-stage water heating, according to the received expressions.

Table 1: Parameter comparison

Parameter	From model with single-stage heating	From model with two-stage heating	Expected analytical result	Analytical result deviation, %
Heat load Q, kW	80 000	80 000	-	-
Electric power on generator terminals, kW	35 013	36 304	37 030	2
Consumed heat power from the turbine Qin, kW	115 598	116 941	116 718	-0.19

According to expressions (6) and (14) for extra generated power from the turbine and considering the electromechanical efficiency coefficient of turbo-generator, the electrical power on the generator terminals has to be 37.030 MW. The same power, obtained from the model with two-stage heating is lower - 36.304 MW.

According to expression (10) for increased consumption of heat power the expected consumed power has to be 116.718 MW. The same power, obtained from the model with two-stage heating is greater - 116.941 MW.

The smaller electric generated power and the greater consumed heat power by the turbine is due to the following differences. In the real steam turbine plant respectively in its model the condensate from the two heaters is heated in low pressure exchanger powered from turbine extraction with absolute pressure 3.2 bar and temperature 167 °C, until the analytical results were obtained for the case of maximum efficiency, i.e. heating the condensate from turbine extraction powered second stage heater. This is the reason for less generated electric power in the real turbine installation. The greater consumed heat power in the real turbine installation is due to heat losses of the heaters, which losses are taken into account in the model. For simplicity of obtained expressions the heater losses are not taken into account.

6. Conclusion

They are obtained the expressions for extra generated and consumed power in a two-stage district water heating by backpressure steam turbine. As an innovation has be noted that these extra powers are related to the heat load (Q) covered by backpressure turbine. It is proven a known fact that the maximum effect of the two-stage district water heating is obtained by an even distribution of heat load between the two stages.

In this case and with water temperature regime 90/60 °C the extra generated power by the turbine is 26 kW per 1 MW of heat power to DHS and extra consumed heat power from the turbine is 14 kW per 1 MW of heat power to DHS.

The two-stage heating of the water by backpressure steam turbine is more energy efficient than single-stage due to the positive balance between extra generated and consumed power. Moreover by the two-stage heating increases the ratio between the electricity and the heat produced from cogeneration back pressure turbine (so-called alpha value).

Calculated are extra produced and consumed from turbine power plant with turbine PR 35-8.8/1.0 in case of two-stage district water heating. The correctness of the results is confirmed by comparison with those of GateCycle-model of steam turbine installation.

The presented results refer to nominal design mode of the turbine and DH-condensers. The deviation of the returned water temperature from nominal design mode changes the turbine outlet pressure and upstream extraction pressure. As a result the heat load is redistributed between the two stages and the efficiency decreases.

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