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Humid Air Turbine as a Primary Link of a Coal-Fired Steam Power Plant

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Abstract

Outlet gases of the humid air turbine (having a temperature of about 125 ^oC and great content of steam) can be used for the preheating of feed water of the steam power plant fueled with coal. So the efficiency of the plant can be increased and its ecological indices can be improved. The attainable incremental efficiency of the humid air turbine and the increased efficiency of the combined plant has been determined for three variants of the repowering of an existing steam power plant. The variant presented in *Figure 4* is recommended for practical application.

Key words: gas turbine, humid air turbine (HAT), combined power plant.

1. Energy Effects of the Primary Gas Turbine Applied for Preheating the Feed Water

The outlet gases of the gas turbine can be used for preheating the feed water of a coal-fired steam power plant. So the regenerative bleeds of the steam turbine can be partially switched off and the power of the steam turbine can be increased. The energy effects of such a cooperation of the gas turbine with the steam power plant can be well characterized by the *incremental energy efficiency*, Szargut (1999), defined as the ratio of the increase of power of the plant to the consumption of gas fuel at a constant consumption of the solid fuel:

$$\eta_\Delta = \frac{\dot{W}_g + \Delta\dot{W}_s}{\dot{E}_{ch\,g}} = \eta_{E\,g} + \frac{\Delta\dot{W}_s}{\dot{E}_{ch\,g}} = \\ = \eta_{Eg} + \frac{\dot{W}_s + \Delta\dot{W}_s}{\dot{E}_{ch\,g}} - \beta\frac{\dot{W}_s}{\dot{E}_{ch\,s}} \qquad (1)$$

where:

$\dot{W}_g$ — electrical power of the gas turbine complex

$\dot{W}_s$, $\Delta\dot{W}_s$ — initial electrical power of the steam turbine and its increase after the introduction of the primary gas turbine,

$\dot{E}_{ch\,g}$, $\dot{E}_{ch\,s}$ — consumption of chemical energy of gas fuel and coal per time unit,

$\beta = \dot{E}_{ch\,s} / \dot{E}_{ch\,g}$ — ratio of the streams of chemical energy of coal and gaseous fuel,

$\eta_{E\,g}$ — energy efficiency of the independent gas turbine complex.

The application of the conventional gas turbine as a primary link of the steam power plant was analyzed by Szargut (1999). An important disadvantage of this system is a relatively great irreversibility of heat transfer within the recovery boiler of the gas turbine. The temperature of the gases flowing out from the gas turbine is as high as 500-600 ^oC, and the needed temperature of the preheated feed water is about 230 ^oC. Therefore the attained energy effects are not very high. The incremental energy efficiency of the gas turbine is not higher than the efficiency of the combined steam-and-gas turbine utilizing the outlet gases for the production of steam.

The outlet gases of a humid air turbine (HAT) have a temperature of about 120 ^oC and therefore the temperature difference between them and the preheated feed water can be small, although it would not be possible to switch off all

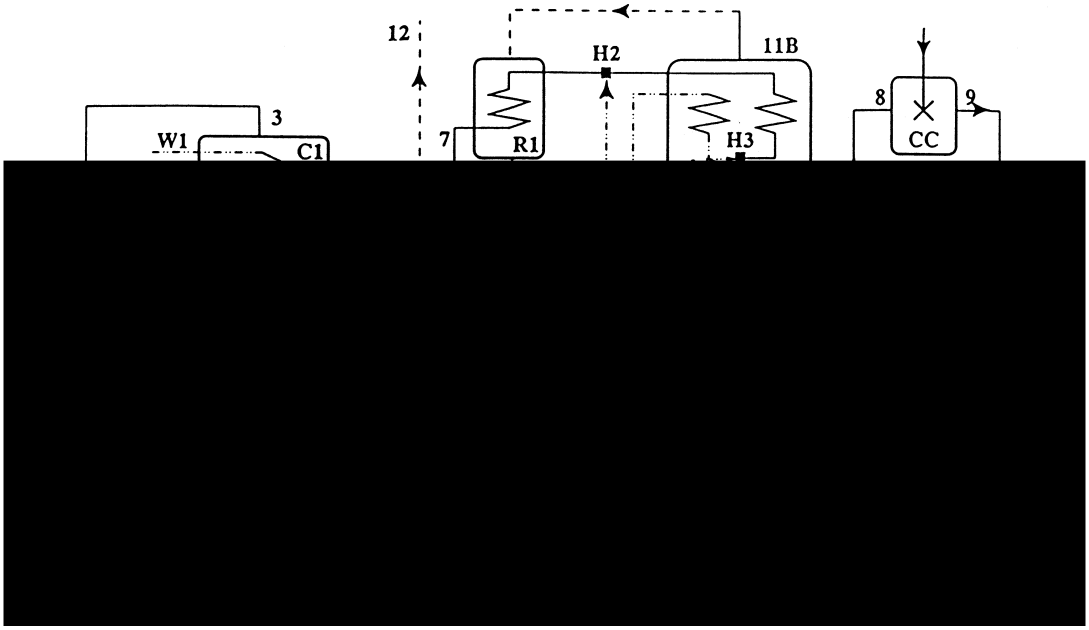

Figure 1. Scheme of the considered humid air turbine (HAT)
CL, CH — *low pressure and high pressure part of the compressor*; CC — *combustion chamber*; T1...T4 — *stages of the turbine*; C1, C2 — *interstage and external cooler*; H1...H3 — *humidifiers*; R, R1 — *parts of the regenerative air preheater*; D1,D2 — *distributors of cooling air*; 1..12 — *states of the working fluid*, W1...W3 - *states of water.*

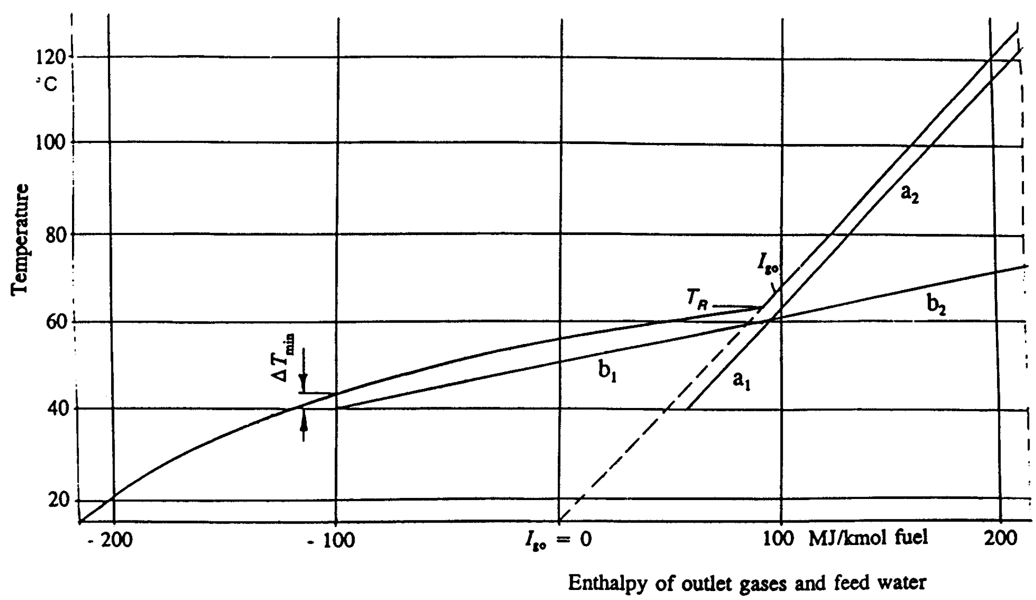

Figure 2. Exemplary enthalpy-temperature diagram of the outlet gases and feed water
I_{go} — *enthalpy of the outlet gases,* T_R — *dew point of the outlet gases*

the regenerative bleeds and to attain a high temperature of the preheated feed water. The additional advantage of the considered application of the HAT results from the independence of its electric power from the utilization of the outlet gases.

The temperature of the outlet gases of the HAT depends on the applied scheme. Many authors assume, that after the compression of air an aftercooler of the compressed air is installed, e.g. Bram and De Ruyck (1996), Gallo (1996), Xiao et al. (1996), Di Maria and Mastroianni (1999). In this case the temperature of the humidified air entering the regenerative preheater becomes lower and therefore also the temperature of the outlet gases flowing out from the regenerative preheater decreases. This method does not improve the efficiency of the HAT. The enthalpy of the outlet gases becomes smaller, but another loss of energy appears i.e. the heat rejected in the aftercooler. The chain of the irreversible processes becomes longer, and therefore the efficiency cannot be better. In the following considerations a scheme without any aftercooler is being considered.

Figure 1 presents the scheme of the considered HAT. This scheme does not contain any scrubber. The liquid water is injected immediately into the compressed air, as assumed also by De Ruyck et al. (1997). In order to avoid liquid droplets after the injection, a staggered injection was assumed, according to Szargut and Szczygiel (1999). An open system of blade cooling was taken into account, according to Szargut et al. (1999). The cooling air is taken from the outlet of the compressor.

2. Introduction of the Humid Air Turbine (HAT) into the Scheme of a Steam Power Plant

Figure 2 presents an exemplary enthalpy-temperature diagram for the gases flowing out from the regenerative preheater of the compressed air. The diagram has been calculated for a system with the temperature 1300 ^{o}C before the turbine. The dew point of combustion gases is 64 ^{o}C.

Because of the shape of the enthalpy-temperature diagram different variants of the preheating of the feed water can be realized. The maximum attainable temperature of the preheated feed water is about 122 ^{o}C. If the water stream were to be constant, the preheating of feed water would run according to the line **a**. In this case the utilized part of the enthalpy of outlet gases would be small.

A better utilization of the enthalpy of outlet gases can be attained by introducing the course **b** of preheating. In this case the flow rate of the preheated feed water is greater but its final temperature is lower than in the case **a**. The exergy losses in the part $\mathbf{b_2}$ of the water preheater are great.

Smaller exergy losses in the water preheater can be attained by the organization of the preheating according to the line $\mathbf{b_1a_2}$. In this case up to the temperature of about 60 ^{o}C the total stream of the feed water would be preheated and in the further part of the preheater the flow rate of the feed water would be about 5 times smaller (this proportion results from the slope of the enthalpy-temperature lines of water).

In the case **a** the ratio of the flow rate of feed water to the flow rate of the outlet gases is small. Therefore at a given initial power of the steam power plant, the necessary power of the primary HAT would be great. In the case b the necessary power of the primary HAT would be much smaller. In the case $\mathbf{b_1a_2}$ this power would be slightly greater than in the case **b.**

3. Influence of the Flow Rate Changes of Bleed Steam on the Pressure Values in Bleeds

The introduction of the primary HAT into an existing steam power plant would change the pressure values in the bleeds of the steam turbine. The reduction of the flow rate of bleed steam in low pressure regenerative bleeds evokes an increase of the steam pressure in these bleeds, hence a decrease of the enthalpy drop in the high pressure part of the turbine and an increase of the enthalpy drop in the low pressure stages. If we assume, that three lower bleeds are partially or totally substituted by the primary HAT, the following formula expresses the increase of the electric power of the steam turbine complex:

$$\Delta \dot{W}_s = \eta_{me}[\dot{W}_h' - \dot{W}_h + \sum_{i=1}^{i=3} (\dot{G}_i' \; h_i' - \dot{G}_i h_i)] \quad (2)$$

where:

$\dot{W}_h, \dot{W}_h'$ — internal power of the high pressure part of the steam turbine before and after the introduction of the primary HAT,

$\dot{G}_i, \dot{G}_i'$ — flow rate of steam through the i-th low pressure stage of the turbine, before and after the installation of the primary HAT,

h_i, h_i' — specific enthalpy drop in the i-th stage before and after the installation of the primary HAT,

η_{me} — electromechanical efficiency of the steam turbine complex.

The increase of the pressure in the low pressure bleeds can be approximately determined by means of the analogy to the pressure drop in long pipelines:

$$\frac{\dot{G}_i'}{\dot{G}_i}=\sqrt{\frac{p_i'^{\,2} - p_{i-1}'^{\,2}}{p_i^2-p_{i-1}^2}} \qquad (3)$$

where:

p_i, p_i' — steam pressure at the inlet to the i-th stage, before and after the introduction of the primary HAT (p_0 denotes the pressure in the condenser).

Within the high pressure part of the steam turbine the steam flow should remain constant after the installation of the primary turbine. Because of the great difference between the steam pressure at the inlet to the high pressure and low pressure parts of the turbine, the changes of pressure in low stages exert (according to Eq.(3)) a very small influence on the pressure at the inlet to the high pressure part.

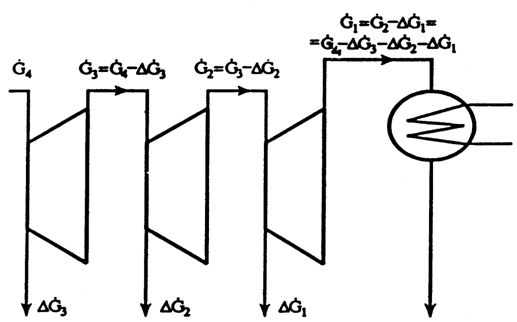

Figure 3. Denotation of the steam streams in the low pressure stages of the turbine

The denotations of steam streams in low stages of the steam turbine have been shown in *Figure 3.* It has been assumed, that the i-th regenerative stream is taken before the i-th stage of the turbine. Therefore the ratio of the streams flowing through the low pressure stages, appearing in Eq.(3), can be expressed as follows:

$$\frac{\dot{G}_i'}{\dot{G}_i}=\frac{\dot{G}_4-\sum_{k=i}^{k=3}\Delta\dot{G}_k'}{\dot{G}_4-\sum_{k=i}^{k=3}\Delta\dot{G}_k} \qquad (4)$$

where:

$\Delta\dot{G}_k, \Delta\dot{G}_k'$ —flow rate of the steam carried for regeneration from the inlet to the k-th stage of the turbine, before and after the installation of the primary HAT.

The decrease of the internal power of the high pressure part of the steam turbine, appearing in Eq.(2), results from the change of the specific steam enthalpy at the inlet to the third low pressure stage:

$$\dot{W}_h - \dot{W}_h' = \dot{G}_4\,(i_3' - i_3) \qquad (5)$$

where:

i_3, i_3' —enthalpy of steam at the inlet to the third stage before and after the installation of the primary HAT.

The power of the primary HAT should be selected according to the amount of heat absorbed by the preheated feed water. The energy balance of the HAT together with the feed water preheater can be formulated as follows:

$$\dot{n}_g\,(H_l + g_w\, i_w) = \frac{1}{\eta_{me\,g}}\,\dot{n}_g\, H_l\, \eta_{E\,g} + \frac{\dot{Q}_{fw}}{\eta_{he}} \qquad (6)$$

where:

$\dot{n}_g, H_l$ — flow rate and the low heating value (LHV) of the gaseous fuel,

g_w — amount of liquid water injected into the compressed air, per unit of fuel,

i_w — specific enthalpy of the injected water,

$\dot{Q}_{fw}$ — heat absorbed by the feed water preheated in the waste heat boiler of the HAT,

$\eta_{me\,g}$ — electromechanical efficiency of the HAT complex,

η_{he} — efficiency of the preheater of the feed water.

The chemical energy in Eq.(6) has been expressed by means of the lower heating value. In this case the state of vapour is accepted as the reference state of H_2O . Therefore the specific enthalpy of liquid water is negative and the zero value of the enthalpy of the outlet gases corresponds to their state after cooling to the environmental temperature but without any condensation of water vapour.

From Eq.(6) the consumption of gaseous fuel can be calculated:

$$\frac{1}{\dot{E}_{ch\,g}} = \frac{\eta_{he}}{\dot{Q}_{fw}}\left(\frac{\eta_{me\,g} - \eta_{E\,g}}{\eta_{me\,g}} + \frac{g_w\, i_w}{H_l}\right) \qquad (7)$$

The electromechanical efficiency of the HAT complex can be expressed as:

$$\eta_{me\,g} = \eta_{el} \frac{\eta_{mT} - \dfrac{\alpha}{\eta_{mC}}}{1 - \alpha} \tag{8}$$

where:

η_{el}	— efficiency of the electric generator,
α	— ratio of the internal power of the compressor and turbine,
η_{mT}, η_{mC}	— mechanical efficiency of the turbine and compressor.

Assuming $\eta_{mT} = \eta_{mC} = 0.98$, $\eta_{el} = 0.99$, $\alpha = 2$, one obtains $\eta_{me\,g} = 0.93$.

4. Exemplary Calculations

Exemplary calculations have been performed for a steam turbine set having the electric power 360 MW. The parameters of the bleed steam in low pressure bleeds are as follows:

p_{u1} = 0.038 MPa, T_{c1} = 74.2 ^oC,
i_{u1} = 2565 kJ/kg, $\Delta\dot{G}_1$ = 17 kg/s,
p_{u2} = 0.106 MPa, T_{c2} = 100.7 ^oC,
i_{u2} = 2695 kJ/kg, $\Delta\dot{G}_2$ = 13 kg/s,
p_{u3} = 0.314 MPa, T_{c3} = 134.5, T_3 = 204.9^oC,
i_{u3} = 2875 kJ/kg, $\Delta\dot{G}_3$ = 18 kg/s.

where T_c denotes the temperature of condensation of the bleed steam, $\Delta\dot{G}$ its flow rate.

The flow rate of steam at the inlet to the condenser is $\dot{G}_1$ = 254 kg/s. The flow rates in the low pressure stages of the steam turbine are $\dot{G}_4$ = 302 kg/s, $\dot{G}_3$ = 284 kg/s, $\dot{G}_2$ = 271 kg/s. The pressure in the condenser is 0.007 MPa, the feed water has a temperature before the first regenerative preheater 40 ^oC, before the second 70 ^oC, before the third 95 ^oC and before the fourth one 132 ^oC. The considered power plant has an efficiency of 36%, the electromechanical efficiency of the turbine set is 97%.

From the cited enthalpy values of the bleed steam there result the internal isentropic efficiencies of the third and second low pressure stages of the turbine (0.87 and 0.78). For the first stage the value 0.78 has been assumed. Hence the steam enthalpy at the inlet to the condenser is 2378.6 kJ/kg, and the values of the enthalpy drop at particular low pressure stages are:

h_1 = 180 , h_2 = 130 , h_3 = 186.4 kJ/kg.

The internal power of the low pressure stages of the turbine are:

$\dot{G}_1 h_1 + \dot{G}_2 h_2 + \dot{G}_3 h_3 =$ 254·186.4 + 271·130 + 284·180 = 133 696 kW.

Variant 1. Preheating of the feed water according to the line a (*Figure 2*)

In the considered case the regenerative bleeds 1 and 2 can be switched off totally and the bleed 3 partially. In the waste heat boiler of the HAT the feed water can be preheated to 122 ^oC, and then additionally heated by means of the steam from the third bleed to a temperature of 132 ^oC. The decreased stream of bleed steam taken from the third bleed results from the energy balance of the preheater fed with steam:

$$\Delta\dot{G}_3{}' \,(i_{u3}{}' - i_{k3}{}') = (\dot{G}_4 - \Delta\dot{G}_3{}') \,(i_{fw\,4} - i_{fw\,3}) \tag{9}$$

where:

i'_{u3}, i'_{k3}	— enthalpy of steam from the third bleed and of its condensate after the introduction of the primary HAT,
$i_{fw\,3}, i_{fw\,4}$	— enthalpy of feed water at the inlet and outlet of the additional preheater.

When solving Eq.(9) the iterative procedure should be applied, because after switching off the bleeds the pressure in the third bleed and also the enthalpy of bleed steam will be changed. In the first step of iteration we obtain $\Delta\dot{G}_3{}'$ = 5.4 kg/s. The changed streams of steam in the low pressure stages of the steam turbine will be:

$$\dot{G}_1{}' = \dot{G}_2{}' = \dot{G}_3{}' = 296.6 \text{ kg/s.}$$

The values of the changed pressure in the bleeds results from Eq.(3):

p_{u1}' = 0.0442, p_{u2}' = 0.114,
p_{u3}' = 0.326 MPa.

The changed enthalpy values of the bleed steam and of the steam flowing to the condenser are:

i_{u3}' = 2881 , i_{u2}' = 2704 , i_{u1}' = 2583 ,
i_0' = 2379 kJ/kg.

The changed values of the enthalpy drop are:

h_1' = 204 , h_2' = 121 , h_3' = 177 kJ/kg.

The second corrected solution of Eq.(9) is: $\Delta\dot{G}_3{}'$ = 5.42 kg/s. Hence the correction is not essential.

The rate of heat absorbed by the feed water in the waste heat boiler of the HAT is:

$$\dot{Q}_{fw\,g} = 296.6\,(512.2 - 167.7) = 102\,208 \text{ kW}$$

Assuming that the HAT equipped with blade cooling has the energy efficiency 0.512, the electromechanical efficiency 0.91, the consumption of the injection water 77.2 kg/kmol fuel, the lower heating value of the fuel 778 000 kJ/kmol, the enthalpy of the injection water - 2402.4 kJ/kg and the efficiency of the gaseous preheater 0.726 (resulting from *Figure 2*) we obtain from Eq.(7):

$$\dot{E}_{ch\,g} = 708\,870 \text{ kW}, \ \dot{W}_g = 363 \text{ MW}.$$

The primary HAT would have a power similar to that of the steam turbine. The changed internal power of the low pressure part of the steam turbine will be:

$$296.6\,(177 + 121 + 204) = 148\,893 \text{ kW}.$$

The decrease of the internal power of the high pressure part (due to the pressure increase in the third bleed) will be:

$$\dot{W}_h - \dot{W}_h' = 302\,(2881 - 2875) = 1812 \text{ kW}.$$

The increase of the electric power of the steam turbine set will be:

$$\Delta \dot{W}_s = 0.97\,(148\,893 - 133\,696 - 1812) = = 12\,983 \text{ kW}.$$

The increase of the incremental efficiency of the HAT in comparison to the separately operating HAT results from Eq.(1):

$$\Delta \dot{W}_s \,/\, \dot{E}_{ch\,g} = 12\,983/708\,870 = 0{,}018 = 1{,}8\%$$

Hence the utilization of the additionally consumed natural gas is not satisfactory in this case.

Variant 2. Preheating of the feed water according to the line b (*Figure 2*)

In this case the feed water can be preheated in the waste heat boiler of the HAT up to 72 ^{o}C. Hence the first regenerative bleed can be totally switched off. The changed flow rates in the low pressure stages of the steam turbine will be:

$$\dot{G}_1' = 271 \text{ kg/s}, \ \dot{G}_2' = 271 \text{ kg/s}, \ \dot{G}_3' = 284 \text{ kg/s}.$$

The changed values of the pressure in the regenerative bleeds will be:

$$p_{u1}' = 0.0405, \ p_{u2}' = 0.107, \ p_{u3}' = p_{u3} = 0.314 \text{ MPa}.$$

The changed values of the steam enthalpy and of the drops of enthalpy:

$$i_{u3}' = 2875\,, \ i_{u2}' =- 2705\,, \ i_{u1}' = 2574\,, \ i_0 = 2381 \text{ kJ/kg}.$$
$$h_1' = 193\,, \ h_2' = 131\,, \ h_1' = 193 \text{ kJ/kg}.$$

Rate of heat absorbed by the feed water in the gaseous preheater:

$$\dot{Q}_{fw\,g} = 271\,(297.2 - 167.5) = 35\,150 \text{ kW}.$$

According to *Figure 2* the energy efficiency of the gaseous preheater will be 147%. From Eq.(7) it results:

$$\dot{E}_{ch\,g} = 120\,400 \text{ kW}, \ \dot{W}_g = 61.6 \text{ MW}.$$

The changed internal power of the low pressure part of the steam turbine:

$$271{\cdot}193 + 271{\cdot}131 + 284{\cdot}170 = = 136\,663 \text{ kW}.$$

There is no change of the steam enthalpy at the inlet to the low pressure part of the steam turbine. The increrase of the electric power of the steam turbine set amounts to:

$$\Delta \dot{W}_s = 0.97\,(136\,663 - 133\,696) = 2878 \text{ kW}.$$

The increase of the incremental efficiency of the HAT in comparison to the separately operatig HAT is greater than in the variant 1.

$$\Delta \dot{W}_s \,/\, \dot{E}_{ch\,g} = 2\,878/120\,400 = 0.024 = 2.4\%.$$

The total energy effect is smaller than in the variant 1, but the utilization of the natural gas is better.

Variant 3. Preheating of the feed water according to the line b_1a_2 (*Figure 2*)

The scheme of the considered case is presented in *Figure 4*. The total stream of feed water flowing from the condenser is preheated in the gaseous preheater up to a temperature of 60 ^{o}C. Only 19% of the initial stream of feed water can be preheated in the subsequent part of the gaseous preheater. This part will be preheated up to 122 ^{o}C. The mentioned proportion of the streams results from the slope of the lines in *Figure 2*. The remaining part of the feed water will be preheated by means of the bleed steam. Within the third regenerative preheater it will be preheated up to 122 ^{o}C, before its mixing with the water from the gaseous preheater. Then the total stream will be additionally preheated up to 132 ^{o}C, by means of the bleed steam.

In order to simplify the scheme, mixing regenerative steam preheaters have been assumed.

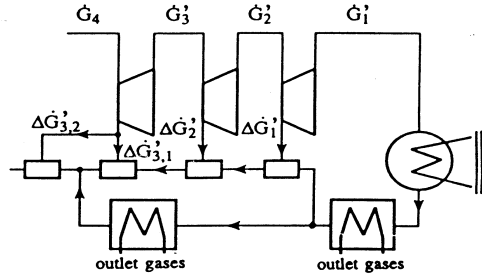

Figure 4. Scheme of the preheating of feed water according to the line b_1a_2

The streams shown in *Figure 3* can be calculated by means of the set of energy balance equations:

$$0.81\,\dot{G}_1^{'}\,[\,i\,(70)-i\,(60)\,] = \Delta\dot{G}_1^{'}\,[i_{u\,1}-i\,(70)] \quad (10)$$

$$(0.81\,\dot{G}_1^{'}+\Delta\dot{G}_1^{'})[\,i\,(95)-i\,(70)\,] =$$
$$=\Delta\dot{G}_2^{'}\,[\,i_{u\,2}-i\,(95)\,] \quad (11)$$

$$(0.81\,\dot{G}_1+\Delta\dot{G}_1^{'}+\Delta\dot{G}_2^{'}\,[\,i\,(122)-i\,(95)\,]=$$
$$=\Delta\dot{G}_{3,1}^{'}\,[\,i_{u\,3}-i\,(122)\,] \quad (12)$$

$$\Delta\dot{G}_{3,2}^{'}\,[\,i_{u\,3}-i\,(122)\,]=\dot{G}_4\,[\,i\,(132)-i\,(122)\,] \quad (13)$$

Additionally the mass balance should be used:

$$\dot{G}_1^{'}+\Delta\dot{G}_1^{'}+\Delta\dot{G}_2^{'}+\Delta\dot{G}_{3.1}^{'}+\Delta\dot{G}_{3,2}^{'}=\dot{G}_4 \quad (14)$$

After solving the set of equations we obtain:

$$\dot{G}_1^{'} = 271.3 \text{ kg/s},\ \Delta\dot{G}_1^{'} = 4.1\ ,\ \Delta\dot{G}_2^{'} = 10.3,$$

$\Delta\dot{G}_{3,1}^{'} = 11.4$, $\Delta\dot{G}_{3,2}^{'} = 4.9$ kg/s. The changed values of pressure in the bleeds of the steam turbine are:

$$p_{u1}^{'} = 0.0405\ ,\ p_{u2}^{'} = 0.1075\ ,$$
$$p_{u3}^{'} = 0.316 \text{ MPa}.$$

and the changed values of steam enthalpy:

$$i_{u3}' = 2877\ ,\ i_{u2}' = 2697\ ,\ i_{u1}' = 2573\ ,$$
$$i_0' = 2378 \text{ kJ/kg}.$$

The changed values of the enthalpy drop:

$$h_1' = 195\ ,\ h_2' = 124\ ,\ h_3' = 180 \text{ kJ/kg}.$$

The rate of heat absorbed in gaseous preheaters:

$$\dot{Q}_{fw\,g} = \dot{G}_1^{'}\ [\,i\,(60)-i\,(40)\,]+$$
$$+0.91\,\dot{G}_1^{'}\ [\,i\,(122)-i\,(60)\,]=36\,130\ \text{kW} \quad (15)$$

The energy efficiency of the gaseous preheater will be 147%, similarly as in the variant 2. The consumption of the chemical energy of natural gas and the power of the primary HAT will be:

$$\dot{E}_{ch\,g} = 123\,770 \text{ kW},\ \dot{W}_g = 63.4 \text{ MW}.$$

The change of the power of HAT in comparison to the example 2 is not great, but the increase of the power of a steam turbine is much greater.

The changed internal power of the low pressure part of the steam turbine is:

$$271.3\cdot 195 + 275.4\cdot 124 + 285.7\cdot 180 =$$
$$= 138\,479 \text{ kW}.$$

According to Eq.(5) the decrease of the internal power due to the increase of pressure in the third bleed:

$$\dot{W}_h - \dot{W}_h^{'} = 302\,(2877 - 2875) = 604 \text{ kW}.$$

Increase of the electric power of the steam turbine set:

$$\Delta\,\dot{W}_s = 0.97\,(138\,479\text{-}133\,696\text{-}604) = 4053 \text{ kW}$$

The increase of the incremental efficiency of the primary HAT in comparison to a separately operating HAT plant is:

$$\Delta\,\dot{W}_s\ /\ \dot{E}_{ch\,g} = 4053/123\,770 = 0.033 = 3.3\%$$

The incremental efficiency of the primary HAT is: $\eta_\Delta = 54.5\%$

The utilization of the natural gas is the best in this case. The required power of the primary turbine is much smaller than in variant 1.

TABLE I. COMPARISON OF THE VARIANTS OF INSTALLATION OF THE PRIMARY HAT

Variant	η_Δ %	$\dot{W}_g\ /\ \dot{W}_s$	$\Delta\,\dot{W}_s\ /\ \dot{W}_g$	$\Delta\,\dot{W}_s\ /\ \dot{W}_s$
1	53.0	1.008	0.0358	0.0361
2	53.6	0.171	0.0467	0.0080
3	54.5	0.176	0.0639	0.0113

5. Conclusions

The research has been performed in order to check the possibility of improving the utilization degree of the expensive natural gas in the HAT cooperating with a steam power plant in a system producing only electricity.

In all the considered variants the installation of the primary HAT does not influence the power of HAT. The power of the steam turbine set becomes greater despite a constant consumption of coal.

The results of the comparison of the considered variants of the installation of the primary HAT in the coal-fired steam power plant are presented in TABLE I.

In the second column the value of the incremental efficiency of the HAT, Eq. (1), is given. Column 3 gives the ratio of the power of the primary HAT to the initial power of the steam turbine. Column 4 presents the ratio of the increase of the power of steam turbine to the power of the primary HAT. Column 5 gives the ratio of the increase of the power of steam turbine to its initial power. The assumed reference efficiencies are: 51.2% for the HAT and 36% for the coal fired plant.

At a constant consumption of coal the greatest increase of power of the steam turbine set can be attained in the case of variant 1, but it requires an installation of very great power of the primary HAT. The utilization of the additionally consumed natural gas is the best in the third variant. Also the ratio of the increase of the power of the steam turbine set to the required power of the primary HAT is the greatest in the variant 3. This can be attained thanks to the reduction of exergy losses burdening the irreversible heat transfer within the waste heat boiler of the HAT.

The improvement of the efficiency of a steam power plant due to the introduction of a primary HAT is not great; but this method can be considered to be an effective tool for improvement of the HAT, because the attained increase of the incremental efficiency of the HAT is sufficiently great.

The third of the considered variants can be recommended for practical application, because in this case the utilization of the expensive gaseous fuel is the best and the investment cost of the primary gas turbine is not very great.

The preheating of the fed water in the steam power plant can also be realized by means of the primary conventional gas turbine. The attainable preheating temperature can be higher than in the case of the HAT and therefore also the increase of the efficiency of the power plant can be greater (Szargut 1999), but the efficiency of the separately operating gas turbine is smaller and therefore the utilization degree of the gaseous fuel in the combined plant is smaller than in the case of the HAT.

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Nomenclature

$\dot{E}$	flow rate of energy, kW
$\dot{G}$	flow rate of steam or water, kg s^{-1}
H_l	lower heating value (LHV) of the gaseous fuel, kJ $kmol^{-1}$
h	specific enthalpy drop, kJ kg^{-1}
i	specific enthalpy, kJ kg^{-1}
$\dot{n}$	flow rate of gas, kmol s^{-1}
p	pressure, MPa
$\dot{Q}$	flow rate of heat, kW
T	temperature, K, °C
$\dot{W}$	power, kW
η	efficiency

Subscripts

E	related to energy
fw	feed water
g	gas turbine or gaseous fuel
h	high pressure part of the steam turbine
he	heat exchanger
m	mechanical
me	electromechanical
s	steam turbine or solid fuel
u	bleed of the steam turbine
w	water
0	condenser

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