EFFECTIVENESS OF USE OF STEAM-GAS CYCLE IN BOTH LOOPS OF BINARY FACILITY

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Thermodynamics of the cycle which is quite new for practical heat and power engineering is reviewed for the purposes of its usage in the second loop of the binary facility instead of conventional Rankine cycle. It is shown that new combination of the cycles in the binary facility provides the reduction of the investments and at the same time higher value of the efficiency coefficient of steam-gas turbine plant in case of using the gas turbines with the unit capacity of no more than 25÷35 MW in the first loop. Comparative analysis of air cycles of the second loop of steam-gas turbine plant is performed.

Key words: *steam-gas cycle, binary facility, nominal conditions, steam turbine, concentrator, air-cooling.*

As it is known modern heat and power engineering predominantly develops based on using the equipment operating with combined steam-gas cycles at the thermal stations. Besides, the combination of Brayton cycle and Rankine cycle are used in binary facilities as a rule, and STIG cycle or Cheng cycle which differs from the STIG cycle by the presence of condenser for returning to the cycle the steam components in the combustion products are used in the contact ones.

Analogous decision (Brayton cycle/Rankine cycle) was made in Gardabani steam-gas power plant project with just the difference that water is injected into the combustion chamber of the first loop, i.e. the first loop is actually operated by steam-gas cycle.

The first stage of the project has been implemented, it included the construction of two gas turbine power units with the total capacity of 110 watts (in ISO conditions). Indeed, the issue of performing the second stage of the construction of the steam turbine operated by Rankine cycle is put on the agenda. As a result the efficiency coefficient of the plant increases from 37% to 50% (in ASO conditions) and the installed capacity makes 150 MW.

As it is known, efficient and reliable functioning of the power system of any country is possible by just the availability of effective spare capacities.

Gardabani gas-steam power plant successfully combines the functions of the spare (auxiliary) capacity especially when based on [1], it is potentially capable of starting and switching to the nominal conditions within less than 10 minutes.

Implementation of Rankine's steam-water cycle at Gardabani power plant will increase its efficiency however, from the other side, it will sharply reduce its maneuverability in case of the necessity of its prompt switching to the maximum capacity. That is why, we think it important to solve the problem in a different way.

At a first sight it is interesting to review the issue of using the combination of Brayton cycle/Brayton cycle (at least for gas turbine equipment (GTE)) of insignificant capacity of $N_{ГТУ} \leq$ 25÷30 watts) as the necessity of using high pressure steam turbine, condenser working on vacuum, etc. is excluded in this case. Consequently, the structure and exploitation of the combined facility will be significantly simplified and the cost of the second loop of the facility will be considerably reduced too.

However, in using air turbine in the simplest cycle of the steam-gas facility in the second loop, possibility of full use of Q_2 actuating medium of the first loop is excluded due to relatively high compression temperature within the compressor. It is accepted that the above mentioned can be avoided by realization of intermediate air cooling in its compression in the compressor, but this measure does not provide maximum possible efficiency.

It is suggested by us to consider combination of cycles in the steam-gas facility which is new for practical heat and power engineering, in particular the Brayton cycle/Poletavkin cycle plus STIG cycle. Such a combination will provide: 1) reduction of air compression temperature value in the compressor of the second loop; 2) increase of the capacity and efficiency coefficient of the second loop facility compared to the Brayton cycle in the second loop; 3) due to insignificant amount of injected water, as a result of lesser values of the compression degree in the second loop, the necessity of using condenser the presence of which would have lead to Poletavkin cycle in the second loop instead of the combination of the Poletavkin cycle and STIG is excluded.

The essence of the cycle in Poletavkin* contact steam-gas facility of [2] is that the air is cooled during the compression process by evaporation of the injected water, and steam-gas mixture obtained in the compressor is supplied to the combustion chamber. Expansion of the steam-gas mixture in the turbine leads to its capacity increase.

Most efficiency of the cycle [2] is achieved in high compression degrees: $\pi=(P_1/P_2)=30\div300$, and in using the cycle [2] in the second loop of the steam-gas facility, the necessity of the realization of maximum values of the efficiency coefficient is excluded due to the absence of the combustion chamber and respectively, due to relatively less air heating in the utilization heat exchanger, as mentioned above, due to less values of the compression degree, but in heat and power engineering even small positive "disturbances" during transformation processes may lead to significant annual reduction of total fuel consumption at the power station.

Figure 1 presents principal diagram of combined stations operated by power transformation according to the combination of the above cycles, and figure 2 shows the cycle of the second loop in T-S coordinates.

On the example of Gardabani gas turbine power station, the first loop of the combined power station contains "twinpac" type gas turbine equipment, i.e. power unit with two FT8-1 gas turbines at 27,5 MW capacity each (in ISO conditions).

Thermodynamic analysis of the steam-gas cycle first of all including identification of turbine operation on steam-gas mixture in the second loop of the facility "can be considerably simplified if we assume that heat content and heat capacity in superheated steam contained in the mixture depend on just the temperature" [3].

In this case, the design of the expansion of the steam-gas mixture is combined with the design of the expansion of each component separately but with the same parameters of the condition [3-5].

*** Water injection into the compressor was first implemented in aircraft power units for forcing the motor pull, but for stationary power equipment, particularly for the steam-gas turbine plant; air cooling during its compression by the evaporation of the injected water was first implemented by P. G. Poletavkin [1].**

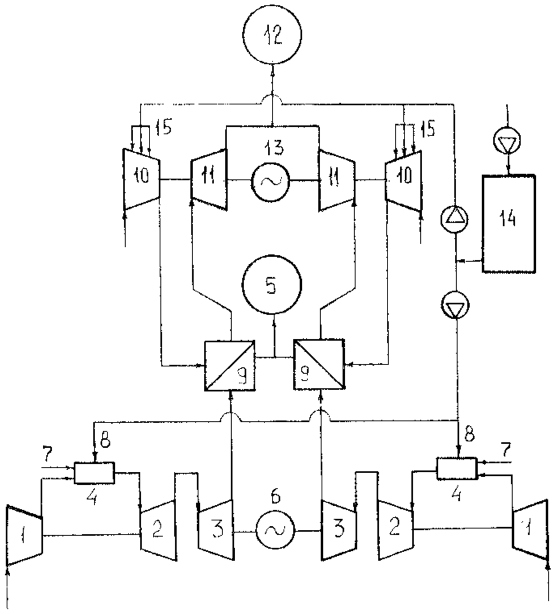

Fig. 1. Principle Diagram of Steam-Gas Facility in Combining Brayton Cycle with Water Injection with Poletavkin Cycle without Cooler with condenser.
First loop: 1 – air compressor; 2 – wire gas turbine; 3 – motor gas turbine; 4 – combustion chamber; 5 – chimney; 6 – power generator; 7 – fuel; 8 – water injection; 9 – interloop water exchanger.
Second loop: 10 – air compressor; 11 – air turbine; 12 – exhaust air pipe; 13 – power generator; 14 – water conditioning shop; 15 – water injection.

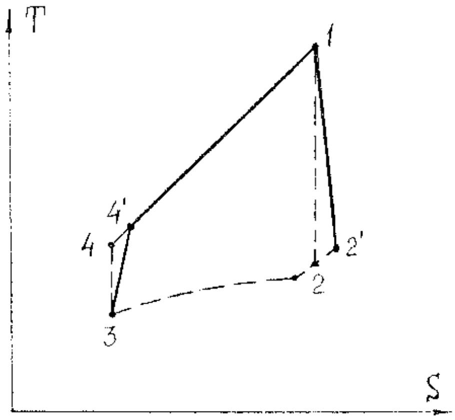

Fig. 2. T-S Diagram of the Cycle of the Second Loop of the Steam-Gas Turbine Plant: 3-4' – actual compression process of humid air in the compressor 10; 4'-1 – steam-air mixture in the heat exchanger 9; 1-2' – actual expansion process of steam-air mixture in the air turbine 11 (at 2' point emission of steam-air mixture).

Isoentropic works of air expansion

$$l_{SB} = i_{1B} - i_{2B} = C_{PB}(T_1 - T_{2B}); \quad (1)$$

Respectively for superheated steam

$$l_{SП}=i_{1П} - i_{2П} = C_{PП}(T_1-T_{2П}), \qquad (2)$$

where i_{1B} and $i_{1П}$ are respectively enthalpy of air and steam in T_1; i_{2B} и $i_{2П}$ - air and steam enthalpy respectively in T_{2B} and $T_{2П}$; C_{PB} and $C_{PП}$ – respectively specific heat capacity of air and steam.

Isoentropic work for expanding the steam-air mixture

$$l_{SCM} = l_{SB} + d_П l_{SП}, \quad \text{кДж/кг.сух.возд.}, \qquad (3)$$

where $d_П = G_П/G_B$ - relation of steam weight in mixture with air weight or steam content.

For isoentropic process, as it is known from thermodynamics, the relation of temperatures is identified by the relation

$$T_{2i}/T_1 = (P_{2i}/P_{1i})^{(Ki-1)/Ki}, \qquad (4)$$

where, in our case, P_{2i} and P_{1i} – partial pressures.

Taking into the account (4), from (1) and (2) will obtain that:

$$l_{SB} = C_{PB}T_1\left[1-\left(\frac{P_{2B}}{P_{1B}}\right)^{(K_B-1)/K_B}\right]; \qquad (5)$$

$$l_{SП} = C_{PП}T_1\left[1-\left(\frac{P_{2П}}{P_{1П}}\right)^{(K_П-1)/K_П}\right], \qquad (6)$$

or, as $P_{1B}=P_1-P_{1П}$ и $P_{2B}=P_2-P_{2П}$,

$$l_{SB} = C_{PB}T_1\left[1-\left(\frac{P_2-P_{2П}}{P_1-P_{1П}}\right)^{(K_B-1)/K_B}\right]. \qquad (7)$$

Partial steam pressure:

$$P_{in} = \frac{P_i}{1+[(R_B/R_П)/d_П]} = \frac{P_i}{1+(0{,}622/d_П)}, \qquad (8)$$

where R_B and $R_П$ – respectively gas constant of air and superheated steam.

So, setting the parameters P_1, T_1, P_2 and $d_П$, by means of the above relations the isoentropic work of the steam-air mixture l_{SCM} is specified:

$$l_B = l_{SB}\eta_{oi}, \quad (9) \qquad \text{и} \qquad l_П = l_П\eta_{oi}, \qquad (10)$$

where η_{oi} – respective internal efficiency coefficient of the turbine.

At the same time, the temperature at the end of actual expansion processes is identified by relations:

for air:

$$T_{2'B} = T_1 - \eta_{oi} (T_1 - T_{2B}), \tag{11}$$

for steam:

$$T_{2'\Pi} = T_1 - \eta_{oi} (T_1 - T_{2\Pi}). \tag{12}$$

Actual expansion work of the steam-air mixture

$$l_{CM} = l_B + d_{\Pi} l_{\Pi} . \tag{13}$$

Respectively, internal efficiency coefficient of the steam-gas turbine plant

$$\eta_i = \frac{l_{CM} - l_K}{q_1}, \tag{14}$$

where l_K – actual compression work of the steam-air mixture in the compressor of the steam-gas turbine plant; q_1- specific heat value brought to the cycle.

Identification of l_K value is essential for the reviewed steam-air cycle especially when its value is specified stage by stage [2].

Parameters at the compressor entrance: temperature T_3=288,15K; pressure P_2=1,033 kgf/cm^2; relative humidity of the suction air φ =60%; efficiency coefficient of the compressor with water injection η_K=0,85.

The temperature of the humid air at the end of adiabatic compression process 3-4 (figure 2)

$$T_4 = T_3 \pi^{(K_{B\Pi}-1)/K_{B\Pi}}, \tag{15}$$

where $\pi = a\pi_0 = (P_4/P_2)$ – degree of pressure increase in the cycle; $\pi_0 = (P_1/P_2)$ - degree of pressure decrease in the cycle; a≈1,05 - coefficient involving hydraulic resistance of the facility loop [2] including the resistance of interloop heat exchanger. $K_{B\Pi}$ – value of compression adiabat of the steam-gas mixture with water drops - $K_{B\Pi}$ =1,08÷1,14 [4].

- **In T_4 partial pressure of water steam $P_{S4\Pi} = P_{4\Pi}$ is defined from the table for saturated water steam [6].**
- **Humidity content of air at the end of adiabatic compression process 3-4**

$$d_4 = 0{,}622 \frac{P_{4\Pi}}{P_4 - P_{4\Pi}} .$$

- **Initial humidity content in T_3 and T_3 respective partial pressure $P_{S3\Pi} = P_{3\Pi}$**

$$d_3 = 0{,}622 \frac{P_{3\Pi}}{P_2 - P_{3\Pi}} .$$

- **Enthalpy of the steam-air mixture referring to 1 kg dry air in T_3 and T_4**

$$i_3 = i_{3B} + d_3 i''_{3\Pi}, \quad \text{kJ/kg of dry air}$$

$$i_4 = i_{4B} + d_4 i''_{4\Pi}, \quad \text{kJ/kg of dry air.}$$

$i''_{3\Pi}$ and $i''_{4\Pi}$ – respectively the enthalpy of saturated steam in T_3 and T_4.

- **Steam content of air at the end of polytrophic compression process 3-4' and respectively at the turbine entrance**

$$d_{4'} = d_1 = d_4 + \frac{1-\eta_K}{\eta_K}\frac{i_4 - i_3}{i''_{4\Pi} - i_ж},$$

where $i_ж$ –enthalpy of the injected water.

- **Amount of water injected into the compressor:**

$$d_B = d_{4'} - \varphi d_3, \quad \text{kg/kg of dry air.}$$

- **Enthalpy of the steam-air mixture at the end of polytrophic compression 3-4'**

$$i_{4'} = i_{B4} + d_{4'} i''_{4\Pi}, \quad \text{kJ/kg of dry air.}$$

- **Actual compressor operation**

$$l_K = i_{4'} - i_3, \ \text{kJ/kg of dry air.} \qquad (16)$$

According to the diagram provided on figure 1, the steam-air mixture from the compressor is supplied to the interloop utilization heat exchanger. In heating the steam-air mixture to T_1, its enthalpy is

$$i_1 = i_{1B} + d_{4'} \cdot i_{1\Pi}, \ \text{кДж/кг.сух.возд.} \qquad (17)$$

where $i_{1\Pi}$ – enthalpy of superheated steam in T_1 and partial steam pressure at point 1.

The amount of heat brought to the actuating medium of the second loop:

$$q_1 = i_1 - i_{4'}. \qquad (18)$$

According to the given calculation method of the expansion process of the steam-air mixture, final air and steam temperature during their separate expansion does not equal to the actual temperature value of the steam-air mixture $T_{2'}$.

It is possible to determine $T_{2'}$ in various ways. The most visual calculation is done my means of the equation of combination of virtual flows of steam and air:

$$C_{PB}(T_{2'} - T_{2B}) = d_1 C_{P\Pi}(T_{2'} - T_{2\Pi}), \qquad (19)$$

i.e.

$$T_{2'} = \frac{C_{PB} T_{2B} - d_1 C_{P\Pi} T_{2\Pi}}{C_{PB} - d_1 C_{P\Pi}}. \qquad (20)$$

The above relations, assuming that during the compression process 3-4' the saturated humid air is compressed, enable to quite accurately identify the true value of the compression work of air moistened during the compression process and the value η_i of the steam-gas turbine plant of the second loop.

Comparative Analysis of Air Cycles of the Second Loop of the Steam-Gas Turbine Plants (SGTP)

In reviewing the cycle in the second loop of SGTP, it is the key issue to use as much heat of the combustion products of the first loop of the facility as possible. Assuming that specific heat capacities of the combustion products and air approximately equal to $c_{рпр} \approx c_{рв}$, which is completely acceptable when running comparative analysis of air cycles if the value of temperature difference in the interloop heat exchanger is Δt=idem, the effective efficiency coefficient of binary cycle will be determined by relation:

$$\eta_{эф}^{БЦ} = \eta_{эф}^{I} + (1 - \eta_{эф}^{I}) \cdot \eta_{эф}^{II} \frac{T_{2ПС} - T_{ОПП}}{T_{2ПС} - T_{ОПС}^{ПР}}, \tag{21}$$

where $\eta_{эф}^{I}$ – effective efficiency coefficient of the first loop of the SGTP; $\eta_{эф}^{II}$ - effective efficiency coefficient of the second loop of the SGTP; $T_{2ПС}$ - temperature of the combustion products at the entrance of the interloop heat exchanger; $T_{ОПС}^{ПР}$ - limiting temperature of the combustion products at the entrance of the heat exchanger equaling to the temperature when using Rankine cycle in the second loop; $T_{ОПС}$ - temperature of the combustion products at the entrance of the interloop of the heat exchanger when using the air cycles in the second loop.

Let's identify the internal efficiency coefficient of the second loop when using the simplest cycle of gas turbine equipment, the gas turbine equipment with stage compression and the suggested cycle in it.

The internal efficiency coefficient of the simplest cycle of the gas turbine equipment:

$$\eta_{i1} = \frac{\tau(1 - 1/\pi^{m}) \cdot \eta_{T} - [(\pi^{m} - 1)/\eta_{K}]}{(\tau - 1) - [(\pi^{m} - 1)/\eta_{K}]}. \tag{22}$$

The internal efficiency coefficient with two-stage compression at the degree of increasing pressure in stage $\pi_1=\pi_2$ и $\pi_1 = \sqrt{\pi}$:

$$\eta_{i2} = \frac{(1 - 1/\pi^{m}) \cdot \eta_{T} - [2(\pi^{m/2} - 1)/\tau \cdot \eta_{K}]}{1 - \{[\eta_{K} + (\pi^{m/2} - 1)]/\tau \cdot \eta_{K}\}}. \tag{23}$$

In (22) and (23) $\pi=P_1/P_2$ – degree of pressure increase in the cycle; $\tau=T_1/T_3$ - temperature increase in the cycle: m=(K-1)/K=0,286; η_T - relative internal efficiency coefficient of the air turbine; - η_K internal efficiency coefficient of the compressor (compressors).

Coefficient including incomplete use of lost heat from the first loop:

$$\eta_{Q} = \frac{T_{2ПС} - (T_{4'} + \Delta t)}{T_{2ПС} - T_{ОПС}^{ПР}}, \tag{24}$$

where Δt =idem – temperature difference in the interloop heat exchanger.

In the simplest cycle of the gas turbine equipment

$$T_{4'} = T_{3}\left(1 + \frac{\pi^{m} - 1}{\eta_{K}}\right), \tag{25}$$

and in the cycle with two-stage compression

$$T_{4'} = T_{3}\left(1 + \frac{\pi^{m/2} - 1}{\eta_{K}}\right). \tag{26}$$

Calculation results $(\eta_i \cdot \eta_Q)=f(\pi)$ of the compared cycles when τ= 683K/288K=2,37; η_T=0,88; η_K=0,85 and temperature difference in the interloop heat exchanger, ΔT=const=50^{0}C are presented on figure 3. Here, there are also the calculation results $\eta_i=f(\pi)$ of the suggested cycle when τ=698K/288K=2,42 and ΔT=var for which η_Q=1 due to realization of the condition $T_{4'} < T_{ОПС}^{ПР}$ in the given cycle (when $\pi \le 10$).

It is seen on figure 3 that the efficiency coefficient value of the suggested cycle in the second loop of the SGTP is significantly higher than the value of conventional cycles within the entire range of π. The efficient coefficient of SGTP of the second loop is achieved within the range of π=6÷8 and equals to $r_i^{max} = 0{,}274$.

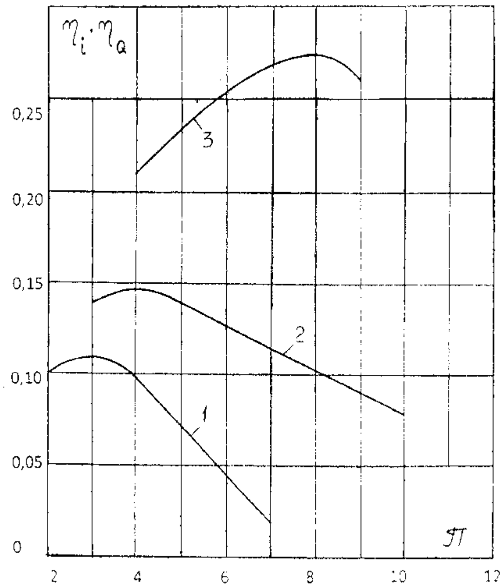

Fig. 3. Dependence ($\eta_i \cdot \eta_Q$)=f(π):
1 – the simplest cycle of the gas turbine equipment; 2 – cycle with two-stage compression; 3 – suggested cycle (for which η_Q =1)

The effective efficiency coefficient of the suggested cycle

$$\eta_{эф}^{II} = \eta_i^{max} \cdot \eta_{\Xi} \cdot \eta_г \cdot \eta_{мг},$$

where η_Σ – coefficient including energy losses at the facility path and on auxiliarities: compressor entrance – interloop heat exchanger entrance – 1%; losses on friction in the interloop heat exchanger – 4,5%; turbine exhaust – atmosphere – 0,07 kgf/cm^2; energy losses on auxiliarities 1%. Totally η_Σ =0,935. $\eta_Г$ and $\eta_{МГ}$ – relatively power and mechanic efficiency coefficient of power generator: $\eta_Г \cdot \Sigma_{МГ}$= =0,98·0,99=0,97. Taking into the account these values, the effective efficiency coefficient of SGTP of the second loop $\eta_{эф}^{II} = 24{,}8\%$.

The effective efficiency coefficient of the gas turbine plant of Gardabani power plant under ISO conditions equals to $r_{эфф}^{I} = 37\%$, and the effective efficiency coefficient of the combined one (in Rankine cycle in the second loop) according to Pratt & Whitney project $\eta_{эфф}^{БЦ} = 49{,}9\%$.

In case of using the suggested cycle instead of Rankine cycle at Gardabani power plant when $\eta_{эф}^{II} = 24{,}8\%$ the efficiency coefficient of the combined SGTP according to (21), as η_Q=1 in the suggested cycle, will be:

$$\eta_{эфф}^{БЦ} = \eta_{эфф}^{I} + (1 - \eta_{эфф}^{I})\eta_{эфф}^{II} = 52{,}6\%.$$

Under the cost estimation, the cost of the second loop under the cycle which is new for practical heat and power engineering makes about $20 million US which is twice less than the case of using Rankine cycle in the second loop of SGTP.

Based on the obtained results we may conclude that Gardabani power plant project on the basis of the suggested cycle has no alternative.

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