

Scientific and Technical Basis for Using External Heat Supply to a Turbine Unit in the Classic Brighton Cycle



Mikhail Laptev, Viktor Barskov, Viktor Rassokhin, Anton Kurnukhin, and Gong Boven

1 Introduction

Now, there is a tendency in Russia to increase the number of low-power power plants. This trend is due to the strategy of long-term development of power engineering until 2030, where the goal is to eliminate the backlog of generating capacities (Fig. 1), as well as to make a smooth transition from a centralized power supply system to a distributed one with the least economic losses [1]. Microturbines on biomass, microturbines on hydrocarbon fuel, solar power units (SPU), and other renewable energy sources (RES) are the potential sources of distributed energy. In the next few years, there will be a need to invest in thermal power plants that were put into operation 40–50 years ago. The microturbine units (MTU) (low-power power plants) are the best option for investment [2]. Such installations have a more developed manufacturing scheme and are also more mobile in contrast to machines running on renewable energy sources, which in turn helps to solve the problem of developing hard-to-reach areas of the far North and Siberia. The transition to microturbine units requires the lowest economic costs, so this product can be implemented at the existing turbine-building enterprises.

Microturbine is a new type of gas turbine unit (GTU), operating in the field of cogeneration type. Now, these machines are used to solve applied problems of energy generation. Structurally, microturbines do not differ from power plants of higher power: an input device, a compressor, a recuperator, a combustion chamber, a turbine, and an output device. However, it is possible to modify the thermal circuit to run on renewable fuel. Thanks to this feature, microturbine installations can be widely used in alternative energy [3]. The authors study the installation that works by burning household waste in specialized waste disposal boilers. This approach will

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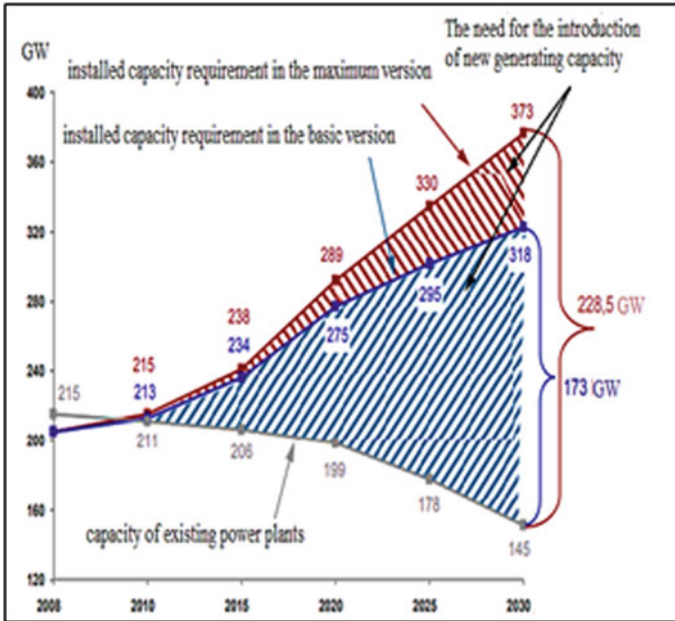


Fig. 1 The need for the introduction of new generating capacity

reduce CO_x and NO_x emissions into the atmosphere due to optimal fuel combustion. Working on this type of fuel allows us to solve two major environmental problems in the world:

- save the ozone layer from destruction;
- reduce the concentration of greenhouse gases in the lower atmosphere.

Thus, microturbine installations are:

- energy-and cost-effective turbine installations;
- allow for a smooth transition from a centralized power supply system to a distributed one;
- they solve two major environmental problems of our time.

2 Literature Review

Power plants with external heat supply are currently not widely used and are mostly experimental with great potential for further development of the power engineering industry. However, over the past ten years, there has been an increased interest in such installations, due to the current problems in the power engineering industry:

- The problem of centralized heat and electricity production;

- Backlog of modernization of the generating capacity from the increased consumption;
- Use of outdated electrical and heat generating equipment;
- Low efficiency of turbines running on alternative energy sources;
- Small interaction of electric power industry enterprises and development institutes;
- High cost of manufacturing and maintenance of installations running on alternative energy sources.

Largely due to these problems, the active development of low-power power plants began. The study and manufacture of such installations are engaged in many countries: the United States, China, South Korea, and Russia. Each country has its own unique approach to the study of these installations, aimed at solving specific problems [4].

USA. Capstone has chosen a unique path aimed at validating the thermodynamic model of the Capstone C30 micro-gas turbine. The methodology is based on a genetic optimization algorithm, where decision variables and goals are set depending on the available experimental data. The results of the studied case emphasize the ability of the method to identify some inconsistencies in the experimental data and lead to a consistent thermodynamic reconstruction of the behavior of the microturbine [5].

As a result, the micro-gas turbine plant revealed the absence of structures that allow achieving the declared values of the overall efficiency and useful power. To do this, a methodology was adopted that determined the optimal set of thermodynamic inputs, using the Euclidean norm to estimate the preferred solution.

Another area that was actively developing was the development of an operational identification method for obtaining simplified dynamic models that were tested on the CapstoneC30 microturbine [6]. For this purpose, a second-order model was identified by adopting a sensitivity theory involving the Lyapunov function, which gives stable solutions in an online environment, updating the model parameters with each change in the physical system. This characteristic can be fruitfully adopted to implement real models in simulators in the time domain, giving accurate results even if some internal parameters change. To investigate the effectiveness of the proposed methodology in obtaining a simplified dynamic model, several simulations and experimental tests were conducted. A good agreement between the simulation and laboratory tests confirms the correctness and applicability of the proposed methodology.

The developed methodology was tested using several simulations and experimental tests conducted on the Capstone C30 microturbine. The results showed that the proposed methodology is characterized by reliability, self-adaptability, high accuracy, stability, and good convergence rate.

China. At the Institute of Engineering Thermophysics, a computational model of heat transfer and pressure drop was established to optimize the design of an annular heat exchanger in a microturbine with a transversely undulating primary surface [7, 8]. The genetic algorithm method is used to solve the optimization problem of a ring heat exchanger with several design variables. Thus, the optimal approach to the design of such a recuperator is formulated. The reliability of the calculated model is verified

by comparing the calculation results with the experimental data of experimental prototype recuperators. Compared to the original design, the optimal design results in a significant reduction in pressure loss if all the constraints are met, indicating that the formulated optimal design approach is effective for a ring heat exchanger. Replacing the optimization goal, the approach to optimal design presented in this paper can be applied to single-purpose optimization for other purposes (compactness, heat transfer area, mass, volume, efficiency, etc.) and even multi-criteria optimization in accordance with the specific requirements of various microturbines [9, 10].

South Korea. Scientists from Korea's Inha University conducted a study to identify the most preferred cycle for a heat recovery system. In their experiment, the Rankine cycle (ORC) and the transcritical CO₂ cycle (tCO₂) were taken, which were applied to a micro-gas turbine plant (MSTU), and subsequently, a comparative performance analysis was performed.

Nonconstructive characteristics of two DND cycles with a change in the load of the MSTU were also analyzed. The performance curve, the efficiency method, and the Stodola equation were used to model the non-constructive operations of all components of the bottom cycle, such as pump, turbine, and heat exchangers [11]. According to the calculations, the output power of the ORC was higher than in the tCO₂ cycle, when the MSTU was operating at full load. On the other hand, since the change in output power when the MSTU load changes are greater in the ORC, the tCO₂ output power becomes greater when the MSTU operates at a partial load below 75%. Therefore, tCO₂ is suitable for a heat recovery system that uses exhaust gas in places where the MSTU must operate at low loads for many hours of its operation. On the other hand, ORC is more suitable in applications where MSTU can operate at almost full load [12, 13].

Russia. Specialists from SPbPU conducted an analysis, the main purpose of which was to find ways to increase the efficiency of the cycle of a simple thermal circuit. To do this, an analysis of four schemes suitable for this type of installation was carried out, because of which the corresponding pros and cons of each of them were identified.

According to the results of the study, the thermal scheme of a two-shaft small-sized gas turbine unit with a compressor driven by a separate electric motor was chosen, as it impresses with its simplicity of design and high reliability. Also, very important advantages of this scheme are the drive of the compressor from a separate electric motor, due to which there is an increase in the efficiency of the MSTU without the use of a regenerative air heater (RV) and the absence of "surge" zones due to the variable speed of the compressor impeller.

Also, specialists from SPbPU participated in the development of a steam turbine unit for TGC-1, operating on the ORC cycle. The main task was to create a new steam turbine plant based on existing prototypes. In addition, there was also an emphasis on improving efficiency. During the design of the installation, several variants of thermal schemes were considered, where the advantages and disadvantages of each of the schemes were identified.

Based on the results of the preliminary calculation, the scheme of a three-circuit power source based on the MTU-500 ORC was selected, which is currently the most

promising both in terms of increasing efficiency through the use of residual heat for thermal needs, and in terms of safety and ease of operation [14].

3 Materials and Methods

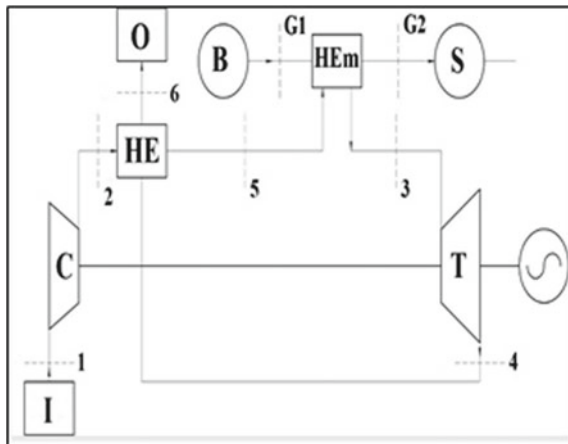
The authors propose a variant of a microturbine installation with an external heat supply (Fig. 2) by replacing the combustion chamber in the thermal scheme with a heat exchanger-mixer.

The description of the scheme is based on the characteristic sections:

- 1–2 Compression of air by the compressor and its displacement into the heat exchanger;
- 2–5 Heating of the working fluid supplied from the compressor;
- 5–3 the Supply of heat to the working fluid;
- G1–G2 Gas supply from the heat recovery boiler to the heat exchanger-mixer;
- 3–4 Expansion of the working fluid in the turbine;
- 4–6 Preheating the air in front of the heat exchanger-mixer;

As can be seen from the figure, the presented thermal scheme of the regenerative type. Based on this installation, a variant calculation of the thermal scheme was performed for three degrees of regeneration: 0.5; 0.7; 0.95—in order to identify the optimal variation. During the calculation of the gas temperature before and after the heat exchanger—mixer, the formulas of water equivalents for air and gas (1), (2) were used for three different values: 0.2; 0.5; 1.0—which also influenced the choice of the optimal strategy.

Fig. 2 Thermal diagram of a microturbine installation:
 I-input device;
 C-compressor; HE-heat exchanger; HEm-heat exchanger-mixer; T-turbine;
 B-heat recovery boiler; S-sparger; O-output device



$$\eta_p^a = \frac{c_p^a \cdot G_a \cdot (T_3^* - T_2^*)}{(c_p \cdot G)_{\min} \cdot (T_{G1}^* \cdot T_{G2}^*)} \tag{1}$$

$$\eta_p^g = \frac{c_p^g \cdot G_g \cdot (T_{G1}^* - T_{G2}^*)}{(c_p \cdot G)_{\min} \cdot (T_{G1}^* \cdot T_2^*)} \tag{2}$$

Based on the calculation, a graph of the dependence of the effective efficiency of the η_e installation on the degree of pressure increase π_k at different degrees of regeneration was drawn up (Fig. 3). It was this graph that played a decisive role in choosing the optimal variant of the thermal scheme.

In the calculations, it was found that the optimal variation of the installation is the scheme with the degree of regeneration equal $\eta_M = 0.7$ and the value of water equivalents equal to 1.0. It was under these parameters was achieved optimal parameters: temperature upstream of the turbine T_3 , the temperature of the exchanger-mixer T_{G2} , efficient work GTU H_E , airflow through the compressor G_A .

Having selected the optimal parameters of the thermal scheme, a variant thermal calculation of the heat exchanger—mixer and the heat exchanger-recuperator was performed. The main goal was to determine the optimal type of heat exchanger. For this purpose, three main types of heat exchangers were calculated: tubular, plate/plate finned. All types differ not only in weight and size and operational characteristics, but the description of heat exchangers is also summarized in Table 1. The thermal calculation was carried out based on the methodology developed at St. Petersburg Polytechnic University. The main initial parameters of the heat exchangers—recuperator and the heat exchanger—mixer were summarized in Tables 2 and 3, respectively.

Fig. 3 Graph of the dependence of the effective efficiency of the η_e installation on the degree of pressure increase π_k at different degrees of regeneration

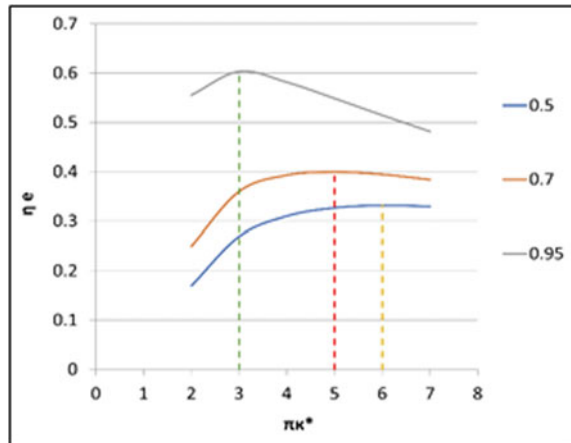


Table 1 Comparative characteristics of heat exchangers

Parameter	Tubular	Plate/plate finned
Surface type	Pipes with cross-sections of various shapes	Thin plate
Weight and size characteristics (conventional units)	5–6	1
Price (conventional units)	0.75	1
Heat transfer (conventional units)	1	3–5
Temperature difference between the heat carrier and the heated medium (°C)	5–10	1–2
Estimated service life (years)	5–10	15–20
Sensitivity to vibration	Sensitive	Insensitive

4 Results and Discussion

Based on the initial data, an optimized thermal calculation of heat exchangers with different combinations of regeneration degrees was performed. During the calculation, it was found that the optimal option for each type of heat exchanger is a combination with $\eta_m = 0.7$ and $\eta = 0.7$. It is with these parameters that the optimal mass and size characteristics of the recuperator are obtained. The data were summarized in comparative Table 4.

Based on the obtained weight and size characteristics, models of the heat exchanger-recuperator were designed, and channels were formed through which the main working fluid and gas flow. The model data for each type is shown in Figs. 4, 5, and 6.

Summarizing all the data obtained, it was found that the tubular heat exchanger has the minimum overall characteristics, but has the highest mass index, so the plate heat exchanger, which is lighter and has the best performance characteristics, was chosen as the optimal one.

Summing up the entire microturbine, it can be stated that installations with an external heat supply have excellent performance. In particular, there was an efficiency value of 33.1%, which is a good result for low-power power plants. The temperature of the working fluid in front of the turbine $T_3 = 873$ K. Among the main advantages of the installation can be noted: multi-fuel, as well as environmental friendliness. The disadvantages of this power plant can be attributed to the high cost, in comparison with powerful gas turbine installations.

Table 2 Initial parameters for the thermal calculation of the heat exchanger-recuperator

№	Parameter	Meaning												Unit of measurement	
		Tubular						Plate							
		0.5	0.7	0.95	0.5	0.7	0.95	0.5	0.7	0.95	0.5	0.7	0.95		
1	Air consumption, G_{a1}	0.81	0.77	0.87	0.8	0.77	0.87	0.8	0.77	0.87	0.87	0.8	0.77	0.87	kg/c
2	Air consumption, G_{a2}	0.81	0.69	0.87	0.72	0.69	0.87	0.72	0.69	0.87	0.78	0.72	0.69	0.78	kg/c
3	The outlet pressure from the compressor, P_2	680,736	486,240	291,744	583,488	486,240	291,744	583,488	486,240	291,744	486,240	583,488	486,240	291,744	Pa
4	The outlet pressure from the turbine, P_4	108,784.4	108,784.4	108,784.3	108,784.4	108,784.3	108,784.3	108,784.3	108,784.3	108,784.3	108,784.3	108,784.3	108,784.3	108,784.3	Pa
5	The outlet temperature from the compressor, T_2	462	425.73	376.1	445.01	425.73	376.1	445.01	425.73	376.06	376.06	445.01	883.08	376.06	K
6	The outlet temperature from the turbine, T_4	745.8	823.83	976.6	771	823.83	976.6	771	823.83	979.54	979.54	771	823.83	979.54	K
9	Diameter, d	0.006	0.006	0.006	0.006	0.006	0.006	0.006	0.006	0.006	0.006	0.006	0.006	0.006	m
10	Hydraulic diameter, d_H	0.004	0.004	0.004	0.004	0.004	0.004	0.004	0.004	0.004	0.004	0.004	0.00384	0.00384	m
11	Cross step, S_1	0.0075	0.0075	0.0075	0.012	0.012	0.0075	0.012	0.012	0.012	0.012	0.012	0.10	0.10	m
12	Longitudinal pitch, S_2	0.0065	0.0065	0.0065	0.056	0.056	0.0065	0.056	0.056	0.056	0.056	0.048	0.048	0.048	m

Table 3 Initial parameters for the thermal calculation of the heat exchanger-mixer

№	Parameter	Meaning												Unit of measurement
		Tubular						Plate						
		0.5	0.7	0.95	0.5	0.7	0.95	0.5	0.7	0.95	0.5	0.7	0.95	
1	Air consumption, G_A	0.81	0.77	0.87	0.8	0.77	0.87	0.8	0.77	0.87	0.8	0.77	0.87	kg/c
2	Gas consumption, G_G	0.72	0.7	0.78	0.72	0.69	0.78	0.72	0.69	0.78	0.72	0.69	0.78	kg/c
3	Pressure at the heat outlet of the heat exchanger of the recuperator, P_5	660,314	471,653	282,992	565,983.36	471,652.8	2,822,991.68	565,983.36	471,652.8	2,822,991.68	565,983.36	471,652.8	2,822,991.68	Pa
4	Pressure at the outlet of the heat recovery boiler, P_{G1}	110,000	110,000	110,000	110,000	110,000	110,000	110,000	110,000	110,000	110,000	110,000	110,000	Pa
5	Temperature at the outlet of the heat exchanger of the recuperator, T_5	604	704.41	949.37	608.45	704.41	949.37	608.45	704.41	949.37	608.45	704.41	949.37	K
6	Temperature at the outlet of the heat recovery boiler, T_{G1}	1300	1300	1300	1300	1300	1300	1300	1300	1300	1300	1300	1300	K
9	Diameter, d	0.006	0.006	0.006	0.006	0.006	0.006	0.006	0.006	0.006	0.006	0.006	0.006	m

(continued)

Table 3 (continued)

№	Parameter	Meaning										Unit of measurement			
		Tubular					Plate								
		0.5	0.7	0.95	0.5	0.7	0.95	0.5	0.7	0.95	0.5		0.7	0.95	
10	Hydraulic diameter, d_H	0.004	0.004	0.004	0.004	0.004	0.004	0.004	0.004	0.004	0.004	0.00384	0.00384	0.00384	m
11	Cross step, S_1	0.0075	0.0075	0.0075	0.012	0.012	0.012	0.012	0.012	0.012	0.012	0.10	0.10	0.10	m
12	Longitudinal pitch, S_2	0.0065	0.0065	0.0065	0.056	0.056	0.056	0.056	0.056	0.056	0.056	0.048	0.048	0.048	m

Table 4 Mass and size characteristics of the recuperator

Parameter	Type of heat exchanger			Unit of measurement
	Tubular	Plate	Plate finned	
Volume	0.016	0.042	0.118	m ³
Mass	197	27.13	32.25	kg
Compact size	252.2	285.71	282.55	l/m
Heat transfer surface area	4.03	12.19	33.34	m ²

Fig. 4 Model of a tubular heat exchanger

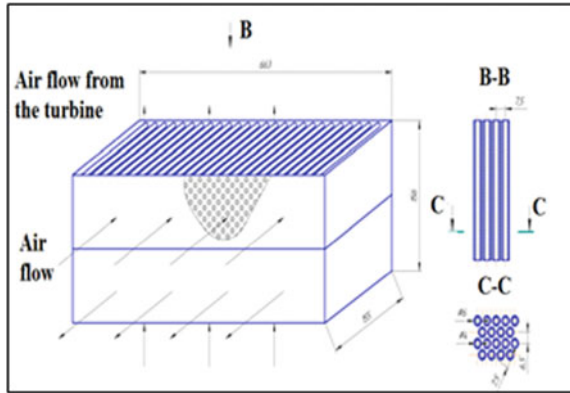
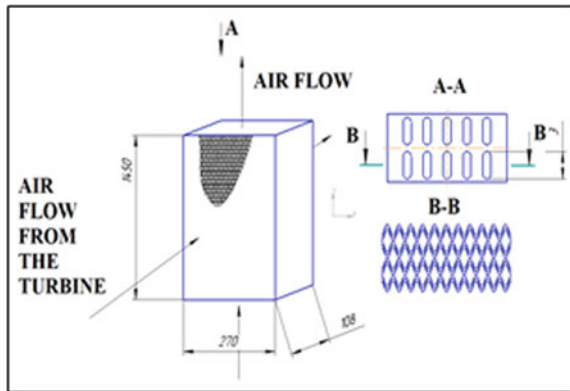


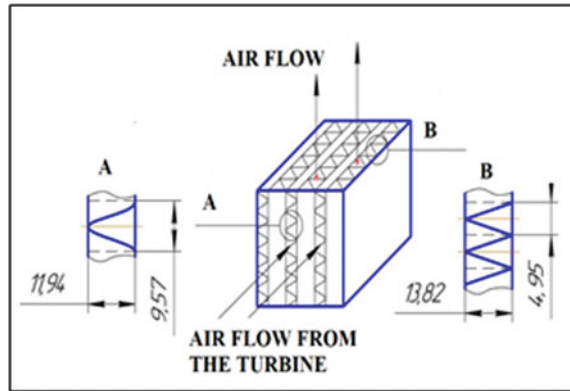
Fig. 5 Plate heat exchanger model



5 Conclusions

Summing up all the above, we can conclude that microturbine installations with external heat supply will help solve a number of problems, as well as create a foundation for future research. In particular, the foundation for the safe and efficient utilization of biomass, with the production of both heat and electricity. If we touch

Fig. 6 Model of a plate-fin heat exchanger



on the economic component, the use of such GTU will help to smoothly make the transition from a centralized power supply system to a distributed one.

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