The working medium for the megawatt class utilization heat and power complex based on Organic Rankine Cycle

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Abstract. The paper proposes a criterion for choosing low-boiling working medium type and the main parameters of the heat recovery unit using the heat of the gas-turbine drive exhaust gases with a capacity of 1 MW for the gas compressor station's own needs and the methodology for determining it at the initial design stage. The data of thermal design and capital costs calculations for the creation of an installation for five types of working mediums are given, as well as the choice of n-pentane as best-satisfying all the requirements.

1. Introduction

Gas turbine drive is most commonly used in the Russian gas transportation system. Usually these are special or conversion gas turbine plants operating in a simple cycle and having the exhaust gas (EG) temperature of 350...550°C. There are many projects of heat recovery units [1, 2], which allow using, with varying degrees of efficiency, the heat energy of the EG to generate electricity without additional fuel costs. Implementation of these projects is complicated by the need to embed these installations into the infrastructure of the gas compressor stations (GCS) which have already been built and put into operation as well as by the possibility of using the generated electricity by local consumers.

In this connection compact plants for electricity generation satisfying GCS’s own needs are of particular interest, such needs make up not more than 20...25% of the plant’s potential. For such plants in addition to the compactness the requirement of minimum purchasing and maintenance costs is of vital importance.

It should be noted that when using a heat source with a potential characteristic of gas turbine plant exhaust gases, heat recovery plants based on Rankine steam water cycle efficiency are superior in energy efficiency to the installation, using Organic Rankine Cycle (ORC) workflow. At the same time, in regions with colder climates, typical of many gas compression stations’ location zones, steam turbine operation is associated with certain difficulties caused by freezing of water in their structures and their destruction. Especially it is manifested in the conditions of emergency shutdown, start-up, repair and even part load modes. In these conditions, heat recovery plants based on ORC get undeniable operational advantages.
2. Requirements review of the organic working medium properties, potentially suitable for use in gas compression stations’ heat recovery plants

Most low boiling working mediums (WM) used in the ORC plants have a freezing point below –50°C (table 1) and this increases their operational technological efficiency greatly.

| Working Medium      | hexamethyl-disiloxane (MM) | Fluorocarbon compounds FC-72, PP1 | n-pentane | fluorocarbon C$_4$F$_{10}$ | refrigerant 134a |
|--------------------|-----------------------------|-----------------------------------|-----------|-----------------------------|------------------|
| Critical point temperature, °C | 246                        | 176                               | 197       | 113                         | 101              |
| Critical point pressure, MPa | 1.94                       | 1.83                              | 3.4       | 2.3                         | 4.1              |
| Saturation pressure at t = 150 °C, MPa | 0.36                      | ~1.0                              | 1.59      | -                           | -                |
| The boiling point at p = 100 kPa | 99.8                       | 56                                | 35.7      | ~2.6                        | ~26.4            |
| Saturation pressure at t = 40 °C, kPa | 11.3                       | ~50                               | 115.7     | 424.7                       | 1017             |
| Freezing point     | ~68                         | ~90                               | ~130      | ~128                        | ~103             |
| The limit of long-term heat resistance, °C | 300                        | 400                               | 300       | 400                         | -                |
| Fire hazard        | combustible                 | non-flammable                     | combustible | non-flammable | non-flammable |
| Explosion hazard   | safe                        | non-toxic                         | dangerous  | non-toxic                    | safe             |
| Toxicty            | non-toxic                   | 5811 / (3011)                     | non-toxic  | non-toxic                    | non-toxic        |
| Cost, rubles/kg    | 600                         | 110                               | 15000     | 350                         |

Practically all WM vapor saturation line in the coordinates t - s has a positive (d$t$/ds$>$0) slope. This eliminates the appearance of steam humidity at the last stages of the turbine and, accordingly, the troubles associated with erosive wear and the need for an intermediate steam separation.

In comparison with water steam, the latent heat of many WMs vaporization is much lower. Heat differences occurring in the ORC turbine, are as a rule low. This leads to a significant increase in mass. In combination with relatively low initial pressures it leads to significant increase in volumetric flow rates of the working medium per unit of power. The latter allows small installations of a megawatt power class to avoid partiality and have acceptable blade sizes. In addition, in a number of cases it is possible to use a direct (without reduction gear) drive of a standard generator with a rotation speed of 3000 rpm with relatively small diametrical dimensions of the turbine.

At the same time, many WMs have a boiling point of +30...+40 °C under excess pressure, and not with a deep vacuum, like water steam. This helps to avoid the problems associated with the outside air suction and reduced specific volumes of WM exhaust steam lead to compact steam pipes.

Among the most famous heat recovery plants are compressor stations plants on the pipeline «Northern Border Pipeline» (USA), equipped with heat recovery units ORMAT® ENERGY CONVERTER (OEC), using n-pentane [3]. At present megawatt-class plants by «AQY ILOM» company (France), using hexamethyldisiloxane as the WM as well as other plants are produced and actively promoted into the market.

Freons, siloxanes, aqueous ammonia, n-pentane, isopentane, butane, isobutane, etc. are used as WM. Various aspects of the use of WM with different properties in geothermal power engineering are considered in [4-6]. In this paper, we analyze the use of various WMs in relation to UHPC with a capacity of 1 MW to satisfy the GCS's own needs.

When choosing a WM in addition to the already mentioned, the important criteria are:

- the most favorable thermophysical properties;
- low toxicity;
- fire danger - explosiveness;
- prospects of production restriction or prohibition, related to their environmental effects (ozone layer, greenhouse effect, etc.);
- high thermal stability;
- acceptable labor input and construction cost taking into account specific properties of the chosen heat carrier;
- availability in the market and an acceptable price;
- the presence of sufficiently complete and reliable information on the characteristics of the WM parameters in the working range.

Labor input and installation costs of the equipment can be determined on the basis of only design studies performed at different stages at different levels of detail. However, these estimates made already at an early design stage which should have sufficient reliability to avoid iteration returns to the initial or intermediate results of the design. The choice of generalized optimization criteria used in the beginning of the design has special importance for this.

3. Variants of utilization heat and power complex constructive schemes

One of the possible options for the utilization heat and power complex (UHPC) scheme is shown in figure 1.

In this variant of the scheme the heat supply to the main circuit of the WM is carried out through an intermediate circuit with a heat transfer fluid. This is necessary in cases when the temperature of the EG exceeds the thermal resistance limit of the WM, so placing its evaporator directly in the GTU exhaust creates the danger of local overheating and decomposition of the working medium. For these purposes thermal oil is typically used which has a high thermal stability and a low freezing point at the same time.

Utilization of thermal oil simplifies heat recovery unit structure 2 (figure. 1) and allows installing the main structure of the UHPC outside gas pumping unit’s technological area, but it decreases energy efficiency of the unit.

From a similar point of view, we can also consider the use of an intermediate heat transfer fluid in the condensation of exhaust steam of WM (figure 2). Using intermediate heat transfer fluid circuit in the process of condensation simplifies UHPC layout reducing leakage of WM or air suctions, reducing the WM refilling volume. However, at the same time, the temperature rises and, consequently, the pressure in the condenser increases.

In the considered options of UHPC the available exhaust gasses heat is only partially used and the efficiency of the cycle, as an independent indicator, is of partial concern. But the indirect effect of the
efficiency for UHPC of fixed power is manifested in heat increase and, accordingly, in the dimensions of all heat exchangers.

The use of a gearbox for the electric generator drive in the turbine unit of the UHPC is necessary for variants that have small volume flows of WM at the inlet to the turbine. The use of the gearbox in these variants allows increasing the internal efficiency of the expansion process and making the turbine more compact, however, the gearbox is expensive, requires a gear lubrication system and supports. Besides, additional losses of mechanical power are inevitable in the gearbox.

If a vacuum condenser is used, then, without special measures to ensure tightness, air suctions can be mixed into the working medium. For some fire-explosive working mediums this is unacceptable, therefore use of atmospheric condensers or condensers at low pressure is preferable. Air suctions must be separated and removed from the closed loop, that can lead to WM leakage. Leakage is also possible in seals of the turbines with condensers under the pressure, exceeding atmospheric. With leakage of WM occurs, its periodic feeding is needed, which leads to an increase in operating costs. Therefore, special attention should be given to the issues of sealing during the creation of UHPC.

With the increase of condensing pressure condenser sizes are reduced. However, on the other hand, this also increases the required flow rate of WM and increases the dimensions of all other heat exchangers. In addition, the efficiency of UHPC decreases.

Selection of WM type and cycle parameters should be performed according to the criterion that considers the effect of all the above aspects on the cost.

4. Generalized criterion of design optimality and method of various UHPCs comparison

Optimization of UHPC construction can be performed on the basis of such criterion, as the cost of the units life cycle, compared with the amount of electricity generated for the whole period of operation.

In the assumption, that the life cycle of all UHPC options and operating costs are the same, the options can be compared by the capital costs of units commissioning. The method of such evaluation is outlined below.

First, the heat calculation was performed, under the following assumptions:

- initial WM steam temperature at the turbine inlet was established considering the maximum possible heat resistance, the maximum initial pressure was taken so that the degree of superheat at the starting point was minimal;
- the pressure in the condenser was taken at the saturation line, based on the condensation temperature of 40°C, but in some cases it was overestimated in order to obtain a minimum overpressure and eliminate possible air suctions to WM;
- gross power was determined by considering mechanical losses and generator efficiency;
- when calculating the net power, the costs of driving a condensate feed pump in the WM circuit (head considered the reserve for regulation), ACC fans and a thermal oil circulation pump;
- WM consumption for the predetermined output was determined for every option, as well as exhaust gas heat supplied into the cycle $Q_{\text{supplied}}$, taken from the condenser $Q_{\text{con}}$, and steam-transmitted to the condensate in the recuperator $Q_{\text{recuperator}}$.

![Figure 2. Steam condensation of WM using the intermediate heat transfer fluid.](image-url)
Further, the effective heat transfer surface of all heat exchangers was evaluated, while:

- the initial and final thermal oil temperature in the intermediate heat transfer circuit were chosen in such a way that the temperature pressure at the WM evaporation in the steam generator (not only integral but also local in counter circuit) would be positive;
- the consumption of thermal oil was determined by the known heat transmitted to it by the EG, and the difference in its temperatures at the inlet to the heat exchanger and the outlet from it;
- the cooling air flow rate in the ACC was calculated based on the waste heat values and its heating degree in the condenser, and to calculate the power to the fan drive their head was calculated on the available analogs ~ 150 kPa.

Thus, for each heat exchanger in the circuit shown in figure. 1, the initial and final temperatures of both heat carriers, their pressures and costs become known.

When estimating the heat transfer coefficients K it is taken into account that for heat recovery unit the heat exchange is mainly determined by the gas side, and data on water heat exchangers can be used for this purpose. The situation is similar for the ACC, in which the heat exchange is determined by the air side and can also be estimated from the available analogues. The situation with the steam generator and recuperator is more complicated, however, it was established through calculations that the relative cost of these devices is not high, and the error in estimating their value will not affect the cost of UHPC considerably.

Now, with the necessary data set for each heat exchanger (HE) the effective area of its heat exchange was determined:

\[
F_{HE} = \frac{Q}{K \cdot \Delta t_{log}},
\]

where \(Q\) is the heat transferred, \(\Delta t_{log}\) - average log temperature head.

The determination of the maintenance cost is based on the cost of analogues per 1 m² of the heat exchange surface for each type of heat exchangers, the required area of maintenance for UHPC, determined as a result of the calculation. Thermal calculation and estimation of the total cost of maintenance allowed us immediately to reject a number of options, as unpromising.

For the rest, a preliminarily evaluation of the turbine design, including type, rotational speed, number of steps, diametrical dimensions, calculation of characteristic angles and velocities for the average diameter. When choosing the rotational speed, only two options were considered (with a direct drive of the generator 1500 or 3000 rpm) and a reduction gear, using a serial gearbox manufactured by OJSC "KTW" (10500 rpm). Depending on the presence or the absence of a reducer, the cost was adjusted consequently. The cost of the turbine part and the generator with the presence in the construction of its own supports, couplings, other devices and units and the control system for all options was assumed unchanged. The cost of the gearbox was estimated at 20 % of the cost of the turbine unit.

One of the most significant components of the cost for some UHPC options was their filling with the working medium. As it turned out, the price of 1 kg of different mediums bodies differs up to 150 times (table 1). To assess the volume of UHPC filling and, accordingly, its cost, the most suitable foreign analogue is chosen, for which the mass of filling is known. Assuming that filling volume is determined by the capacity of the pipeline system, and the medium velocity in the pipelines for all options will be constant and equal to the velocity of this analogue, to calculate the UHPC filling volume it was assumed that it is directly proportional to the ratio of the fluid working medium and analogue volume flows.

Thermal oil circuit filling was calculated according to the heat transmitted to the heat recovery unit, based on a ratio of 1 kg/kWh, and the cost of the filling - based on the price of thermal oil 350 $/kg for all options.
The comparative calculation of the UHPC cost considers the costs of fire safety systems (for combustible WM), research and scientific application, design and survey, construction and installation.

The proposed method of comparing the UHPC options by cost is very approximate, it should be refined and improved based on the results of real design and manufacturing, but at the early stages of UHPC creation it can be used even now.

5. The results of thermal calculations and the cost of various UHPC options
For all working mediums, the net electrical power of UHPC is assumed to be ≈ 1 MW.

For each of the selected options of the WM, a series of calculations were performed with different sets of initial parameters and pressures in the condenser. The purpose of these calculations was to determine the influence of the WM type on the efficiency of the cycle and on the cost of UHPC, as well as the effect of cycle parameters on these indicators.

5.1. Use of hexamethyldisiloxane (MM)
Calculations performed using hexamethyldisiloxane showed that at moderate initial temperatures (up to 175...200 °C) and pressures (0.4...0.7 MPa) the condition of even a small excess pressure during condensation reduces the efficiency of UHPC to 7...9 %, which made it necessary to exclude such options from consideration. When the pressure in the condenser is reduced to 26 kPa, provided at the temperature of 60 °C, and simultaneously, at the initial temperature of 225 °C, the net efficiency increases to 17 %. In this case it was possible to use a five-stage turbine with a direct (without gearbox) drive of the generator with a rotation speed of 3000 rpm. The estimated cost of UHPC in this structure type, 133 million rubles, is quite competitive in relation to other options.

With a further increase in the initial temperature, the dimensions of the heat recovery unit and its cost increase, and the total cost of UHPC, although it continues to decrease, but not so intensely. The results of UHPC calculations using hexamethyldisiloxane are given in table 2.

The drawbacks of the UHPC option using hexamethyldisiloxane include a vacuum condenser and the ability of this WM to ignite, leading to an increase in the safety costs.

| Option                                      | 1  | 2  | 3  | 4  |
|---------------------------------------------|----|----|----|----|
| Initial pressure, MPa                      | 0.440 | 0.440 | 0.719 | 1.106 |
| Initial temperature, °C                    | 175 | 175 | 200 | 225 |
| Pressure in the condenser, kPa              | 125 | 26 | 26 | 26 |
| Condensation temperature, °C                | 108 | 60 | 60 | 60 |
| WM flow rate, kg / s                       | 62 | 27 | 23 | 20 |
| Condensate-feeding pump drive power, kW     | 48 | 26 | 36 | 49 |
| ACC fan drive power, kW                    | 39 | 38 | 32 | 28 |
| Thermal oil flow rate, kg/s                 | 112 | 62 | 46 | 33 |
| Net efficiency, %                          | 7.8 | 14.0 | 16.1 | 17.7 |
| Turbine rotor speed, rpm                    | <1500 | 1500 | 3000 | 3000 |
| Number of stages                            | - | 5 | - | - |
| Pitch diameter, mm                          | - | 1000 | 657 | - |
| Cost of all heat exchangers , million rubles | 105.8 | 46.6 | 41.8 | 40.3 |
| Cost of turbogenerator (TG), million rubles | 44.0 | 44.0 | 44.0 | 44.0 |
| Cost of WM and thermal oil filling, million rubles | 7.7 | 4.0 | 3.4 | 3.0 |
| The total cost of UHPC-1000, million rubles | 233.4 | 140.4 | 132.7 | 130.0 |
5.2. Use of refrigerant R134a
Freons are widely used in refrigeration machines, there are examples of their use in geothermal energy as well. One of the most serious obstacles to the WM application of this class is their ozone-depleting properties. Refrigerant R134a belongs to those WM, whose influence on the ozone destruction is minimal, its use, at least for the time being, is not prohibited by international agreements. At the same time, this WM is non-flammable, non-explosive and non-toxic. For refrigerant R134 a high condensing pressure ~ 1 MPa at a temperature of 40 °C is inherent. In order to ensure an acceptable pressure drop in the turbine, the initial pressure must not be lower than 5 MPa, which complicates the sealing problems and increases the requirements to the strength of the casings and pipelines.

For refrigerant R134 two set of calculations were made with initial parameters in the supercritical region, the results are given in table 3. For the option with the initial temperature 135 °C turbine structure was evaluated.

Table 3. Process parameters for refrigerant R134a.

| Option | | |
|--------|---|---|
| | 1 | 2 |
| Initial pressure, MPa | 4.1 | 5.0 |
| Initial temperature, °C | 120 | 135 |
| Pressure in the condenser, kPa | 40 | 40 |
| Condensation temperature, °C | 1017 | 1017 |
| WM flow rate, kg/s | 74 | 59 |
| Condensate-feeding pump drive power, kW | 330 | 264 |
| ACC fan drive power, kW | 137 | 110 |
| Thermal oil flow rate, kg/s | 103 | 69 |
| Net efficiency, % | 7.7 | 9.4 |
| Turbine rotor speed, rpm | 10500 | 10500 |
| Number of stages | 2 | 1 |
| Pitch diameter, mm | - | 330.0\textsuperscript{a} |
| Cost of all heat exchangers, million rubles | 76.0 | 62.1 |
| Cost of turbogenerator (TG), million rubles | 54.0 | 54.0 |
| Cost of WM and thermal oil filling, million rubles | 16.0 | 12.9 |
| The total cost of UHPC -1000, million rubles | 186.3 | 165.0 |

\textsuperscript{a} An axial-radial turbine

It should be noted that the efficiency of UHPC on the refrigerant R134a turned out to be lower, and the cost is higher than on hexamethyldisiloxane. The power spent to drive the pump and ACC fan, reaches 30 % of the power at the generator terminals. A further increase in the initial parameters, in principle, can increase efficiency slightly and reduce the cost of UHPC, but also, simultaneously, leads to an undesirable increase in the pressure in the closed loop.

In addition, the drawback of this variant of UHPC is the need for a gearbox to drive the generator, which complicates and increases the cost of the structure.

5.3. Use of fluorocarbon C₄F₁₀
The fluorocarbon C₄F₁₀ at a temperature of +40 °C is condensed under a pressure of 425 kPa (table 1). By this parameter it is more attractive than refrigerant R134a. It is non-toxic, safe and non-flammable. However, its cost at the market is one of the highest, which should undoubtedly affect the cost of the entire plant, considering its filling.

The results of calculations for fluorocarbon C₄F₁₀ are shown in table 4. In comparison with UHPC on hexamethyldisiloxane, mass expenditures of UHPC turned out to be 3-4 times higher, and net efficiency is lower. The turbine can be made with a rotation speed of 3000 rpm. UHPC cost with this
Thermophysics is the highest from the considered variants mainly due to the high cost of the WM and has no prospects to be equal with the cost of UHPC, for example, on hexamethyldisiloxane.

5.4. UHPC on FC-72 (PP1)
Fluorocarbon compounds produced under the trademarks FC-72 and PP1 have very similar thermodynamic properties, and therefore the calculation of the cost of UHPC with these WMs differs only in the cost of their filling.

These WM at the temperature of +40 °C condense under the pressure of 56 kPa. This is a vacuum, but less deep than with hexamethyldisiloxane. The FC-72 and PP1 are safe (used for medical purposes), non-flammable, and this promises high performance. The WM is quite expensive, but still their price is 3-5 times less than the price of fluorocarbon C₄F₁₀.

The calculation data for the UHPC options using FC-72 and PP1 are presented in table 5. Here, options 1 and 2 are oriented to the overpressure in the condenser and differ in the initial parameters, and option 3 is designed for the vacuum condenser. For option 3, a flow-through part of a 5-stage turbine with a rotation speed of 1500 rpm (for a rotational speed of 3000 rpm volume flow is too large) was designed in detail.

The cost is calculated based on the price of FC-72 equal to 5811 rubles/kg, and the price of PP1 equal to 3011 rubles/kg (the estimated cost of UHPC with this WM is given in brackets).

As follows from these data when using WM PP1 the cost can be obtained approaching the minimum of the considered options, but it requires a generator with rotation speed of 1500 rpm (serial generator has 3000 rpm).

In addition, for a variant of the minimum cost a vacuum condenser is used, which can also be considered as its disadvantage.

We should also note that with this WM data on its thermodynamic characteristics are more limited than with the others. In particular, there is no data on the speed of sound, which makes it impossible to perform a reliable profiling of the blading.

| Table 4. Process parameters for fluorocarbon C₄F₁₀. |
|--------------------------------------------------|
| Option  | 1    | 2    | 3    |
| Initial pressure, MPa | 1.99 | 1.99 | 2.98 |
| Initial temperature, °C | 110  | 160  | 160  |
| Pressure in the condenser, kPa | 132  | 93   | 83   |
| Condensation temperature, °C | 40   | 40   | 40   |
| WM flow rate, kg/s | 425  | 425  | 425  |
| Condensate-feeding pump drive power, kW | 200  | 141  | 204  |
| ACC fan drive power, kW | 126  | 100  | 87   |
| Thermal oil flow rate, kg/s | 77   | 72   | 64   |
| Net efficiency, % | 8.2  | 10.1 | 11.3 |
| Turbine rotor speed, rpm | 3000 | 3000 | 3000 |
| Number of stages | -    | -    | 3    |
| Pitch diameter, mm | -    | -    | 410  |
| Blade length input/output, mm | -    | -    | 20/65 |
| Cost of all heat exchangers, million rubles | 91.0 | 72.1 | 64.5 |
| Cost of turbogenerator (TG), million rubles | 44.0 | 44.0 | 44.0 |
| Cost of WM and thermal oil filling, million rubles | 153.4 | 108.8 | 96.5 |
| The total cost of UHPC -1000, million rubles | 330.2 | 260.8 | 238.7 |
5.5. **UHPC on n-pentane**

N-pentane is the most accessible at the market, it has relatively low cost. Its condensation in the ACC occurs at a low excess pressure. At the same time, its use is associated with the need to implement complex fire safety measures and accordingly, with additional financial costs. The calculation results for the UHPC on n-pentane with different initial parameters are presented in table 6.

| Table 6. Process parameters for n-pentane. |
|------------------|------------------|------------------|
| Option | 1 | 2 | 3 |
| Initial pressure, MPa | 0.666 | 1.463 | 2.833 |
| Initial temperature, °C | 120 | 160 | 200 |
| Consumption of WM, kg/s | 25 | 16 | 13 |
| Condensation temperature, °C | 41 | 41 | 41 |
| Pressure in the condenser, kPa | 120 | 120 | 120 |
| Power of the condensate-feeding pump, kW | 38 | 61 | 97 |
| Power to fan drive VK, kW | 97 | 64 | 52 |
| Thermal oil consumption, kg/s | 78 | 55 | 25 |
| Net efficiency, % | 10.3 | 14.6 | 17.3 |
| Turbine rotor speed, rpm | 3000 | 3000 | 3000 |
| Number of stages | - | 4-5 | - |
| Average diameter, mm | - | 620 | - |
| Blade length input/output, mm | - | 15/51 | - |
| Cost of all heat exchangers , million rubles | 53.8 | 38.3 | 32.8 |
| Cost of turbogenerator (TG), million rubles | 44.0 | 44.0 | 44.0 |
| Cost of WM and thermal oil filling, million rubles | 3.9 | 2.8 | 2.3 |
| The total cost of UHPC-1000, million rubles | 151.2 | 126.8 | 118.0 |

Figure 3 shows the graphical dependence of the UHPC-1000 cost with different WMs on the initial temperature. The use of n-pentane makes it possible to minimize the cost of UHPC-1000 compared with other WMs, to use a turbo-installation without a reducer and, therefore, is preferable to others of the considered options.
6. Features of the turbine part of UHPC-1000

The properties of many low boiling WM are such that with the expansion their density is reduced much faster than the density of air or water steam (table 7). In this connection the design of the expansion process for the low boiling WMs considered requires large meridian opening of the flowing part, leading to efficiency reduction.

For all WMs density ratio is greater than 2, as for water steam and air it is less than 1.7. The occurrence of even small pressure differences in one stage results because of the low speed of sound to the appearance of essentially supersonic flow regimes and to a decrease in efficiency. This makes it necessary to increase the number of turbine stages.

Optimal loading of stages for almost all WM options is achieved at moderate circumferential speeds of 50...100 m/s, realized in limited diametrical dimensions with the standard for the generator rotor speeds of 3000 rpm or 1500 rpm, except for refrigerant R134a. For the selected option of the working medium – n-pentane with initial parameters of 160 °C and 1.5 MPa and a pressure of 120 kPa in the condenser, the flow part of the axial five-stage turbine has been calculated, as shown in figure 4, has been worked out, considering the possibility of manufacturing by the technology and equipment of OJSC "KTW".

### Table 7. Changes in the parameters of WMs in the isentropic expansion process.

| Type of WM       | Hexamethyl disiloxane | Refrigerant R134a | Fluorine-carbon C₄F₁₀ | FP-72 | n-pentane | Water steam | Air |
|------------------|------------------------|-------------------|-----------------------|-------|-----------|-------------|-----|
| Initial pressure, MPa | 0.44                   | 5                 | 2.98                  | 1.356 | 1.463     | 0.2         | 0.5 |
| Initial temperature, °C | 175                   | 135               | 160                   | 170.0 | 160       | 175         | 160 |
| Speed of sound, m/s | 130.9                  | 133.3             | 87.2                  | -     | 176.2     | 517.4       | 417 |
| Pressure ratio   | 2                      | 2                 | 2                     | 2     | 2         | 2           | 2   |
| The ratio of densities | 2.18                  | 2.19              | 2.353                 | 2.54  | 2.185     | 1.69        | 1.64|
| Isoentropic heat difference | 14.31                 | 14.45             | 7.56                  | 4.95  | 27.53     | 130.4       | 78.3|
Figure 4. Turbine setting on n-pentane.

7. Conclusion
The most acceptable in terms of efficiency, cost of manufacturing and installation of the heat recovery unit is the use of n-pentane, even though it requires the implementation of a complex of fire and explosion safety measures.

The cycle on n-pentane having moderate initial parameters (temperature 160 °C, pressure 1.5 MPa) seems to be optimal both for the implementation of high energy efficiency indicators (efficiency 14…15 %) and for the manufacturing technology of a 4-5-stage turbine based on technologies of OJSC "KTW" with direct drive of an electric generator. The development of the turbine setting showed that the claimed internal efficiency of the turbine 0.8…0.81 is provided liberal.

Closest to n-pentane UHPC cost is hexamethyldisiloxane and PP1, however they are less available at the market, and for implementing acceptable technical and economic parameters with their use a vacuum condenser is required. These working mediums can be considered only as backup.

All of the low boiling WMs from the considered ones in comparison with water steam and air have a reduced speed of sound and higher rates of increase in specific volumes during expansion in the turbine, which compels to increase the number of its stages. At the same time, reduced heat drops at the stages facilitate the task of providing a generator drive with a standard speed of 3000 rpm and in moderate dimensions.

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