Utilization of heat and power complex with a capacity of 1 MW

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Abstract. The results of a compact and mobile heat recovery plant design based on the organic Rankine cycle (ORC) with a capacity of 1 MW, which can meet the gas pumping station auxiliaries in power, are presented. The choice of n-pentane as a working fluid, as the cheapest and most affordable on the market and providing condensation of exhaust steam at the atmospheric pressure, and thermal oil ‘Thermolan LT’ as an intermediate heat carrier has been substantiated. The choice of the work process parameters is focused on reliable long-term work with a minimum amount of maintenance. The plant is designed with the possibility to place its main units outside the technological zone of the main production. The design of the plant main components is presented. The turbine is axial, with five stages with a direct generator drive without a reduction gear. The turbine and generator are placed in a ventilated container. The steam generator is a vertically placed shell-and-tube heat exchanger. The air condenser is with horizontally arranged heat exchange tubes and screw-knurled fins. The recuperator is a shell-and-tube heat exchanger with horizontally arranged longitudinally finned tubes. It has been substantiated that for low-potential heat sources and in the conditions of low air temperatures, the use of heat-recovery plants based on the ORC is the optimal solution.

1. Introduction
In the industrial production of the Russian Federation that has developed up to the present moment, there is a significant amount of low-grade heat, which has been used in the technological processes and released uselessly into the atmosphere. These are enterprises of the metallurgical and chemical industry, enterprises of the gas transmission system of the Russian Federation and some electricity generating enterprises.

Thus gas pumping in the RF consumes a great amount of energy constituting 8 ÷ 9% of transported fuel energy [1]. At the same time, thermal emissions into the atmosphere constitute its largest part.

It is quite natural to improve these technological processes by generating electricity through the use of waste heat.

Implementation of projects for such modernization often encounters difficulties related to the need for serious technological changes made to the infrastructure of the facilities already in operation, the need to take them out of service for the period of project implementation, the placement of additional units at existing sites, the emergence of additional functions to manage and maintain heat recovery plants during operation.

In this regard, plants of reduced capacity, which designed to generate not the maximum possible amount of electricity, but only the amount necessary to meet the auxiliaries of these enterprises, are of significant interest. It is expected that the introduction of such plants due to their compactness and their delivery in turnkey solutions will be characterized by a significant simplification of their implementation compared to plants focused on maximum power production.
2. The choice of the type of a heat recovery plant and its basic parameters

2.1. The plant type
The traditional solution to increase the efficiency of heat utilization in the technological cycle is to organize a steam-turbine topping cycle based on water as a working fluid. At the same time, the operation of such plants, especially in the low temperature conditions characteristic of the climate of many regions in the Russian Federation, and in the absence of accessible sources of cooling water, requires close attention of the staff. The risks of emergency situations, for example, freezing of air condensers pipes, can lead to costs not only due to power outages, but can also distract the attention of staff from the implementation of the main technological process.

An alternative to steam turbine topping cycle can be heat-recovery plants using organic working fluids (OWF) that implement the Rankine cycle (ORC). Possessing energy efficiency indicators comparable to those of steam turbines, OWF usually have low freezing temperatures, which exclude the occurrence of serious problems not only during operation, but also in the process of starting and stopping at low atmospheric temperatures. At the same time, on the basis of almost all OWF, quite effective cycles can be organized at temperatures of heat sources not exceeding 200...250°C. This allows the use of intermediate heat-transfer fluids, thermal oils, and due to this, the main units can be transferred from the technological zone of the main production and placed in a convenient location. Thermal oil is a substance having a higher heat resistance and evaporation temperature than OWF. Due to these properties, its phase transitions are excluded in the process of heat removal. Note that the use of an intermediate heat carrier reduces the danger of OWF local overheating in the steam generator.

Such plants are created and operated abroad [2, 3]. In the domestic practice, their closest analogue is a geothermal ORC plant as a part of a binary-cycle Pauzhetskaya GeoPP [4] based on freon as a working fluid.

Thus, for low-grade heat sources and in conditions of low air temperatures, the use of heat utilization plants based on the ORC is the most promising solution both from the standpoint of operational reliability and minimization of operating and maintenance costs.

In this paper, some technical solutions are proposed for a megawatt-type heat utilization plant based on the use of exhaust gas (EG) heat of a gas-turbine unit that drives a gas-pumping unit (GPU). The thermal scheme of a plant is shown in figure 1.

2.2. The choice of working fluid
The choice of an organic working fluid (OWF) of a plant based on the ORC is decisive in terms of not only achieving maximum power efficiency, but also for the design layout of the heat recovery plant as a whole.

For a plant that generates electricity only for its auxiliaries and uses only a smaller part of the available heat, the determining factor should be the minimum cost of its life cycle. It depends on the cost of the plant itself, its service life, overhaul life, cost of safety, maintenance and refurbishment. It should be noted that efficiency, as one of the criteria for grade of such a heat utilization plant, affects the life cycle cost indirectly, namely: due to the increase in efficiency, the thermal power of the main heat exchangers decreases, as well as the area of their heat exchange surfaces and the volume of OWF circulating in a closed circuit and thus the installation cost is reduced. Economic analysis has shown that a significant component of all costs can be the one of refilling the unit with OWF, which varies greatly depending on the type of the working fluid. A comprehensive study of the effect of OWF on these costs is presented in [5], and it was found that the best OWF for heat utilization plants from this point of view and from the standpoint of availability at the market is n-pentane, some of the thermophysical properties of which are listed in Table 1.
Figure 1. Thermal scheme of a heat utilization plant based on the ORC:
1 – a gas turbine driven gas pumping unit,
2 – heat recovery plant,
3 – steam generator,
4 – turbogenerator,
5 – recuperator,
6 – air – cooled condenser (ACC),
7 – thermal oil circuit,
8 – OWF circuit.

Table 1. Some of n-pentane thermophysical properties.

| Name, dimension | Value |
|-----------------|-------|
| Liquid n-pentane at $t = 20 \, ^\circ C$ and $p = 100 \, \text{kPa}$: |       |
| Specific heat capacity (kJ kg$^{-1}$ K$^{-1}$) | 2.32 |
| Specific thermal conductivity (W m$^{-1}$ K$^{-1}$) | 0.113 |
| Density (kg m$^{-3}$) | 626 |
| Saturation temperature at atmospheric pressure 760 mmHg (°C) | 35.7 |
| Specific heat of vaporization | 357.6 |
| Steam n-pentane at 160 °C and 1.5 MPa: |       |
| Specific heat capacity (kJ kg$^{-1}$ K$^{-1}$) | 2.67 |
| Specific thermal conductivity (W m$^{-1}$ K$^{-1}$) | 0.032 |
| Density (kg m$^{-3}$) | 40.2 |
| Sound speed (ms$^{-1}$) | 174.3 |

‘Thermolan LT’ was chosen as the thermal oil, which is characterized by stable heat resistance, availability at the market and moderate price. In addition, the high heat capacity of this substance (3 kJ/kg) reduces its required flow rate and, accordingly, the power spent on its pumping through a closed loop.

It should be noted that minimization of leakages from the plant closed loop and suction into it is of paramount importance both for reasons of reducing the frequency of OWF replacement and safety reasons (for combustible OWF).

An air-cooled condenser (ACC) is one of the most probable sources of air suction under vacuum or leakage if there is excessive pressure in the condenser, since it has the greatest heat exchange surface and a branched manifold system with a large number of connections in comparison with other heat exchangers of the plant. In this respect, n-pentane has an advantage over other OWFs, since when cooling the ACC with atmospheric air, n-pentane condensation occurs at pressures close to atmospheric, and pressure drops in potential leak or suction points are minimal. It is necessary to ensure that the minimum overpressure in the condenser is maintained, as air suction is even more undesirable than leakage. This is due not only to changes in the properties of the working fluid, but also to the danger of creating a fire hazardous environment in a closed loop. Taking into account the above mentioned, the design parameters of the ACC were calculated on the basis of providing a pressure at the inlet to the condenser of 120 kPa with a pressure drop of 10 kPa in it.
2.3. Cycle initial parameters

The choice of the initial temperature at a relatively high level is important for obtaining high efficiency. At the same time, due to the limited heat resistance of the OWF, an unjustified increase in the initial temperature reduces the reliability of the plant, reduces the period between scheduled OWF replacements, which in turn increases the relative time and cost of maintenance.

In this heat recovery plant, ensuring long-term trouble-free operation with a minimum amount of maintenance is of paramount importance, therefore the initial temperature of 160°C is selected, which is lower than 200°C that is usually practiced for this working fluid.

The initial pressure must be high enough. At the same time, the choice of the initial pressure was closely related to the selected value of the initial temperature so that the combination of temperature and pressure corresponded to the dry steam zone along the entire path of its expansion in the turbine.

The margin of the initial pressure should take into account the error of the automated control system during its operation under transient conditions.

In this project, this margin is provided by the selected excess of the saturation temperature with respect to the initial temperature by 15°C. Taking this into consideration, the steam pressure at the turbine inlet is chosen to be 1.45 MPa.

Steam flow rate was determined on the basis of the need to obtain output power of ≈ 1000 kW, and amounted to 16.4 kg/s. Table 2 presents the main parameters of the work process according to this project.

| Name, dimension                      | Value          |
|--------------------------------------|----------------|
| OWF flow rate (kg s⁻¹)               | 16.4           |
| Initial temperature (°C)             | 160            |
| Initial pressure (MPa)               | 1.45           |
| Turbine outlet pressure (kPa)        | 123            |
| Available heat drop (kJ kg⁻¹)        | 101.8          |
| Internal efficiency of the turbine % | 80.7           |
| Real heat drop (kJ kg⁻¹)             | 82.2           |
| Turbine outlet temperature (°C)      | 103.6          |
| Heat exchanger regeneration ratio    | 0.5            |
| The steam temperature at the outlet of the heat exchanger (°C) | 68.5 |
| Heat drop in the recuperator (kJ kg⁻¹) | 69.0 |
| The pressure at the outlet of the condenser (kPa) | 110 |
| Saturation temperature (°C)          | 38.3           |
| Supercooling of condensate (°C)      | 3              |
| Condensate temperature at the condenser outlet (°C) | 35.3 |
| Heat flow in the condenser (kW)      | 6849           |
| Pressure behind condensate pump (kPa) | 436.8         |
| Pressure behind the feed pump (MPa)  | 2.17           |
| The temperature of the liquid OWF at the inlet to the steam generator (°C) | 65.4 |
| Mechanical loss (%)                  | 2.6            |
| Generator efficiency                 | 0.9595         |
| Power at generator terminals (kW)    | 1260           |
| Power to condensate and feed pumps drive (kW) | 100.7 |
| Efficiency of net power unit (%)     | 14.3           |
| Drive power ACC fan (kW)             | 100.2          |
| Thermal oil pump drive power (kW)    | 74.4           |
| Power to provide to the consumer (kW) | 985           |
3. Basic layout solutions

3.1. Layout of the main components
The heat utilizer is the only component of the plant located directly in the area of the GTU operation. Thermal oil heated by the heat of exhaust gases in the heat exchanger is transported through the pipeline to the area where the remaining units of the heat utilization plant are located: steam generator, turbogenerator, recuperator, ACC, thermal oil and n-pentane circulation pump, located at a distance from the GTU. After transferring heat to n-pentane, the thermal oil will be transported in the opposite direction towards the heat utilizer. The circulation pump (for pumping the thermal oil) and its drive are completely sealed and are in a closed loop, which prevents its leakage.

3.2. Turbogenerator
When choosing the turbine parameters, the following was taken into account: the expediency of using a direct (without a gearbox) drive of a serial electric generator with a standard speed of 3000 rpm, minimizing the number of stages as well as technologically acceptable blade sizes.

Based on the variant studies, a 5-stage turbine modification was chosen (figure 2). The decrease in the number of stages in relation to the selected modification leads to the appearance of supersonic speeds and increased opening angles of the flow part in the meridian section, which negatively affects its internal efficiency. The appearance of supersonic speeds in the flow part of the turbine at moderate heat drops at the stages is due to the characteristics of almost all low-boiling working fluids, namely, low sound speeds compared to those for air or water steam, and increased opening angles with a more intense decrease in density with decreasing pressure in the isentropic expansion process.

At the same time, a selected number of stages allowed implementing a turbine with a relatively low pitch diameter, acceptable length of the blades and angles of flow outlet from the nozzle apparatus of the first stage. Peripheral speeds at a pitch diameter are below 100 m/s, which, at low temperatures of the expanding OWF guarantee the absence of any problems in terms of ensuring adequate safety margins.

![Figure 2. Turbine setting.](image-url)

The turbine rotor rests on sliding (sleeve) bearings having a high resource. The turbine face seals with a sealing fluid system ensure the almost complete absence of OWF leakage from the turbogenerator into the environment.
The turbine shaft drives into rotation a synchronous generator TPS-1.5-2M2 Ultrasonic cooled with antifreeze. The turbogenerator is made in a container design modification (figure 3).

**Figure 3.** Container design of the turbogenerator.

### 3.3. Heat recovery unit
Installing a heat exchanger in the exhaust path of a gas turbine should not create additional resistance, which noticeably reduces the power of the gas turbine unit (GTU). At the same time, the start or stop of the heat recovery unit should not interfere with the normal operation of the GTU. The implementation of this property is achieved by installing a bypass gas pipeline and a gate into the exhaust path, which, if necessary, ensures the bypass of the EG past the heat exchanger and ventilation of the volume in which it is located, in the absence of a flow of thermal oil.

### 3.4. Steam generator
Steam generator is a vertically located shell-and-tube heat exchanger. It consists of three parts: an economizer, in which the heat-transfer fluid is heated to the phase transition temperature, the evaporator and the steam superheater (figure 4). The tube bundle consists of tubes $\varnothing 20 \times 2.5$ with a length of 5850 mm. In the intertubular space 21 transverse partitions are located.

**Figure 4.** Steam Generator.

The heat exchange in the steam generator is organized according to the counterflow scheme. Thermal oil heated by GTU exhaust gases enters the intertubular space of the upper part of the steam generator and, moving downward, through the outer surface of the tube bundle gives off
heat initially to gaseous, then to liquid n-pentane. The working fluid of the second loop, liquid n-pentane, from the heat exchanger enters the lower part of the steam generator and moves upwards along the internal cross section of the tubes, and already in the form of superheated steam leaves the upper part of the steam generator.

3.5. Condenser
The air-cooled condenser (figure 5) consists of four heat exchange modules with heat exchange tubes 32×2 arranged horizontally (with a slight slope) in 6 rows with screw-knurled finning in each. Steam condensation is organized inside the tubes in a two-row pattern. The first row includes 113 tubes; the second one includes 30 tubes.

The length of the modules is 12000 mm. The design of the modules has load-bearing elements to ensure proper rigidity. In the flow of cooling air generated by top-mounted fans with blade diameters of 2.5 m, four heat exchange modules are located in two sections; each section has two modules installed along V - figurative pattern. Each section is cooled by air sucked through the heat exchange modules with three fans. In this case, the fans, respectively, are also arranged in two rows.

![Figure 5. Air – cooled condensing unit.](image)

3.6. Recuperator
The recuperator (figure 6) is made in the form of a shell-and-tube heat exchanger and is essentially a part of the pipeline from the turbine to the condenser. The liquid condensate flows inside the tubes 16×2 and as it moves to the other end it is heated by the exhaust steam of the turbine. Heat transfer is organized according to the scheme of a counterflow. The steam enters the intertubular space of the heat exchanger in the exit zone of the heated condensate to the portion of its distribution across the cross section and moves to the opposite end, heating the condensate. For intensification of heat transfer from the steam side, the tubes are made with longitudinal fins.
When selecting equipment for this heat recovery plant, priorities were given to domestic samples. Practically all large equipment used in this project is of domestic production.

**Conclusion**
- A 1000 kW heat recovery plant for generating electricity to meet the gas compressor station own needs, performed according to the ORC scheme with intermediate heat carrier based on n-pentane in the main loop, ensures reliable operation in wide operating conditions, including those at low air temperatures.
- The main technical solutions of the project are aimed at minimum costs when integrating the unit into the existing infrastructure of the plant at minimum cost in manufacturing and commissioning.
- Equipment, mainly of the Russian origin, was selected to complete the heat recovery plant.

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