P a r t 7

ENERGY SAVING

7.1. Energy saving at the electric and thermal energy generation

7.1.1. Energy saving in different flowsheets

Ilyin E.T. CJSC "Integrated power systems"; Tishin S.G. MPEI (TU)

Energy saving can be divided into three relatively independent spheres: energy saving during the processes of electric and thermal energy generation; energy saving in systems of power distribution from power generating sources to customers and inside customers' systems.

Energy saving at constant flowsheets of electric and thermal energy generation is achieved by:

- application of frequency drives of auxiliary mechanisms;
- rational brightness of equipment;
- supporting of optimal parameters in process of operation according to their load;
- improvement of operation modes and timely halt in reserve of general and auxiliary equipment at the lowered loads;
- application of cogeneration, joint generation of heat and electric power.

All these and other measures are, certainly, well-known too. They give the exact effect and should be applied. But unfortunately, sometimes technical solutions, leading to unwarrantedly high specific fuel consumption, are made.

Lets' consider, for example, Kramatorskaya CHPP (fuel – antracite culm). As the majority of power plants, it was constructed by quetches (stages). Afterwards, at the first quetch of CHPP turbines were dismantled, but three boilers, produced at LMZ, were in operation with steam production of 90…110 t/h each and parameters of steam at the boiler outlet of 3 MPa and 425°C. The second quetch consists of two boilers TP-170 and VPT-25-3, parameters of steam are 9 MPa and 510°C. The third quetch consists of four boilers BKZ-160-100 and two turbines PT-60-90/13, parameters of steam are 9 MPa and 535°C.

Steam losses and steam condensate losses at the first CHPP quetch were compensated by chemically cleaned water, using a scheme of pre-clearance and two-staged Na-cationization.

In design process of CHPP development in order to compensate the losses of the working substance at the second and the third quetches, three-staged evaporating installation was provided, that presents to be the most reasonable in comparison with chemical demineralizing.

Solution, accepted in the project, on dumping of the heating steam drainage of the first stage into deaerator with pressure of 0,6 MPa excluded a possible gradual warming up and start-up of the installation, and also only one possible method of its productivity regulating at the accepted thermal flow diagram by throttling the steam out of collector with pressure of 0,8 … 1,3 MPa, because "the flood" of the heating section of the first stage (hydraulic shocks) occurred. But even in cases when it succeeded to start-up the installation, the quality of distillate was rather poor and far from the requirements of "Operating rules" (ORs).

The reason of poor quality of distillate (secondary steam) is explained by the following. It is known that at low pressures in condition of satiety (boiling of water inside the vaporizer and creation of the secondary steam in it) the factor of salt solubility is negligible at the vapour stage in comparison with its solubility in water, and by this all salts are in water.

In fig. 7.1 a principle design thermal flow diagram of the evaporator installation is shown. It includes four apparatuses, which operate by the three-staged scheme. As the first and the second stages, apparatuses with the heating surface of 550 m^2 each were applied. But the third stage of the installation included two parallel start-up apparatuses with the heating surfaces of 585 m^2. The heating steam was supplied to the first stage of the installation from the over-station collector with pressure of 0,8 … 1,3 MPa.

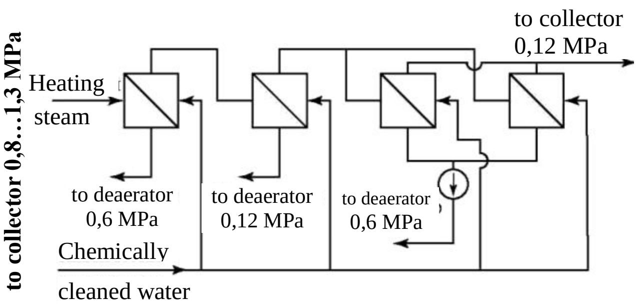

Fig. 7.1. Principle flow diagram of evaporation installation under the project of the Ukrainian Department of the Institute VNIPI- Energoprom

As the heating steam in the second and third stages, the secondary steam was applied. The steam was produced from the chemically cleaned water in the first and the second stages of the installation correspondingly. From the third stage the secondary steam entered the over-station collector at a pressure of 0,12 MPa. Condensate of the heating steam was removed from the first stage to deaerator with pressure of 0,6 MPa; and from the second and third stages into deaerator with pressure of 0,12 MPa.

Salt content of the secondary steam (and, consequently, distillate) is almost completely defined by humidity and mass of the secondary steam, removed from the evaporation surface. By its turn, steam humidity in these conditions is a function of specific velocity of moisture evaporation in the apparatus:

$$w = f\,(W_0'')^n .$$

Exponent n depends on specific velocity of the secondary steam in apparatus W_0'', referred to the diametrical section of the body.

For values of W_0'' being lower than some maximum $W_0''_{\text{limit}}$, the exponent n is 3.

At increasing of W_0'' which exceeds the maximum value, exponent n in the formula for estimating the secondary steam humidity, sharply increases to 6...7.

Calculations showed that in the design scheme at full load, specific steam velocity $W_0''_{,\text{II}}$ at the second stage of the installation was 2,4 times as large as the maximum value of 1,3 m/s and made 1,3·2,4 = 3,12 m/s. Therefore, in this body of the installation humidity of the secondary steam was higher of one that was achieved at the steam velocity, equal to the maximum one by factor of $3{,}12^7/1{,}3^3$ = 1100 and correspondingly by the same factor the salt content of distillate increased.

Owing to the mentioned and a number of minor reasons, all attempts to adjust and start-up this evaporation installation did not give the positive results. In this situation the staff of CHPP found the only possible solution. It was concluded in that the evaporation installation was conserved, industrial and heating turbine bleed-offs were cut off (PT turbine installations were transferred to the completely condensing mode), heat was released to industrial customers from the first stage of CHPP through the pressure-reducing desuperheating station PRDS — 3,3/1,3 MPa, steam was supplied to the boiler plants from the same boilers through PRDS — 3,3/0,12 MPa.

Condensate returned from the production and condensate from boilers applied for compensation of losses of working substance at the second and third stages of CHPP. All this result in sharp decrease in thermal economy of CHPP, that is, considerable increase in specific fuel consumption and correspondent increase in harmful emissions into environment. Specific consumption of reference fuel for power generation was b_{ref}^{e} = 543 g/(kW·h) [at the best SDPPs it's approximately 320 g/(kW·h)]. Under proposal of Thermal Power Plants Department of MPEI, the evaporation installation was reconstructed according to the flow diagram, shown in fig. 7.2.

After balancing and commissioning the four-staged in-

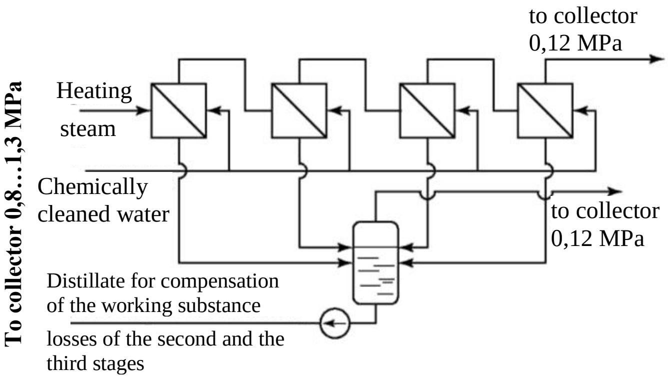

Fig. 7.2. Principle heat flow diagram of the evaporation installation after its reconstruction

stallation operates rather steadily with the quality of distillate, better than the standards from the operating rules. The start-up of installation allowed reducing the fuel consumption at CHPP approximately by 30 t ref.f/h that accordingly improved economic and ecological parameters. However, it took CHPP staff 5 years to correct mistakes of designers, and significant means were spent. It is easy to estimate, how much extra fuel was combusted at CHPP during these years and how much additional harmful emissions were discharged into environment by this reason.

Recently more and more industrial enterprises refuse from CHPP services, in spite of sufficient fuel safety, which can be obtained at combined generation of electric and thermal energy.

Let's assume that some enterprise requires steam in the amount of D_s with parameters p_s, t_s and electricity too. Two options of power supply to the enterprise are possible.

Option 1. Construction of a boiler-house and a SDPP, supplying heat and electricity.

Fuel consumption for heat and electricity generation is estimated as follows:

$$B_Q = \frac{Q_T}{Q_R^N \eta_b}; B_E = \frac{N_E}{Q_R^N \eta_{SDPP}}$$

where Q_T and N_E —thermal and electric energy correspondingly, consumed by the enterprise in time unit, kW; Q_R^N — calorific value of fuel kJ/kg; η_b and η_{SDPP} — efficiency of the boiler and SDPP, accordingly.

Total fuel consumption is defined as follows:

$$B_\Sigma = B_Q + B_E.$$

In fig. 7.3 the principle scheme heat and electrical power supply of the enterprise from separate installations is shown.

Option 2. Electric and thermal energy is generated in one installation (boiler + counter-pressure turbine) (fig. 7.4).

By this, parameters of steam at the turbine outlet and its

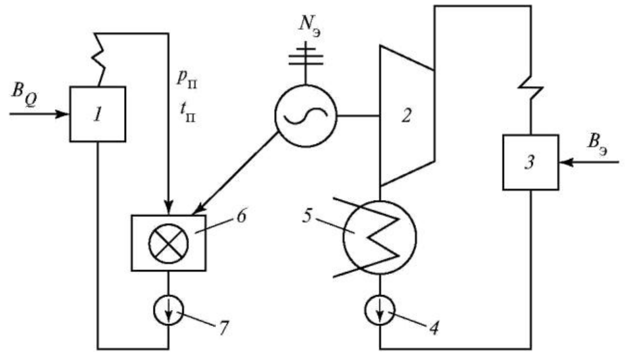

Fig. 7.3. A principle block-diagram of enterprise power supply at separate generation of thermal and electric energy:
1 — steam boiler of the boiler-house; *2* — turbo-generator; *3* — steam boiler of SDPP; *4* — condensing pump; *5* — condenser; *6* — enterprise; *7* — condensate return pump; $B_Э=B_E$; $N_Э=N_E$; $P_n=P_s$; $t_n=t_s$

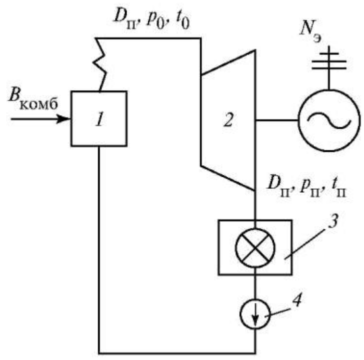

Fig. 7.4. Principle scheme of enterprise heat and electrical power supply from combined installation:
1 — HPC boiler; *2* — turbo-generator; *3* — enterprise; *4* — condensate return pump; $B_{комб}=B_{comb}$; $D_П=D_S$

consumption are the same as in the first option at the boiler outlet, supplying steam to the enterprise.

In this flow diagram it is always theoretically possible to select such values of steam parameters p_0, t_0, at which generator will produce the needed capacity N_E to the enterprise.

Comparison of the considered options of enterprise power supply shows that at the combined generation there are no losses with the cooling water, which are inevitable at N_E capacity generation at the condensing power plant in the first option.

In case of transfer from the option of separate power supply of the enterprise to the combined one, heat losses in condenser are eliminated, making about 50%. By this reason fuel consumption for power generation in the second option will be twice less, that is, $\Delta b \approx$ 0,160 kg ref.f/(kW·h). By this, fuel consumption for heat generation in the compared options is the same.

In the combined installation in comparison with the separate one, fuel saving per hour can be defined by the formula:

$$\Delta B = B_{sep} - B_{comb} = \Delta b \cdot N_E,$$

So, in case of using the counter-pressure turbine of P-100 type in the second option, fuel safety $\Delta B = 0{,}16 \cdot 100 \cdot 10^3 = 16 \cdot 10^3$ kg ref.f/h or 16 t ref.f/h.

Amount of heat, supplied to the customer, may significantly dependent on the season or other circumstances. By this reason, turbines with regulating steam bleed-offs found wide application at CHPPs. At such installations at decrease in steam consumption, supplied to the customer, consumption of steam, entering the condenser through the turbine low pressure part, increases that allows not to reduce electric capacity of the turbogenerator.

Today the greatest part of fuel saving is referred to electricity. Consumers of electricity are in the privileged position because in process of setting the tariffs on heat and electricity, the developed by ORGRES proportional method of distribution of the general fuel consumption of CHPP for heat and electricity generation is used.

The proposed method of separate profitability (see it.6.1.5) results in more uniform distribution of the economic effect of the combined generation of heat and electricity, assuring, by this, the interest of power companies and consumers in the combined power generation.

In real operational conditions a considerable part of time separate power units and power plants, as a whole, operate under partial loads. In this case, especially, for supercritical units, a transfer to the changeable parameters or combined regulation, depending on the level of power unit load, assures considerable efficiency of operation. By this, the total effect of operation is provided not only due to the growth of thermal dynamic efficiency, but also by assuring the reliability and reduction of power expenses for auxiliaries.

During part-load operation, steam consumption for the turbine is decreased. If multiple steam nozzle control and constant pressure before the valves is used, the regulating stage is exposed to the greatest loads. Decrease in steam consumption results in reduction of the pressure at the stage outlet proportionally to the seam consumption change. But since a part of valves is fully open, the heat drop in these streams and steam consumption through the fully open valves increases. As a result, a decrease in steam consumption at the permanent pressure before the regulating valves will lead to increase in flexural stresses in the working blades of the regulating stage.

At multiple steam nozzle control, part-loaded modes, an efficiency of the regulating stage of the High Pressure Cylinder and of the whole turbine decreases. It is connected with steam pressure drop in partially open valves and with heat drop increase in the regulating stage, the economical efficiency of which is always lower, than of the following stages.

Beside this, in case of the permanent pressure mode owing to a pressure drop, steam temperature decreases. And this reduction can be rather significant. For example, in case of the permanent pressure mode operation and load reduction by 50%, steam temperature after the regulating stage can be almost reduced by 70°C. Systematic alternation of a turbine load level at its operation in the load schedules regulating mode result in a constant change of a temperature of the rotor metal and the turbine body in a zone of the regulating stage, causing additional thermal tensions and a low-cycle fatigue of the metal, that means decrease in reliability.

Sufficient benefit in efficiency of the changing pressure is obtained due to reduction of expenses for auxiliaries of the feeding pump.

For example, using of the changing pressure allows to reduce a power of the feeding pump drive of a power unit with the capacity of 300 MW at unloading to 50% of more than by 1 MW in comparison with the permanent pressure operation mode.

Disadvantage of using the changing pressure is a decrease in the unit mobility. In this case the unit mobility is completely defined by the boiler mobility, inertia of which is rather significant and is measured by minutes. Therefore, recently the combined regulating is widely applied. Under the combined method of regulation an operation at the permanent initial pressure to the moment of full opening of one of the valve groups and a transfer to the changing pressure at the following load reduction is meant. Having high economical efficiency, this method provides the small temperature fluctuation in the regulating stage and has a sufficient mobility and high economy in the whole regulating range of power unit loads.

Significant economy can be achieved at the plants by maintaining and holding of the optimal deep vacuum in the turbine condenser. Frequently the reason of vacuum decrease in the turbine is pollution of condensers, which, as a rule, occurs from the water side. Owing to that, a heat transfer factor decreases, water consumption reduces because of hydraulic resistance.

All pollutions by their character can be conditionally divided into three groups:

- mechanical;
- biological;
- salt. In most cases there is a combination of these pollutions.

There are different methods of condenser cleaning, but according to the gained operational experience, today the most effective method is a system of persistent ball cleaning. The essence of such clearing is that through the pipe system of a condenser the balls, made of elastic material, are persistently pumped. Diameter of these balls is a bit bigger than a bore of the condenser pipe. Once the ball enters the condenser pipe inlets, under water pressure it is absorbed into this pipe and pushed through it. By this, the ball itself is a bit deformed due to its elastic properties and wears in some way the inner surface of the pipe, cleaning it from deposits. Application of this system allows increasing power generation without additional expenses on the average by 1,5…2% for steam-turbine plants, combusting organic fuel, and by 3…4% for nuclear power plants. Persistent cleaning of pipe system not only provides better conditions of heat exchange, but also decreases the probability of corrosion in condenser heating plates, because frequently corrosion processes occur more intensively under the deposit layer. A principle configuration of the plant for the ball cleaning system is shown in fig 7.5.

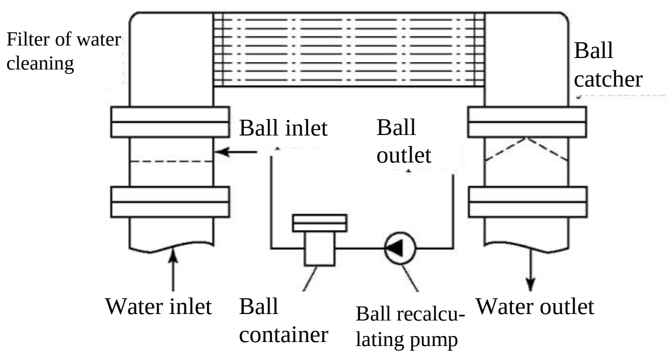

Fig. 7.5. Principle block-diagram of persistent ball cleaning

The stream of circulating cooling water carries the elastic balls through the condensing pipes. They clean the surface of pipes and in a ball catcher, installed at the outlet of cooling water from the condenser, they are separated from the water stream, after that they are sucked off by the pump and again supplied to the inlet of cooling water in the condenser through the branch pipe at the cooling water filter. In circulation loop the balls are loaded through the sluice, installed at the pump pressure pipeline of balls circulation.

The operational experience shows that expenses for installation of the ball cleaning system are repaid in one or one and half a year.