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ASSESSMENT OF EFFICIENCY INDICATORS OF THE STEAM-TURBINE MINI-THERMAL POWER PLANTS BURNING COAL

D.T. Nguyen, D.N. Fam, G.R. Mingaleeva

Kazan State Power Engineering University, Kazan, Russia
mingaleeva-gr@mail.ru

Abstract: *We present results of calculations of main units of steam-turbine mini-thermal power plants of low power (6, 11.4, 12, 20 and 25 MW), intended for operation in autonomous mode. Basing on material, thermal and exergy balances we defined efficiency indicators: exergy efficiency and specific fuel expenses. The units of 6, 20 and 25 MW have the highest exergy efficiency at the level of 33 %. This fact testifies to a possibility of efficient combination of boiler and turbine equipment of this power as well as integration of technological schemes of mini-thermal power plants on the basis of drying and mill equipment, steam boilers and turbines of low power.*

Keywords: *mini-thermal power plant, fuel preparation, steam boilers, steam turbines.*

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Introduction

According to the expert estimates, the share of energy costs in the cost of production of various industries ranges from 3 to 55 %, which basically exceeds the world level in the relevant industries. The highest share of fuel and energy costs is observed in the refining industry - up to 54,7 %. In addition, the dynamics of growth in tariffs for heat and electricity exceeds the growth rate of prices for products, which contributes to an increase in the share of expenditures on energy resources [1]. Construction of a personal source for production of electrical and thermal energy, i.e. a small thermal power plant (mini-TPP), will help to achieve stabilization of costs for this item of expenditure.

Mini-TPPs can be used in remote and underdeveloped regions that are far from centralized power supply (approximately 50–70 % of the Russia territory), in order to meet own needs of certain facilities - residential or industrial, as well as in the event of an emergency.

The relevance of development of small distributed power engineering is also associated with economic recovery in the areas of centralized power supply and, accordingly, with difficulties in operational connection or lack of technological capabilities to connect to power grids.

Nowadays the fuel and energy balance of Russia is significantly shifted to the use of natural gas, whose share in production of heat and electricity exceeds 70 %. Along with the well-known and indisputable advantages of this type of fuel compared to others, this situation poses a real threat to the country's energy security [2]. The main type of fuel for the European part of Russia and the Ural regions, where more than 82 % of the population lives, is natural gas. It is known that the supplied fuel is produced mainly in one gas producing region of the Tyumen region and is transported over a distance of more than 2 thousand km. The existing gas transmission system has a complex and extensive structure and "bottlenecks", the accidental or intentional damage of which can cause enormous damage to the economy of the country. In addition, the gas transmission system itself, the basis of which was formed during the existence of the Soviet Union, is significantly worn out [3]. It is noted that more than 70 thousand km (42 % of the total length) of gas pipelines have been in operation for more than 30 years and require reconstruction. A similar situation occured with compressor stations.

Under such conditions, the new energy facilities being commissioned should be diversified to the maximum extent by the type of fuel used, and small distributed energy facilities should be oriented by local, cheaper fuel in order to be competitive at the cost price of the energy produced. The development of small distributed generation is evidenced by the fact that at present, despite the reforms in the energy sector, competition remains extremely weak and the consumer, even quite large, does not have a choice of supplier of heat and electricity. The available analytical estimates confirm that the cost of centralized power supply is compared with the price of its own generation [4]. At the same time, distributed sources of energy supply can already constitute a significant competition in the energy market and contribute to the establishment of lower prices.

At present, the share of small autonomous energy in Russia is estimated for about 5–7 %, mainly these are diesel power plants with a unit capacity of 340 kW and a total of 17 million kW, producing up to 50 billion kWh and consuming about 17 million tons of fuel equivalent per year [5].

World experience shows that the share of small autonomous energy in the production of electrical energy can be at least 2 times higher, as it is in industrialized countries, where this trend is supported by law. In the UK, for example, part of taxes is returned to owners of environmentally "clean" mini-power plants, in Germany there are benefits and compensations for owners of power plants, and centralized networks receive surplus energy from them at favorable rates. Owners of autonomous energy sources are also exempt from taxes in some US states, they are also compensated for a part of capital expenditures [6, 7]. Thus, in many countries whose governments are interested in developing healthy competition in the energy market, the development of small autonomous energy is ensured by legislative and organizational measures.

At the federal level, in the perspective until 2030, it is planned to significantly increase the share of coal and other solid fuels in production of heat and electricity, since its explored reserves are huge and amount to 193,3 billion tons (of which brown coal is 101,2, coal is 85,3 and anthracite is 6,8 billion tons), which provides the Russian economy with this type of fuel for 550 years [8].

Currently, mini-TPPs actually exist and, although they are designed and built for specific operating conditions, some typical schemes can be identified and classified according to the main equipment: steam boilers with steam turbines, gas turbines with waste-heat boilers and diesel generators [5]. Solid fuel (coal, peat, industrial carbon-containing waste) in these schemes can be burned in the furnaces of steam boilers or processed into energy gas, which is then burned in furnaces of gas boilers or combustion chambers of gas turbines. Equipment for thermal processing of solid fuels – gas generators and pyrolysers – is not yet commercially available. Therefore, introduction of technologies of small distributed autonomous generation, using coal and other types of solid organic fuel, is advisable to start from traditional for Russian energy schemes with steam turbine plants.

Main part

Further we consider the technological scheme of a steam turbine mini-TPP, shown in Fig. 1 [9].

The fuel (Kuznetsky lean coal) in a pulverized state enters the burners of a steam boiler, which is equipped by a two-stage steam superheater, which is designed to prepare steam with the required parameters for a steam turbine. Two stages of economizer and air heater are interstaged into the boiler backpass. In addition, to control the steam temperature before the second stage, a surface steam cooler is installed, to which feed water flows after the first stage of economizer. Feed water and condensate are heated in heaters of low (LPH) and high (HPH) pressure. Network water is heated by steam from a turbine extractor in the main boiler. In peak mode, hot steam from a reduction and cooling unit (RCU) is used to heat the network water.

The algorithm for calculation this scheme assumes the sequence of actions shown in Fig. 2

Kuznetsk coal (T grade) is used as fuel. At the first stage, a steam turbine is selected that corresponds to the required power, which is determined taking into account the redundancy of the units. According to the flow rate and parameters of steam which must be supplied to the turbine, a steam boiler operating on solid fuel is selected. The quality characteristics of the coal used as fuel determine the type and composition of the dust preparation system. Calculation of this block is the most time-consuming, so here it is given in details. Key performance indicators are calculated based on provisions of exergy analysis, which allows one to compare multi-purpose objects, for example, producing heat and electricity, as well as by-products [10, 11].

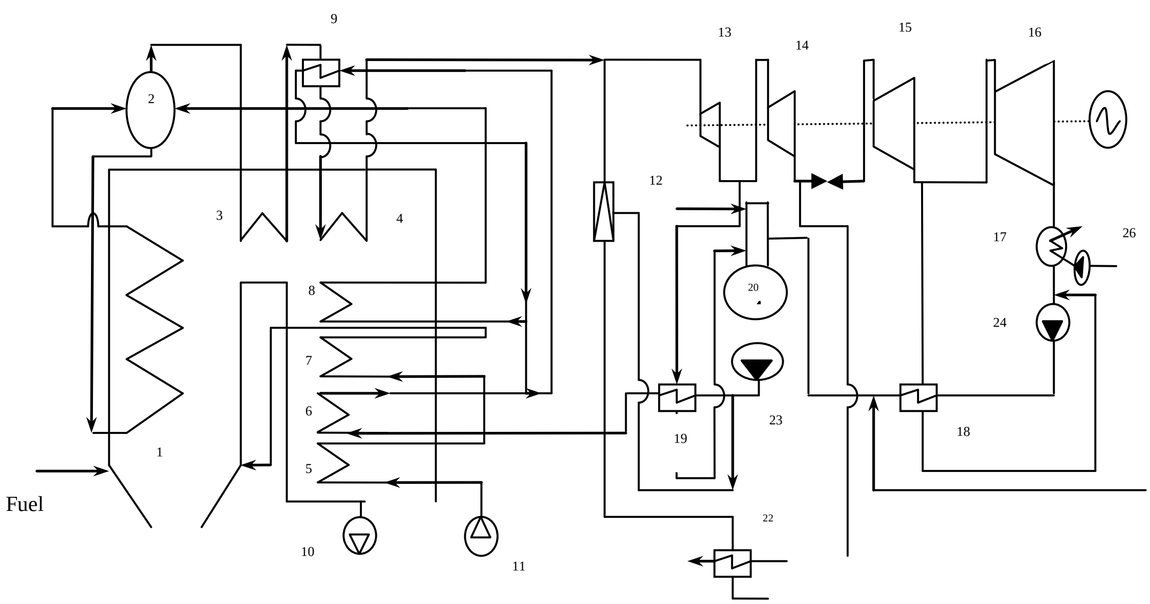

Fig. 1. Technological scheme of a steam turbine mini-TPP: *1* – Furnace; *2* – Boiler drum; *3, 4* -– The first and second stages of steam superheater, respectively; *5, 7* – The first and second stages of air heater; *6, 8* – The first and second stages of water economizer; *9* – Steam cooler; *10* – Smoke pumps; *11* – Blower fan; *12* –CU; *13, 14, 15, 16* – Turbine compartments; *17* – Turbine condenser; *18* – PND; *19* – LDPE; *20* – Deaerator; *21* – Mains heater; *22* – Peak heater; *23* – Feed pump; *24, 25* – Condensate pumps; *26* – Circulation pump; *27* – Network pump

The steam turbine was selected according to the required power and in conjunction with the parameters of steam produced in the steam boiler. The existing range of low-power steam turbines (up to 25 MW) is not very diverse. These turbines are manufactured by the Kaluga Turbine Plant.

The calculation of steam boiler during combustion of Kuznetsk coal (grade T) was carried out in accordance with the regulatory method [12].

For drying and coal pulverizing from typical

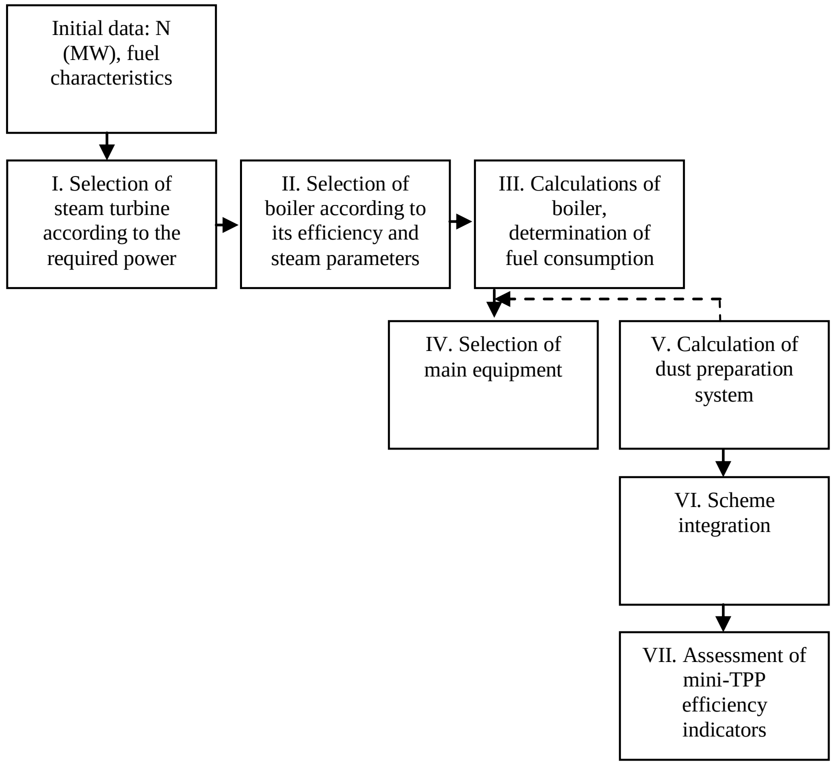

Fig. 2. The algorithm for calculating the technological scheme of the steam turbine mini-TPP

schemes we have chosen the most compact closed individual system with direct injection of coal dust into the boiler furnace and air drying under pressure. For this system we determined the flow rate of the drying agent (air). In such systems, three types of grinding mills can be used: ball drum (BDM), hammer (HM) and mid-roll (MRM). From the existing size range, only ball drum and hammer mills are suitable for mini-TPP.

The heat balance of this type of system is written as follows (all components are indicated in kJ per 1 kg of raw fuel):

$$q_1 + q_{meh} - q_{evap} - q_2 - q_h - q_5 = 0. \tag{1}$$

where q_1, q_2 is heat of drying agent at the inlet to mill and at the outlet from it, respectively; q_{meh} is heat released from grinding bodies of mill; q_{evap} is heat consumed for fuel moisture evaporation; q_h is sensible heat of fuel; q_5 is heat losses to the outside environment through the equipment walls.

Since the system operates under pressure, the component associated with the external air intakes into the system is not taken into account. The calculation of the heat balance components is as follows.

First, we set the temperature of drying agent at the mill inlet. When the fuel is dried with hot air, the upper limit of initial temperature of the drying agent is determined by 10 °C below the air temperature behind the air preheater based on data from [13].

Then we determine the amount of heat released as a result of mill grinding bodies operation q_{meh}. During calculation of q_{meh} we determine the energy costs of grinding. At this, the type of mill is taken into account. If BDM is installed in the system of this type, then the specific energy consumption for grinding, kWh/t, is determined by the formula

$$E_{gr} = \frac{N_n + N_{add}}{B}, \tag{2}$$

where N_n is power consumed by electric motor from the network, kW, and it equals to:

$$N_n = \frac{N_{v.l.}}{e_{em}}, \tag{3}$$

e_{em} is electric motor efficiency, which is in the range 0,92–0,94; N_{add} is additional power consumed for cooling and exciting of motor, for drive of oil pumps and some other needs, kW (for synchronous motors N_{add} ≈50 kW, for high-speed asynchronous motors N_{add} ≈15 kW).

In expression (2) B is grinding efficiency of mill determined by the formula

$$B = K_n K_{ven} a \varphi^{0,8} \Psi_d^{0,6} V_d, \tag{4}$$

Where $K_n = K_{arm} K_{ek}$ is coefficient equal to a product of the following values: K_{arm} is armor form factor equal to 1,0 for unworn wavy armor; K_{ek} is coefficient taking into account the decrease in the performance of mill during operation (assumed to be 0,9); K_{ven} is coefficient taking into account the influence of drum ventilation on the mill performance, its value is determined using graphs based on the results of industrial operation of systems [14]; a is auxiliary value, depending on fuel properties and the resulting dust; Ψ_d is degree of drum filling with balls; V_d is internal volume of the mill drum. All components in the formula (4) are determined according to the relationships presented in [14].

When determining the ventilation coefficient K_{ven} for ventilated BDM, the optimal flow rate of drying agent through the mill is calculated taking into account the grinding conditions, , according to the following relationship, m^3/h:

$$V_{MVopt} = \frac{0,9V_d}{\varphi}\left(1000\sqrt[3]{K_{lo}} + 36R_{90}\sqrt{K_{lo}}\sqrt[3]{\Psi_d}\right) \tag{5}$$

where φ is dimensionless value, which characterizes drum rotation frequency [14]; K_{lo} is grinding form factor; R_{90} is share of particles, remaining at the sieve with particle dimensions of 90 μm.

The mill fan is selected in such a way that its performance is close to V_{MVopt}, i.e. $K_{ven} \approx 1,0$.

In (3) the power consumed by drum rotation (MW), reduced to the motor shaft, is calculated from

$$N_{v.d} = \frac{1}{\eta_{dr}}\left(0,122 D_d^3 L_d n_d \rho_{db} \Psi_d^{0,9} K_{br} K_{tl} + 1,86 D_d L_d n_d S_d\right), \tag{6}$$

where η_{dr} is drive efficiency without taking into account the electric motor efficiency (for gear-driven mills and gearbox η_{dr} =0,865, for friction-driven mills and gearboxes, as well as for gearless gears without gearbox η_{dr} =0,885, for mills with friction drive without gearbox = 0,905); ρ_{db} =04,9 t/m^3 is bulk density of the balls; K_{tl} is coefficient, which takes into account the properties of the grinding fuel; it can be determined from the reference data [14], depending on the fuel type and degree of drum filling with balls; L_d is drum internal length, m; n_d is drum rotation frequency, rpm; S_d drum wall thickness, including armor (along the midline of waves), m.

While determining heat consumption for fuel heating q_h, the temperature t_2 behind the mill is determined from the reference data depending on the criterion of fuel explosiveness K_f [14].

When a hammer mill (HM) is installed into the system, the energy costs for fuel grinding are determined by the formula

$$E = \frac{N}{B} = \frac{N_i N_{i.r.}}{B}\,, \tag{7}$$

Where relative power N_i is determined from

$$N_i = N_{i0} k_{ab} k_{kon}\,, \tag{8}$$

where N_{i0} is relative power, which depends on the rotor speed and the type of separator, is selected according to the reference data [14, 16] (deviation of N_{i0} from the values determined by the schedule is allowed within 15 %); k_{ab}, k_{kon} are operational factors [14].

In (7) $N_{i.r.}$ is idle run power (9 kW), which is determined from:

$$N_{i.r.} = 7 \cdot 10^{-5} u^3 D L \beta c_b \sqrt{m_D} \tag{9}$$

where D is rotor diameter; c_b is coefficient, which takes into account the grinding chamber construction (for mills with open grinding chamber c_b =1, for mills with closed grinding chamber with rotor closing angle of at least 260° c_b =0,6); L is rotor length, m; m_D is the number of hits on a circle, pcs; β is coefficient, which takes into account the relative height of beater:

$$\beta = 1 - 0,7\left(1 - \frac{2h}{D}\right)^4, \tag{10}$$

where h is total beater height, including lugs, m.

The performance of hammer mills with centrifugal and inertial separators is determined by the formula [14], t/h:

$$B = c \cdot 10^{-5} u^3 L m_D^{0,25} \left(1,43 N_i - 1\right)^{0,7} P_f P_v K_{ex} K_{cl}\,, \tag{11}$$

where B is capacity, t/h; c is coefficient, which takes into account the influence of separator design on mill operation (for mills with inertial separator c =1,5, for mills with centrifugal separator c =1,4); u is circumferential rotor speed, m/s; L is rotor length, m; m_D is amount of beaters on a circle, pieces; N_i is relative mill power; P_f is coefficient, which takes into account the influence of fuel physical properties and dust particle size on mill performance; P_v is coefficient taking into account the influence of ventilation on mill performance; K_{ex} is coefficient taking into account the decrease in mills performance in operating conditions due to wear and tear (usually K_{ex} =0,85); K_{cl} is coefficient taking into account the influence of rotor degree of closure (for open-rotor mills K_{cl} =1, for closed-rotor mills K_{cl} =0,7).

For thermal calculation of coal-preparation plant, one takes into account only relationship between dust humidity, temperature of drying agent at the end of installation and initial moisture fuel. Such connection is detected separately for conditions of fuel drying with a mixture of flue gases with air and for fuel drying by hot air. When conducting a thermal calculation of the dust system, the dust humidity is taken according to combustion conditions in boiler, and the temperature at the end of the installation is selected according to the graphs [14].

After thermal calculation, the accepted value of t_2 must be consistent with the relative humidity of drying agent leaving the installation. In all cases, the agent temperature t_2 must be higher than the dew point temperature of water vapor. If the adopted temperature t_2 does not satisfy the specified conditions, then the thermal calculation is repeated, the amount of the drying agent increases, or its initial temperature decreases.

At known temperatures at the inlet to the installation and at the outlet from it, according to the reference data, the heat capacity of dry air (at the inlet) and moist (at the outlet) are determined [14, 16].

A preliminary calculation of drying agent flow rate at the intake to the installation is carried out. Then, the moisture content of the drying agent at the outlet of the installation, calculated to 1 kg of dry gas, is calculated. The weight and volume amount of wet drying agent at the end of installation is determined when drying with hot air.

After the calculations, it is necessary to verify the obtained value of the drying capacity of the mill, t/h, according to the formula

$$B_s = \frac{V_2}{1000 V_{wet}}, \tag{12}$$

Where V_{wet} is amount of wet air, m^3/kg; V_2 is amount of wet drying agent at the end of installation, m^3/h, which is calculated depending on the designed mill capacity B_{design}:

$$V_2 = 1000 V_{wet} B_{design}. \tag{13}$$

The drying capacity must be higher or equal to the designed one, i.e. $B_s \geq B_{design}$. If the available V_2 (for example, the capacity of a mill fan or a fan mill) is less than that obtained from (13), then it is necessary to increase the value adopted earlier or to increase the temperature of the drying agent t_1. In the absence of this capability, the plant capacity will be limited by the drying conditions.

The capacity of the mill fan V_{MF}, m^3/h, installed in front of the pressure mill, is determined by the formula:

$$V_{MF} = \frac{1000 g_1 B_{design}}{\rho_{0air}} \cdot \frac{273 + t}{273}, \tag{14}$$

where t is air temperature before fan, ºC; g_1 is air consumption at the inlet to installation, kg/kg of wet coal; ρ_{0air} is air density, kg/nm^3.

The design capacity for selection of mill fan is taken with a margin of 5 %. According to the technical characteristics of the selected fan is determined by the pressure created by it in the system.

On the basis of the obtained calculated data on the energy and material flows entering into each element of the installation and leaving it, their exergy, exergy efficiency of individual units and the whole technological scheme were determined.

The exergy method is the most common method of thermodynamic study of various energy conversion processes. It allows one to visually determine the degree of perfection and sources of losses in installations and find ways to improve them.

This method is widely used in studies of systems operating on the principle of combined generation of heat and electrical energy [10, 11], however, it is extremely rare applied to small-scale energy facilities operating on solid fuels.

The most important components of exergy are physical and chemical, and in the whole they give thermal exergy E_t. Physical exergy E is the result of a mismatch between temperature and pressure of the considered substance and that of the environment. Exergy, arising from the difference in compositions, is chemical exergy E_{ch}. Chemical exergy of coal can be determined in various ways [17]. In this work, the specific chemical exergy of coal e_{ch}, kJ/kg, is determined by the ratio proposed by V.S. Stepanov [18]:

$$e_{ch} = \left[1,009 + \frac{0,131\mathrm{O} + 0,116W}{100\text{-}(A+W)}\right] Q_l^p, \tag{15}$$

where O is oxygen content in coal in terms of work mass, %; W and A is humidity and ash-content in terms of work mass, %; Q_l^p is low calorific value of coal, kJ/kg.

The main process in coal pulverization systems is drying combined with grinding in coal-grinding mills. All components of exergy balance of drying and grinding processes can be determined by the known dependencies (J/s or W) [5], exergy efficiency $\eta_{s.m.}$ is calculated by the formula:

$$\eta_{s.m.} = \frac{E''_{coal} + E_{evap} + E''_{dr.a.}}{E'_{dr.a.} + \sum_{i=1}^{n} L_i + E'_{coal} + E_{meh}}, \quad (16)$$

where L_i is electric power spent on equipment for drying and grinding; n is amount of devices having an electric drive; E_{evap} is exergy, spent on evaporation of moisture from coal; $E'_{dr.a.}$, $E''_{dr.a.}$ is exergy of the drying agent at the inlet and at the exit of the mill, E'_{coal}, E''_{coal} is exergy of coal entering the mill, and ground coal, E_{meh} is exergy of heat released during grinding of coal.

The exergy efficiency of the steam boiler is calculated using the expression:

$$\eta_{s.b.} = \frac{E''_s}{E'_{p.c.} + E''_{f.w.}}, \quad (17)$$

$E''_{f.w.}$ is exergy of feed water, supplied to waste heat recovery boiler; E''_s is exergy of steam, produced in waste heat recovery boiler; $E'_{p.c.}$ is exergy of combustion products, supplied to waste heat recovery boiler.

Exergy efficiency of steam turbine is determined as follows:

$$\eta_{s.t.} = \frac{N_e + E''_{s.t.}}{E''_s + \sum_{i=1}^{m} L_i}. \quad (18)$$

N_e is electric power generated by steam turbine generator; $E''_{s.t.}$ is exergy of steam sent for heat; E''_s is exergy of steam supplied to the steam turbine from waste heat recovery boiler; L_i is electric power spent for auxiliary equipment; m is amount of auxiliary equipment units of a steam turbine having an electric drive.

For a more complete assessment of efficiency of a mini-TPP operating on solid fuel and generating heat and electricity, the exergy efficiency of a steam turbine mini-TPP is used, which does not include internal flows of a steam turbine plant:

$$\eta_{mini\text{TPP}} = \frac{N_e + E''_{s.t.}}{E'_{coal} + \sum_{i=1}^{k} L_i + E'_v + E'_{f.w.} + E_{meh}}, \quad (19)$$

where k is the total number of auxiliary equipment units of mini-TPP, having an electric drive. The remaining designations in formula (19) are the same as in formulas (16)–(18).

Results and discussion

The selection of equipment was carried out on the basis of well-known methods for calculating and designing fuel preparation systems, boiler plants and steam turbines, the chosen brands for boilers, steam turbines and mills are presented in Table. 1. Analysis of the presented results shows that efficiency of waste-heat recovery boilers of various capacities in the steam generation range from 25 to 90 t/h is approximately at the level of 48 %, with the exception of the KE 25-14-225C boiler, which has lower rates. Exergy efficiency of low-power steam turbines differ significantly. The highest value corresponds to the turbine P-6-1.2 / 0.5 of 6 MW capacity. Turbines of 11 and 12 MW capacities have lower exergy efficiency – 44,5 and 46,9%, respectively. Also, low efficiency possess the HMT 1300/2300/735 and BDM 250/390 coal mills, which are selected for mini TPP with capacities of 12 and 20 MW, respectively.

Exergy efficiency of mini-TPP and specific coal consumption are considered as performance indicators, the results are presented in Table. 2

Table 1

Main equipment for steam-turbine mini-TPP

№	Electrical capacity of mini TPP, MW	Waste heat recovery boiler	Exergy efficiency of boiler, $\eta_{s.b}$, %	Steam turbine	Exergy efficiency of turbine $\eta_{s.t}$, %	Coal grinding mill	Exergy efficiency of drying-mill unit, $\eta_{d.m}$, %
1	6	КЕ 25-14-225С	42,1	П-6-1.2/0.5	79,7	HMT 1000/710/980	27,88
2	11,4	Е 65-3.9-440 КТ	48,2	К 11-1. (KTZ)	44,5	BDM 220/330	27,07
3	12	КЕ 65-3.9-440 КТ	48,2	К 12-4.2	46,9	HMT 1300/2300/735	24,74
4	20	Е 75-3.9-440 КТ	48,3	STU-20	67,5	BDM 250/390	24,26
5	25	Е 90-3.9-440 КТ	47,7	STU-30	70,4	HMT 1300/2030/735	28,68

Table 2

Performance indicators of mini-TPP

№	Electrical capacity of mini TPP, MW	Coal consumption, kg/s	Exergy efficiency of mini-TPP, %	Specific coal consumption, g/MW
1	6	0,757	33,6	126
2	11,4	2,25	21,3	197
3	12	2,246	22,4	187
4	20	2,6	32,3	130
5	25	3,112	33,7	124

Conclusions

The units of 6, 20 and 25 MW have the highest exergy efficiency at the level of 33 %, which indicates the possibility of effectively combining the boiler and turbine equipment of this capacity, as well as completing the technological mini-TPP schemes based on the corresponding drying and mill plants, steam boilers and turbines of small power.

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Authors of the publication

Nguyen Dyk Tuoang – Power engineering department, Kazan State Power Engineering University, Kazan, Russia.

Fam Dang Nyat – Power engineering department, Kazan State Power Engineering University, Kazan, Russia.

Guzel R. Mingaleeva **–** Head of Power engineering department, Kazan State Power Engineering University, Kazan, Russia.

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