Chair of Petroleum and Geothermal Energy Recovery

Master's Thesis

Improving the energy efficiency of the compressor station by generating electricity

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AFFIDAVIT

I declare on oath that I wrote this thesis independently, did not use other than the specified sources and aids, and did not otherwise use any unauthorized aids.

I declare that I have read, understood, and complied with the guidelines of the senate of the Montanuniversität Leoben for "Good Scientific Practice".

Furthermore, I declare that the electronic and printed version of the submitted thesis are identical, both, formally and with regard to content.

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Kurzfassung

Die Steigerung der Energieeffizienz von Kompressorstationen ist eine der wichtigsten Herausforderungen der Industrie, da die Verdichtung von Gas der energieaufwendigste Prozess des Gastransportes ist. Ununterbrochene Energiezufuhr zu den Kompressorstationen ist einer der Schlüsselfaktoren für die Systemintegrität und die Industrie insgesamt. Faktoren wie steigende Strompreise, das Altern der Stromnetze und die steigende Anzahl von Unterbrechungen des Stromnetzes dadurch, die hohen Konstruktionskosten für neue Leitungen und herausfordernde klimatische Bedingungen in neu zu entwickelnden Gasfeldern machen es für die Industrie interessant, Kraftwerke und Stromnetze selbst zu entwickeln.

In dieser Masterarbeit wurde die Kompressorstation Nummer 7 der Firma "Surgutneftegas" als Forschungsobjekt ausgewählt. Das Ziel dieser Arbeit ist es, die vorteilhafteste Energiequelle, sowohl von der technischen als auch von der wirtschaftlichen Seite, zu finden. Zu diesem Zweck wurden die Kapazitäten der Turboexpander, der Gasturbine und des Gas-und-Dampf-Kombikraftwerkes theoretisch berechnet. Die Ergebnisse der thermodynamischen und wirtschaftlichen Berechnungen werden am Ende der Masterarbeit präsentiert

Abstract

Improving the energy efficiency of compressor stations is an urgent problem in the gas industry, as gas compression is the most energy-intensive heat and power-consuming process in the hydrocarbon pipeline transportation. In this regard, it should be noted that uninterrupted power supply to compressor stations is a key factor both for the overall system integrity and for the industry as a whole. Factors such as the growth of power tariffs, ageing of power transmission lines and, consequently, the increase in the number of disconnections due to power line accidents, the high cost of new network construction and difficult climatic conditions in the areas of newly developed fields predetermine the interest in the development of projects to create power plants for own needs.

In this thesis compressor station №7 of the "Surgutneftegas" company was chosen as the object for research. The aim of the work was to determine the most advantageous source of power generation both from the technical and economic sides. For this purpose, the theoretical capacity of turboexpander, gas turbine unit and combined cycle gas turbine unit were calculated. The results of thermodynamic and economic calculations are presented at the end of the thesis.

Table of content

1 Introduction

Oil and gas industry enterprises receive electricity from both the main suppliers of the power grid network and from their own power plants. Improving the energy efficiency of compressor stations is an important issue, since gas compression is the most energy-intensive heat and power process in the main hydrocarbon transport. In this regard, it should be noted that the uninterrupted supply of compressor stations (CS) with electricity is a key factor both for the overall integrity of the system and for the industry as a whole.

The program of switching compressor stations to receive electricity from centralized sources adopted in the last century has shown its inefficiency. Such reasons as the wear and tear of power lines, unexpected outages due to outdated equipment, as well as rising prices for electricity tariffs do not allow to ensure uninterrupted power supply to enterprises. [1]

It is expected that in the future there will be a tendency of increased electricity prices. At the same time, price regulation by authorities will not be able to stop this growth. This is due to the annual rise in the price of energy carriers such as oil, gas, coal and others that occupy a significant share in the fuel balance of power plants. An important factor in the price increase is the aging of the main equipment of the enterprise that requires timely replacement.

The key factor of electricity price increases is the inclusion of federal and regional taxes in the tariffs, as well as fees for upcoming capital repairs. This trend is expected to continue for many years to come. High electricity prices mainly affect the profitability of production enterprises such as «Gazprom» and «Surgutneftegas» as they increase the cost of gas production and transportation. These circumstances confirm the necessity to search for new energy-saving technologies to reduce energy production costs.

The difficulty of finding investments in the country's energy sector and the issues of their insufficient payback, environmental, transport and other problems that prevent the construction of large-capacity power plants aroused great interest to the problem of construction of own electric power sources at gas industry enterprises.

Compressor stations in most countries of the world use electricity generated by captive power plants for their own needs. The greatest demand is for installations for combined production of electric and thermal energy in which both gas turbine and steam turbine engines with a capacity of up to 30 MW are used as power drives for electricity generators.

Currently, there is a wide range of domestic and foreign facilities for equipping captive power plants. These electric units are available in a variety of configurations and sizes. At the same time, gas-fueled units are widely used, in contrast to devices that run on liquid fuel. [2]

There are a number of factors that increase interest in the construction of power plants for own needs in oil and gas companies:

1. Most compressor stations of gas main pipelines located in Western Siberia and the North of Russia require electric power supply for many thousands of kilometers via

high-voltage transmission lines, which in turn increases the cost of produced hydrocarbons.

2. High requirements for reliability and safety of power supply to industrial gas production, transport and processing facilities.
3. Most equipment in compressor stations requires repair and reconstruction due to aging and other factors.

The latter factor has been of particular importance in recent years. The Russian gas industry is going through the period of compressor station modification due to changes in the volume dynamics of transported gas. This factor determines the relevance of the topic related to the increase in reliability of power supply of compressor stations.

An alternative and reliable solution to the problem in these conditions is to create local power supply systems with a minimum length of outgoing power lines from mobile block power plants equipped with electric units with a piston or gas turbine drive. It is suggested to compare the efficiency of 3 different units such as turboexpanders, gas turbine units and combined cycle gas turbine units. On this basis, it is proposed to calculate the efficiency of power plants for own needs by comparing the costs of purchasing electricity on the side with the costs of its own production.

2 Power supply systems for compressor stations of main gas pipelines

2.1 Reliability of compressor station units

For compressor stations, the key factors in the choice of power supply schemes are the territory where the facility is located and its climate conditions. The main sources of power supply are power lines from the power system, gas turbine power plants running on gas and diesel fuel and power plants for own needs. [3]

It is worth saying that most of the damages in the electrical networks are caused by bad weather conditions. It is therefore impossible to predict exactly when and where disturbances occur. Distribution stations and electrical installations that transmit electricity are the most frequently affected. This leads to disruption of compressor stations and in some cases to their complete shutdown. [4]

All interruptions of the centralized power supply lead to high financial costs in the millions of dollars. These amounts are made up of a variety of factors:

1. Downtime of the power supply equipment
2. The ambient temperature
3. Cost of energy prices, etc.

Due to the unpredictability of power interruptions, they are considered to be the most severe and difficult factors to be eliminated. If the power transmission is interrupted, consumers lose the ability to operate normally, resulting in equipment downtime. These sudden interruptions lead to disruption of technological processes and in turn, leads to certain economic damage.

In case of power supply interruptions to the operating equipment, an emergency mode is created for the whole system of the compressor station. Various alarms and automatic control protection systems are required to prevent accidents and minimize failures in the power distribution system. These measures help to prevent failure of main and auxiliary equipment due to power line surges.

At the design stage of compressor stations, the fact that the reason for stoppages of gas compressor units is the unevenness of power supply is often not taken into account. Taking into account the complexity of process equipment and the peculiarities of power supply systems, it is worth noting that a number of important factors affects the operation of the system as a whole: [5]

1. Several independent redundant power supplies shall be available in the system for maximum power reliability. In the event of an emergency, one of the power supplies will take over the entire power supply while others, capable of running both parallel and in series, will remain in reserve.
2. For gas-turbine compressor stations it is necessary to provide for the installation of an emergency source with automatic disconnection of consumers with the help of devices

for automatic under-frequency load shedding. In case of loss of alternating voltage, the power supply for control and automation devices shall be provided by a battery.

3. The electrical scheme of the compressor station must be flexible and provide for both the possibility of expanding and connecting external input from the power system, as well as various operational, repair and emergency modes. [6]

To ensure reliable operation of compressor stations, gas turbine power plants, distribution power substations, main gas pipelines, as well as drives of compressors themselves, protective devices shall be provided at technological facilities to detect failures that lead to unstable operation. Such disruptions include all kinds of damage and destruction of equipment parts, including sudden voltage surges in power transmission lines.

2.2 Types of power supply equipment

Uninterrupted work to provide compressor stations of gas main pipelines with electricity is possible in case of the following:

1. The power supply must be provided from three independent mutually redundant power sources, which ensures the highest reliability of the power supply
2. Reserve sources are used – for example captive power plants, capable to work in case of an accident up to several days until failures in the system are eliminated
3. Emergency source - power stations with diesel drives, which provide voltage recovery in 1 minute
4. As the source of temporary power supply can be accumulator batteries, charged either from a compressor station or from non-renewable energy sources, ensuring stable operation of all equipment under emergency conditions in the power supply system
5. Captive power plants, where turboexpanders, gas turbines and even combined-cycle plants can be used as a source of electricity production, if climatic conditions in the region are not severe [7]

Possible disruptions and reasons for breakdowns in the operation of compressor stations of gas main pipelines include the following:

1. Scheduled outages of power supply via independent power supplies
2. Voltage drop and surge for a long period of time
3. Unexpected interruptions of power supply to the compressor station due to transition from one established mode of electrical installation to another [8]
4. Sudden shot-term outages of the power supply line or captive power plants

On the basis of the operation experience gained, it can be stated that different types of voltage deviations from the set values do not bring any significant harm to the equipment operation mode. The use of independent power supply sources for a compressor station becomes necessary in case of forthcoming repair measures at one of the main energy sources. Even in this case, the system reliability does not meet the established norms, which leads to short stops of the operating equipment. When the main voltage drops, power loss increases and, as a result, the use of fuel to compensate and maintain normal operation increases. In addition, the performance of the engines of their own need’s drops, which may lead to lower power generation. [9]

In power supply systems, failures of main and auxiliary equipment are classified as sudden and gradual. The latter, in the case of a compressor station, are most often caused by abrasive wear and tear of moving parts, as well as ageing of components. Sudden failures are explained by the influence of natural, climatic and human factors.

In cases of sudden, prolonged shutdowns of supply lines or power plant shutdowns of own needs, as a rule, there are shutdowns of compressor stations and the conventional operation mode of the entire gas transmission system is violated. Prevention of such cases is the most important task in the design and operation of power plants, networks and electrical installations.

The main causes of power supply system failures at the compressor station are wear and tear of power equipment, instrumentation and all automatics. To improve energy efficiency and reliability of compressor stations of the main pipeline one of the best ways is to use power stations of own needs. Due to severe temperature fluctuations in Russia, it is difficult to choose a universal electric generating unit that would suit all enterprises. Ways of solving this problem requires a detailed study.

2.3 Energy consumption at the compressor station

A compressor station of the main pipeline consists of different structures and equipment for increasing gas pressure during its production, transportation and storage. The compressor station itself consists of gas treatment units, compressors and gas air cooling units. The compressor station equipment operation is provided by technological pipelines with shut-off and regulating valves as well as oil and power supply systems.

Shortened service life, wear and tear and breakage is a consequence of the disruption in the power supply to electric drives of pumps of oil supply systems, turbine oil air cooling units and water automatic cooling units. There are a number of electrical devices in the work place, including fire alarms, ventilation and lighting, the interruption of which, in the event of an emergency, can cause harm to human health. This equipment is classified in Category 1 according to the rules of electrical installations of the Russian Federation.

In the first category of reliability of power supply for electric receivers, there is a special subgroup for uninterrupted operations for which it is necessary for an accident-free stop of equipment in order to prevent explosions and fires. Such consumers include instrumentation and control systems, emergency lighting, communication units and fire pumps.

As it can be seen from Figure 1, reliability of operation of almost all CS systems depends on the reliability of their electrical equipment and reliability of power supply sources. [10]

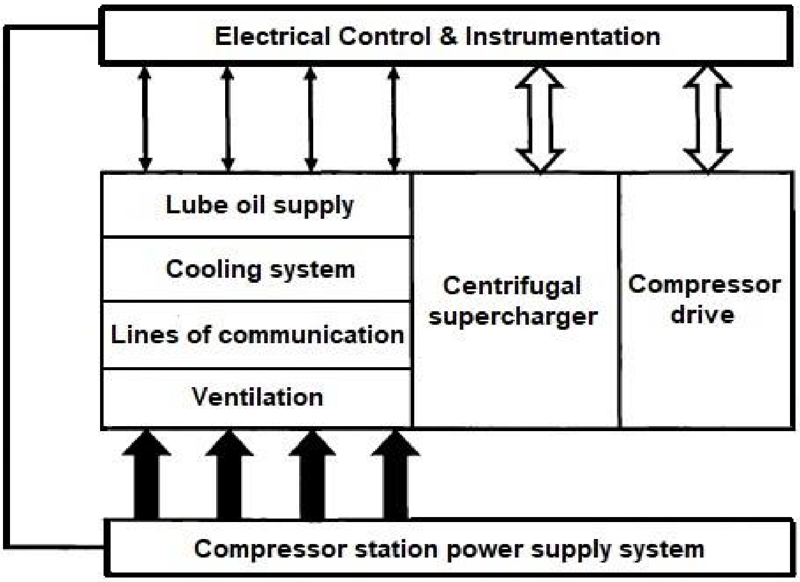

Figure 1 - Structure of systems providing technological process of compressor stations with gas turbine drive of centrifugal supercharger[1]

Electricity consumers, whose interruption leads to a decrease in productivity of the whole compressor station, but not to its complete shutdown, are classified as category 2. Electric receivers of this category in operating modes should be provided with electricity from two independent mutually redundant power sources. In case of power supply failure from one of the power supply sources for electric receivers of the second category, power supply interruptions within 30 minutes are allowed.

Electric receivers of the third category work from one power source and allow a power break for the duration of repair. These include mechanical repair shops, garages, material and equipment warehouses and auxiliary buildings. [11]

2.4 Captive power plants

At the moment the majority of oil and gas companies are located in the northern part of Russia, which climate is characterized as severe. This factor complicates the supply of electricity to gas compression and transportation facilities, and therefore hundreds of kilometers of power transmission lines have to be laid in difficult conditions. As a result, there is irreversible loss of electricity in the 110 kV highways, the major part of which is spent on corona discharges. They are caused by humidity and icing on high-voltage lines and conventional power lines. [12] Considering the latter factor, the cost of electricity to melt the ice should be taken into account.

For this problem a rather reliable and technically and economically advantageous solution is the installation of power plants of own needs directly on the territory of the objects. Their use

[1] A.N. Nazarov. “Increase of reliability and efficiency of compressor station operation due to improvement of power supply systems,” 2007, Moscow. p.25

minimizes the length of transmission lines and reduces the risk of emergency situations due to sudden interruptions in power supply. Free piston linear generators, gas turbine units and steam power machines can be used to generate electricity for power plants for their own needs.

The main advantage of using autonomous energy sources is independence from monopolistic organizations that dictate the price for electricity. Another advantage of using power plants for their own needs is obtaining an additional source of income from the sale of manufactured products. Therefore, in order to determine the efficiency of implementation of new technologies it is necessary to make appropriate calculations including comparison of the costs of own electricity production and purchases from third parties.
The following technical requirements are applied to power plants of their own needs: [13]

1. Power plants for own needs should be located in close proximity to the reserved object on the territory of the compressor station and comply with the current regulatory documents and norms
2. The territory of the power plant should have a fire water supply system, lightning protection, outdoor and security lighting network with the use of technical means of protection.
3. The power plant shall have, in addition to the local control panel, a remote-control panel and a cable (up to 50 m long) connecting the remote control to the power plant control panel.
4. Power plant weight has to be no more than 25-30 tonnes
5. It is preferable to have two start-up systems in a power plant: pneumatic and electric.
6. The productivity of fuel injection pumps should exceed the fuel consumption of power plants of their own needs at full load.
7. In case of operating failure or in case of deviation of basic parameters from the normalized ones, the protection system should stop the generator operation.

For simplicity and convenience of maintenance the power plants of their own needs are carried out in container-module configuration. This configuration allows for the installation and commissioning of an autonomous power supply source in a short time.

The foundation should be developed taking into account the specifics of soil. In addition, the foundation of the engine and the electric generator should be considered as well.

To increase the efficiency of power generation at power plants of their own needs, it is necessary to use technologies that allow to utilize the heat of exhaust gases. In this way it is possible to supply hot water to nearby settlements and administrative buildings of the compressor station. Due to the simplicity of construction and use, block power plants with gas turbine engines are preferable. The advantages are as follows:

1. Several times smaller in size and weight compared to other types of engines.
2. Because of their mobility, these power plants can be easily transported from one place to another.

3. Quick replacement and easy repair of broken equipment in the conditions of the operating compressor station.
4. The level of noise and vibration remains within the permissible ranges, which corresponds to the regulatory requirements
5. High reliability of gas turbine engines allows their operation in both the harshest and hottest climatic conditions.
6. Low oil consumption and loss due to evaporation, leakage through flange connections, etc. On average, a 5 MW gas turbine unit consumes 1.3 tons of oil per year. [14]
7. Excellent performance and automation
8. Effective diagnostic techniques for gas turbines
9. The ability to use different types of fuel allows gas turbines to use oil distillation products, natural and liquefied gas and diesel fuel.

Free piston linear generator - a linear combustion engine without connecting rods, in which the movement of the piston is determined not by mechanical bonds, but by the ratio of forces of expanding gases to the load. Figure 2 represents different types of this energy converting machine.

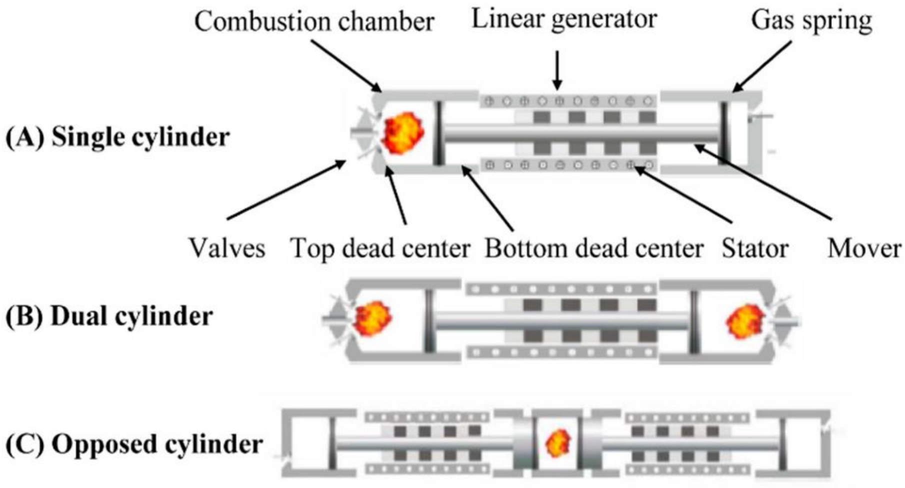

Figure 2 - Illustration of free piston linear generator[2]

By comparing a gas turbine engine and a free piston linear generator, the following advantages of the latter can be highlighted:

1. Increased rigidity and mechanical reliability of the motor design
2. Lower friction in the cylinder piston group increases the mechanical efficiency of the motor and its service life.
3. Small amount of harmful emissions to the atmosphere

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[2] Wang, X.; Chen, F.; Zhu, R.; Yang, G.; Zhang, C. “A Review of the Design and Control of Free-Piston Linear Generator,” 2018, p.2

4. High efficiency due to optimal combustion modes

Currently, companies producing block power plants of their own needs provide a range of capacities up to 50 MW, which allows companies to supply electricity not only to their customers but also to third parties.

A high degree of automation of production processes, including sequential start-up of units, alarm of normal operation mode failure and transition to emergency operation mode allow power plants of their own needs to work with a minimum number of operators and machinists of process equipment.

2.5 Evaluation of operational reliability of power supply to compressor stations

To improve the reliability of gas pumping units, oil and gas corporations collect and process information on the total number of failures associated with: [6, p. 37]

1. Wear and tear of movable parts of the compressor module
2. Stops due to sudden interruption of power supply
3. Outages due to problems with instrumentation and oil supply systems
4. Number of shutdowns due to incorrect technical operation of the systems and staff actions

The number of forced stops of all gas turbine units per year is about 30 %. The reason for this, as many years of research show, is interrupted power supply due to breakdowns in power generating systems. [15]

However, due to breakdowns in power supply to gas-pumping units, the total downtime is about 3 hours, which is 1.5% in percentage terms. This also includes the time to repair gas compressor units, which is the most significant part of all downtime. For this period, in order to continue normal operation of the entire plant, reserve equipment is used.

Based on the above, it can be concluded that the total downtime of the gas pumping unit due to power outage is comparable to the downtime due to a serious failure of one of the parts.
The share and causes of forced stops in "Surgutneftegas" of gas turbine units GT-750-6 and GT-10, as well as GT-10I, GT-25I, power supply of which is carried out from own generators, are presented in Table 1. It can be stated that the number of forced outages of Russian and imported gas compressor units is approximately the same. The main difference will be the reasons that caused these interruptions. Most often in Russian equipment breakdowns are caused by mechanical parts, while in imported equipment it is electrical equipment and instrumentation and control systems. If we talk about the reliability of the power supply system of the compressor station as a whole, there are a number of factors such as scheduled inspection and repair of equipment, verification of all sensors, etc., which maintain all systems in working condition.

Table 1 - Share and reasons for forced stoppages of gas turbine units[3]

Reasons for stoppages	**The share of forced stops of various gas turbine units, %**		
	GT-750-6	**GT-10**	**GTK- 10I, GTK-25I**
Destruction of components and mechanical faults	14,2	16,1	5,6
Disruptions in power supply	40,1	32,5	7,1
Disruptions in instrumentation and automation	10,6	12,1	59,7
Failures of the lube oil supply system	15,1	19,3	11,3
Failures of other installed systems	5,99	14,8	12,2
Other and undetermined reasons	14,01	5,2	4,1

The abbreviation GT-750-6 means, that the gas turbine has a power of 6 megawatts and the temperature of gases after combustion chamber is equal to $750℃$. In case of GT-25I it has a meaning that gas turbine power is 25 megawatts and this unit is imported.

Due to the extremely severe climate, power transmission lines in the northern regions of Russia are susceptible to more severe damage due to icing. It also leads to serious disruption of the entire system with significant power shortages and mass blackouts of high voltage lines.

The following 4 reasons usually lead to an accident:

1. Tree falls on overhead power lines
2. Accidents caused by third parties
3. Wear and tear of power lines
4. Influence of wind and ice on wires

Before an industrial facility under construction is put into operation and begins to receive electricity from the main transmission lines, it will operate for the first few months from the power plants of its own needs. It happens that the compressor stations subsequently leave

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[3] A.N. Nazarov, "Increase of reliability and efficiency of compressor station operation due to improvement of power supply systems," 2007, Moscow, p.34

captive power plants as the main power source, and the power transmission line is used as a reserve.

Reliability and cost studies should be carried out before choosing which of the compressor station's power supply sources is kept as the main one. The efficiency of a particular source should be justified on the technical and economic side. In the past, companies preferred to receive electricity via power lines because the technology of using power plants for their own needs was not so common. Due to scientific and technological progress, autonomous power plants have proven to be reliable and have become more frequently used at production facilities.

In recent years, due to the sanctions imposed in Russia, imported goods have been replaced by domestic ones. Equipment of the oil and gas industry are also included in this list. [16] Therefore, when choosing equipment with high reliability one should pay attention to what details it is assembled from and in what country it is produced. These features also include the small series of parts in operation. However, the main factor for power plants for their own needs is an uninterrupted supply of electricity throughout the life of the equipment.

The main factor affecting the reliability of the captive power plants is to determine the required capacity of the entire compressor station, the number of electric generators and the choice of the main generating equipment. It is necessary to determine the number of generators at the captive power plants, and if required, increasing their number during operation depending on changes in the technical condition of units and auxiliary equipment.

More often, external power transmission lines are used as an electric power source at the first category objects. To increase the reliability of the facility operation, the possibility to implement power plants of their own needs in case of emergency mode should be provided. It is also possible to operate captive power plants as the main source of electricity, and the transmission line to be switched on in case of accidents.

In practice there are examples when an object of the first category has only one external source which is a transmission line. In this case, the second source should be a power plant of own needs, which is operated in a stationary mode, and the transmission line can be used to transfer the excess electricity into the system and is included in the operation in case of power plant failures.

As an electric receiver of the first category consider possible variants of schemes of power supply system of a compressor station: [17]

1. The first scheme of the energy supply system includes two independent inputs from the external energy system (power lines). Gas-diesel generators and storage batteries are used as emergency sources.
2. The second power supply scheme consists of two independent systems. The first one is an input of the external power system via a power line, and the second one is a power station of own needs. Accumulator batteries and gas-diesel-generator units are

used as an emergency source. The main source of power supply is an external source, the power line, while the reserve source is a captive power plant.

3. The third power supply scheme consists of two independent systems. The first one is a captive power plants, and the second one is an input of the external power system via a power line. Accumulator batteries and gas-diesel-generator units are used as an emergency source. The main source of power supply is a captive power plant, while the backup source is the power transmission line.
4. The fourth power supply scheme consists of two independent systems. The first one is an input of the external power system via a power line, and the second one is a captive power plant. The external power supply source and the captive power plants operate in parallel. The advantage of this scheme is the fact that in case of emergency stops or repair of power plants of their own needs, the load is redistributed to another source of power supply. Due to the simultaneous operation of two power supply stations, the overall reliability of the system, as well as economic indicators, increases due to the fact that there is no longer a need to have backup generating sets.
5. In the fifth power supply scheme, the power plant of its own needs is the only one, the main source of power supply, and as emergency sources are used gas-diesel generators and storage batteries.

Transition from the first to the fourth power supply option for the compressor station involves the construction of captive power plants (CPP). The external power supply (power lines) and CPP run in parallel. The CPP capacity is selected so that it covers the maximum load of the compressor station's power consumption, operating at nominal mode. Under these conditions, with a power consumption capacity of more than 2500 kW at captive power plants from an economic point of view, it is advisable to use power units with gas turbine drive. Possible options for using captive power plants to supply the compressor station include such installations as:

1. Turboexpanders
2. Gas turbine units
3. Combined cycle gas turbine units

Successful use of these units has found wide application in oil and gas industry. The technical and economic comparison of alternative ways of generating electricity is an important issue that requires more detailed research.

2.5.1 Turbogenerator units

A turboexpander, also referred to as a turbo-expander or an expansion turbine, is a centrifugal or axial-flow turbine, through which a high-pressure gas is expanded to produce work that is often used to drive a compressor or generator. Installations that generate mechanical energy through the expansion of natural gas and drive an electric generator are called turboexpander recovery units or turbo generator units (TGU). [18] Figure 3 shows the structure of a turboexpander unit.

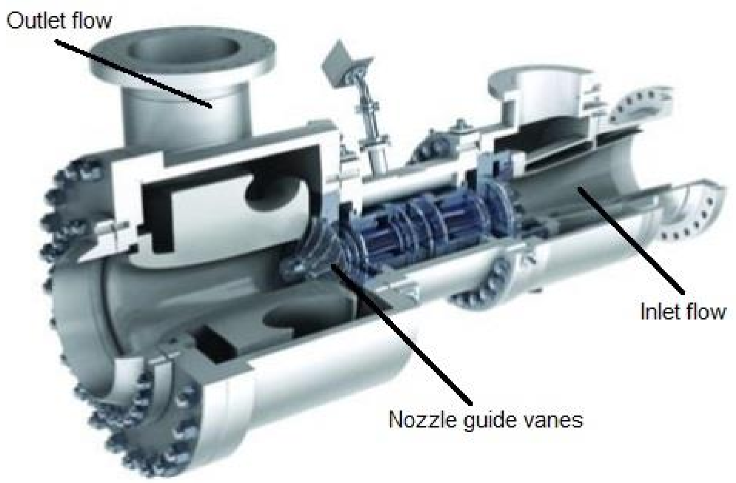

Figure 3 - Turbo generator cross section[4]

TGU consist of inlet and outlet flow parts, nozzle guide vanes (NGV), alternator and output connections. Turbo generator units use excess gas pressure to generate electricity at gas distribution stations or gas distribution centers (GDC), while turboexpander units are used in various schemes for cooling, cleaning, drying and separation of low-boiling hydrocarbons from natural gas. In the gas industry, turboexpanders are used for:

1. Starting up the gas turbine unit of the gas pumping unit, as well as rotating its rotor during the shutdown to cool it. In this case the turbo expander operates on the transported gas.
2. Cooling of natural gas in its liquefaction units.
3. Natural gas cooling in units at field preparation for transportation through the pipeline system.
4. High pressure compressor drive for gas supply to underground storage facilities.
5. Electric power generation at gas distribution stations of the natural gas transportation system to its consumers using gas pressure differentials between high- and low-pressure pipelines.

Figure 4 shows an electric power generation system that uses a heat source, a cooling medium (air, water or other), a circulating working fluid and a turboexpander. The system can accommodate a wide variety of heat sources such as:

1. Geothermal hot water
2. Exhaust gas from internal combustion engines burning a variety of fuels (natural gas, landfill gas, diesel oil, or fuel oil)

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[4] "Softinway.com" [Online]. Available: https://blog.softinway.com/driving-turboexpander-technology/ [Accessed April 2020].

3. A variety of waste heat sources (in the form of either gas or liquid)

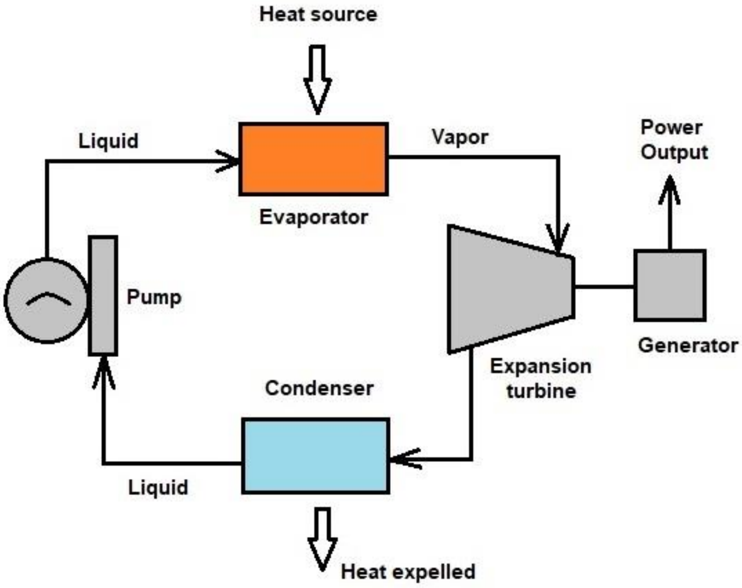

Figure 4 - Schematic diagram of power generation using a turboexpander

The circulating working fluid (usually an organic compound for organic Rankine cycle) is pumped to a high pressure and then vaporized in the evaporator by heat exchange with the available heat source. The resulting high-pressure vapor flows to the turboexpander, where it undergoes an isentropic expansion and exits as a vapor–liquid mixture, which is then condensed into a liquid by heat exchange with the available cooling medium. The condensed liquid is pumped back to the evaporator to complete the cycle.

The system in the figure implements a Rankine cycle as it is used in fossil-fuel power plants, where water is the working fluid and the heat source is derived from the combustion of natural gas, fuel oil or coal used to generate high-pressure steam. The high-pressure steam then undergoes an isentropic expansion in a conventional steam turbine. The steam turbine exhaust steam is next condensed into liquid water, which is then pumped back to steam generator to complete the cycle.

When an organic working fluid such as R-134a is used in the Rankine cycle, the cycle is sometimes referred to as an organic Rankine cycle. At present, the issues of creating possible projects with the use of turboexpanders and aerodynamic regularities used in their design and operation are already well studied, but there are still many issues that require additional research. [18]

Due to the individuality of physical parameters, component compositions and consumption of natural and associated gas of each field, it is often necessary to develop process diagrams, introducing new correction factors, creating new methods of calculation of losses. In other words, it is necessary to perform additional research to obtain reliable information about the unit's operation, especially if the turbine operates in the area of condensation of some natural gas fractions. In recent years, with the growth of computing capabilities, the degree of development of turbine expander flow part designs using numerical experiments has significantly increased. [19]

2.5.2 Gas turbine power plant

Gas turbine power plants (GTPP) are heat machines in which the thermal energy of a gaseous working body is converted into mechanical energy. The gas turbine itself, the combustion chamber and the compressor are the main components. There is a set of auxiliary systems that are combined with each other, which serves directly to ensure the work and control in the installation. To generate 10 – 20 kW of power, just one unit is needed. [20]

Industrial gas turbines vary in scale from small mobile plants to massive, complicated installations weighing over 100 tons in purpose-built buildings. When the gas turbine is used for shaft control only, the thermal efficiency is roughly 30%. Purchasing electricity, however, can be cheaper than producing it. [21]

Another important benefit is their ability to be turned on and off in a few minutes, providing power during peak, or unplanned, demand. Because single-power plants are less powerful than combined-power plants, they are typically used as peak power plants, which run from several hours a day to a few tens of hours a year, depending on the region's energy demand and generating capacity. In areas with a shortage of base-load and load following power plant capacity or with low fuel costs, a gas turbine powerplant may regularly operate most hours of the day. A large single-cycle gas turbine typically produces 100 to 400 MW of electric power and has 35–45% thermal efficiency. [22]

The design of the gas turbine unit includes two main parts combined into one housing: a gas generator and a power turbine, shown in Figure 5.

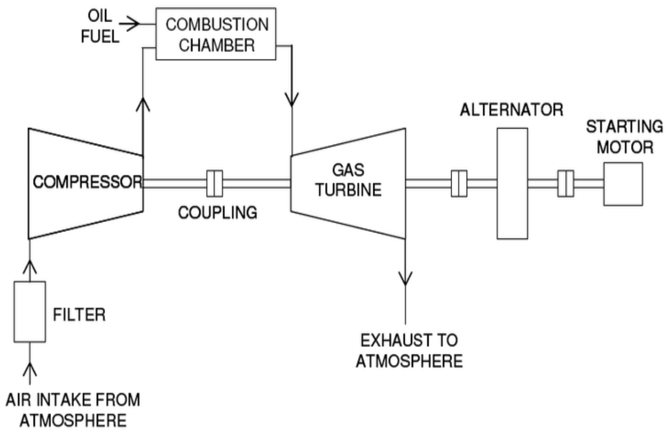

Figure 5 - Schematic diagram of a simple GTPP[5]

[5] "Researchgate.net" [Online]. Available: https://www.researchgate.net/figure/Schematic-diagram-of-a-Simple-Gas-Turbine-Power-plant_fig2_308887222 . [Accessed February 2020].

The principle of operation is based on the fact that air enters the compressor. Then it is compressed at high pressure and sent to the combustion chamber, in the compressed state through the air heater and air distribution valve. Simultaneously with the air, the gas enters the combustion chamber through nozzles, which is burned in an air stream. As the gas and air burn, which forms a flow of glowing gases, this flow begins to act at great speed on the gas turbine blades and they begin to rotate. Thermal energy is converted into mechanical energy, which causes the turbine shaft to rotate. The turbine shaft acts on the compressor and the electric generator, and from the generator terminals, the electrical energy is sent to the consumer network through the transformer.

2.5.3 Combined cycle gas turbine

A combined-cycle plant is an electric generating station that serves for the production of both electric and thermal energy. Thermal energy is used for additional electricity production. The main difference from gas turbine installations is the increased efficiency. The operation of a combined-cycle gas plant is made possible by using either natural gas or petroleum products as the source fuel. The principle of operation of the most economical and widespread classical scheme is as follows. The device consists of two blocks: a gas turbine and steam power units. In a gas turbine installation, the turbine is rotated by the gaseous products of fuel combustion. Passing through a gas turbine, the combustion products give it only a part of their energy and at the exit from the gas turbine still have a high temperature. From the outlet of the gas turbine, the combustion products enter the steam power plant, into the recovery boiler, where water is heated and the resulting water vapor is formed. The scheme of operation of a combined-cycle gas plant is shown in Figure 6.

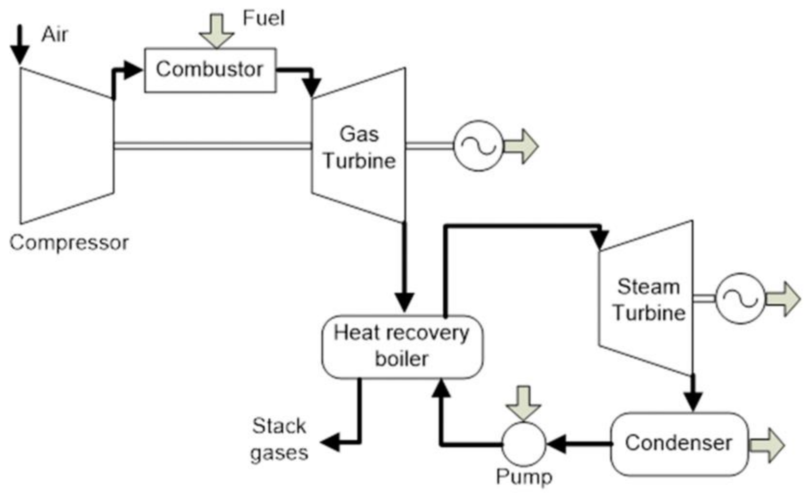

Figure 6 - Working principle of a combined cycle power plant[6]

[6] "Electricala2z.com" [Online]. Available: https://electricala2z.com/electrical-power/combined-heat-power-plants-steam-gas-micro-turbine-fuel-cell/attachment/combined-cycle-gas-turbine. [Accessed May 2020].

In the first gas turbine cycle, the efficiency rarely exceeds 38 %. The products of combustion that have been used in the gas turbine plant, but still remain at a high temperature, are fed into a so-called recovery boiler. At this point, steam heated to a temperature of 500°C and a pressure of 80 bar sufficient for the operation of the steam turbine, to which another generator is connected. The second steam cycle uses about 20 % of the energy of the burned fuel. In total, the efficiency of the entire plant is about 58 %. Steam power units are well developed. They are reliable and durable. Their unit capacity reaches 800-1200 MW, and the efficiency factor, which is the ratio of produced electricity to the calorific value of used fuel, is up to 48 %. An increase in efficiency when combining steam turbine and gas turbine units is obtained by reducing the total consumption of fuel gases. There are three main types of steam-gas plants: [23]

1. A gas turbine unit operating on a steam-gas mixture, which is formed by injecting water (or steam) into the gas path in front of the turbine
2. A high-pressure steam generator
3. A conventional steam generator operating on hot gases discharged into it from a gas turbine.

2.6 Description of compressor station № 7 at “Fedorovskoye oilfield” of Surgutneftegas Company

"Surgutneftegas is one of the largest vertically integrated oil companies in Russia, uniting exploration, production, oil and gas processing and household enterprises. Each of the enterprises, being a part of a single technological chain, forms the full cycle of the company's production operations, in particular, the management of infield collection and utilization of associated gas.

The company pays great attention to the extraction and utilization of associated petroleum gas. For this purpose, a large number of facilities for associated gas separation, transportation and utilization are in operation at Surgutneftegas fields. One such facility is a compressor station №7 of the “Fedorovskoye” oilfield.

The main purpose of a compressor station is to compress air, natural or associated petroleum gas, nitrogen and oxygen. As it is quite difficult to integrate the compressor into the production line, manufacturers began to combine the compressor and its auxiliary equipment in a special station, which can operate 24 hours a day. At present, such compressor stations are an integral part of the operation of production lines, which are actively used in many industries today.

Compressor station №7 is designed for compression, purification and drying of associated petroleum gas of the first stage of separation of booster pump stations of the Fedorovskoye oilfield in order to supply it to the system of high-pressure gas pipelines for transportation to the state district power plant through reduction units. In Figure 7 one of the compressor units is shown.

Figure 7 - Compressor unit at compressor station №7

The compressor station includes:

1. The system of pre-treatment of gas coming to the compressor unit reception
2. Two compressor lines with gas compressor units, a gas turbine engine and a centrifugal compressor
3. System of gas drying from droplet liquid by absorption method
4. Absorber regeneration system
5. Condensate collection and pumping system at booster pump station
6. Gas turbine engine lubrication system
7. A system for the lubrication and oil sealing of compressor units
8. Flare system
9. A system for preparing and supplying the engine with fuel and starting gas
10. Heating, water and sewage systems
11. The system of gas supply and discharge lines
12. Energy supply system
13. Fire alarm and extinguishing system
14. System of automatic control of technological process

2.6.1 Description of technological process of a compressor station

In contrast to natural gas, the component composition of associated petroleum gas may vary greatly from field to field. Moreover, even in the same oil field, the component composition of associated petroleum gas will vary over time.

The component composition of associated petroleum gas for compressor station No. 7 is shown in Table 2.

Table 2 - Associated petroleum gas component composition and physical properties under normal conditions

Component composition		**Volume ratio %**
Element	**Name**	
CH_4	methane	95,04
C_2H_6	ethane	0,24
C_3H_8	propane	0,69
i-C_4H_{10}	iso butane	1,36
n-C_4H_{10}	normal butane	0,51
i-C_5H_{12}	iso pentane	0,79
C_6H_{14} + higher	hexanes	0,25
CO_2	carbon dioxide	0,13
H_2S	hydrogen sulfide	0,02
Density	kg/m³	0,7335
Humidity	%	100
Calorific heat value	kcal/m³	10325
Droplet liquid content	g/m³	up to 5
Mechanical impurities g/m3	g/m³	0,1

Associated petroleum gas through the pipeline in the volume of 205,000 m^3/hand pressure of 0.4MPa arrives at the site of receiving separators for cleaning from mechanical impurities and dripping liquids. The purified gas is directed to two centrifugal compressors, made in block design.

Gas in the low-pressure compressor is compressed to a pressure of 1.9 - 2.2 MPa, the gas temperature at the outlet of the first stage is 80-90 °C. Gas after first stage of the compressor enters the air-cooling unit via a pipeline. In this pipeline, the gas passing through the tubes is cooled by the air flow supplied by fans to a temperature of 30 - 50 °C.

In a high-pressure compressor, the gas is compressed to a pressure of 5.5-6.6 MPa and with a temperature of 110°C the air-cooling unit it enters the pipeline. In this pipeline, the gas passes through finned tubes and is cooled by the air flow supplied by the fans to the temperature of 40 - 50°C. Gas after centrifugal compressors made in block design with a maximum pressure of 6 MPa, having passed two cooling stages on the technological overpass is supplied to the gas drying plant. To supply fuel gas with parameters required for normal operation of the gas turbine engine, the fuel gas treatment unit is provided. Fuel gas treatment units (FGTU) are designed for high pressure gas reduction, after treatment, heating and uninterrupted fuel gas supply to the gas turbine engine. Fuel gas treatment unit can provide fuel gas for simultaneous operation of two gas turbine units and made in block design. There is a 10 m³ receiver installed at the FGTU site to provide fuel gas reserve and maintain fuel gas pressure at least 2.5 MPa during the start of the next unit.

The receiver is equipped with an external heater and safety valve system. Gas from the receiver's safety valves, in case of an accident, is discharged into the gas pipeline to the emergency gas flame.

Fuel gas, passing through the receiver or passing the receiver, is fed to the fuel gas filters of gas turbine units. The starting gas after the control unit with output parameters necessary to start the gas turbine unit is fed through the pipeline to the turbo-compressor unit. Technological scheme of the compressor station №7 is shown in Figure 8

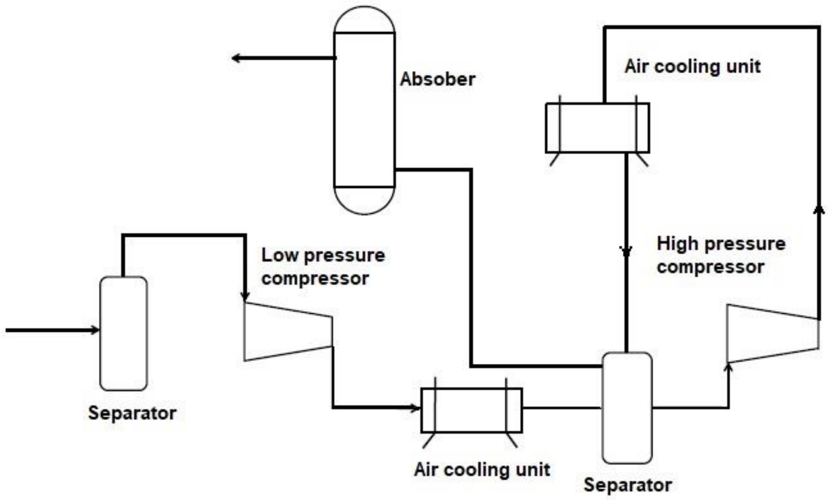

Figure 8 - Technological scheme of the compressor station №7

2.6.2 Description of gas drying and triethylene glycol regeneration scheme

Gas after centrifugal compressors and air cooling system with pressure of 6.6 MPa by the technological overpass is supplied to the gas drying and triethylene glycol (TEG) regeneration unit. A gas drying system and a TEG regeneration unit consists of:

1. Column type absorber,
2. Regeneration block,
3. TEG pumps,
4. Filters,
5. Heat exchangers,
6. Degasser,
7. Regenerator overhead accumulator - designed for drying gas from moisture by absorption method

Gas with a maximum volume of 260,000 m^3/h enters the absorber and its lower separation section to separate the droplet liquid and mechanical impurities. The gas then passes through a plate nozzle where it comes into contact with TEG. The dried gas is brought up to the separation section where the carried away glycol is separated from the gas. To cool the dried gas, air-cooled gas end units are provided before delivery to the customer.

The saturated TEG is collected on a blind plate of the absorber and, under the control of the level regulator, is discharged into the degasser where the gas dissolved in the glycol is released. The degassed TEG enters the filter and magnetic treatment unit. The fine filter retains particles of mechanical impurities larger than 20 μm. The magnetic treatment unit converts stiffness salts in the solution into a state that prevents their deposition on heat-exchange tubes at subsequent heating.

From the TEG filter unit it is fed to the desorber and then to the buffer tank heat exchanger. The heated TEG enters the desorber, passes through the regular nozzle and enters the evaporator. To bring the TEG to the required concentration, the TEG is transferred from the evaporator to the heat exchanger through the steam column. The TEG is discharged through the regular nozzle of the steam column, where desorption gas is fed. Dry gas is used as desorption gas. The desorption gas is heated in the chimney coils before being fed into the steam column. Regenerated TEG from the buffer tank of the regenerator enters the TEG pump unit for the dehumidifying unit through an air-cooling unit.

3 Methodology for calculating the capacity of electric generating units

3.1 Thermodynamic calculation of the turboexpander

Distinctive feature of turboexpander units is that the gas in front of the turboexpander must be heated such that the gas temperature at the outlet of the turboexpander is not lower than 0 °C. This is connected with ensuring normal working conditions for both the turboexpander itself and the gas pipelines. Gas in front of the turboexpander is usually heated to 80 - 120 °C. For gas heating, heat exchangers are usually used in which the heat carrier is water. This water in turn is heated in boilers running on natural or associated gas.

In order for the gas to make the rotor blades rotate, this gas must first be directed and accelerated in the desired direction. For this purpose, so-called nozzle arrays are used. They are fixed sections with fixed blades placed between rotating rotor disks. Then the gas enters the rotating blades of the rotor disks, which are located just behind the nozzle arrays and the gas energy is converted into the rotation of the turbine rotor. The set of nozzle arrays and working blades is called the turbine blading and shown in Figure 9.

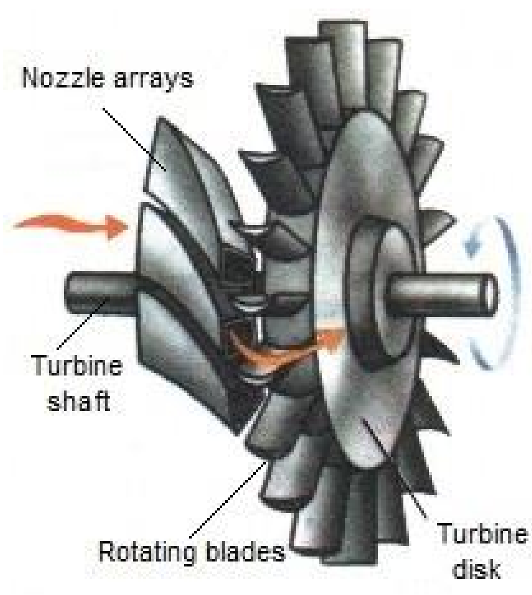

Figure 9 - Graphical representation of the turbine blading[7]

Assumptions made in the calculations:

1. Pure methane is flowing through the gas pipeline. The gas is considered as an ideal gas.
2. There is no heat loss due to friction.

[7] "Energetika.com" [Online]. Available: http://energetika.in.ua/ru/books/book-4/part-1/section-3/3-2. [Accessed May 2020].

3. Calculation is performed according to tabular values of thermophysical methane parameters [24, p. 124]

Initial data for calculation of turboexpander is shown in Table 3:

Table 3 - Initial data for calculations

Parameter	**Nomenclature**	**Value**
Inlet gas pressure	P_{in}	$6.6\ MPa$
Outlet gas pressure	P_{out}	$3.2\ MPa$
Inlet gas temperature	T_{in}	$363\ K$
Rotor speed	n	$7000\ rpm$
Volumetric gas discharge under standard conditions	V_o	$56.94\ \frac{m^3}{s}$
Diameter of the gas supply pipeline	D	$0.3\ m$
Gas enthalpy before turboexpander	i_o	1735.7 $\frac{kJ}{kg}$
Gas entropy before turboexpander	s	9.824 $\frac{kJ}{kg*K}$

In order to make further calculations it is necessary to convert density from standard to working conditions. As it was assumed that gas is an ideal then the gas density under working conditions is calculated as shown in **eq. 1** [25]

$$\rho_w = \rho_i * \frac{T_{n.c}}{T_{in}} * \frac{P_{in}}{P_{n.c}} = 0.7335 * \frac{273}{363} * \frac{6.6}{0{,}1} = 36.4\ \frac{kg}{m^3} \tag{1}$$

Where ρ_i is the density under standard conditions, $T_{n.c} = 20℃$ is the temperature under standard conditions, $P_{n.c} = 101325\ Pa$ is the pressure under standard conditions.

In order to find real volumetric gas discharge it is necessary to recalculate it from standard to working conditions. Hence, the volumetric gas discharge under working conditions is calculated by **eq. 2** [26]

$$V_0' = V_0 * \frac{T_{in}}{T_{n.c}} * \frac{P_{n.c}}{P_{in}} = 56.94 * \frac{363}{273} * \frac{0.1}{6.6} = 1.15\ \frac{m^3}{s} \tag{2}$$

The gas flow through turboexpander can be calculated by **eq. 3** [27, pp. 99-106] (all further equations were taken from one source [28]

$$G = \rho_w * V_0' = 36.4 * 1.15 = 41.86\ \frac{kg}{s} \tag{3}$$

The supply pipeline cross-section area is calculated as shown in **eq. 4**

$$F_i = \frac{\pi * D^2}{4} = \frac{3.14 * 0.3^2}{4} = 0.071\ m^2 \tag{4}$$

The gas velocity in a pipe is the constant movement of the gas volume in that pipe at given cross-section. The gas velocity at the turboexpander inlet is calculated as shown in **eq. 5**

$$C_i = \frac{G}{F_i} = \frac{41.86}{0.071} = 590\frac{m}{s} \tag{5}$$

The adiabatic drop is the difference in heat content between the gas before and after the turbine in the adiabatic expansion of the gas from the full pressure before the turbine to the final static pressure behind the turbine. The adiabatic drop in the turboexpander is calculated as shown in **eq. 6**

$$\begin{aligned} h_{dr} &= \frac{k}{k-1}P_i * \nu^\circ * \left[1 - \left(\frac{P_{out}}{P_{in}}\right)^{\frac{k-1}{k}}\right] = \\ &= \frac{1.345}{1.345-1}6.5 * 10^6 * 0.02788 * \left[1 - \left(\frac{3.2}{6.5}\right)^{\frac{1.345-1}{1.345}}\right] = 130.6 \end{aligned} \tag{6}$$

where k is the heat capacity ratio [29, p. 11], ν° is the specific volume [24].

The process of steam acceleration in the nozzle array is associated with the expansion process. The speed corresponding to the adiabatic drop can be calculated by **eq. 7**

$$C_{ad} = \sqrt{2000 * h} = \sqrt{2000 * 130.66} = 511\frac{m}{s} \tag{7}$$

The degree of reactivity of the turbine characterizes the distribution of gas expansion between the nozzle and the turbine impeller. We assume the degree of reactivity on the middle radius θ'=0.2. Let's preliminary estimate the coefficient of velocity in the nozzle array which is equal to $\varphi_1 = 0.95$ and the angle of flow outlet from the nozzle grating α_1=13°.

Then the optimal ratio of velocities will be calculated as shown in **eq. 8**

$$x_f^{opt} = \frac{\varphi * cos\alpha_1}{2 * \sqrt{1-0{,}2}} = \frac{0.95 * cos13^0}{2 * \sqrt{1-0{,}2}} \tag{8}$$

There is a parameter such as the optimal ratio of velocities, with the correct determination of which the maximum efficiency of the turbine can be achieved. The plot of this dependency is shown in Figure 10.

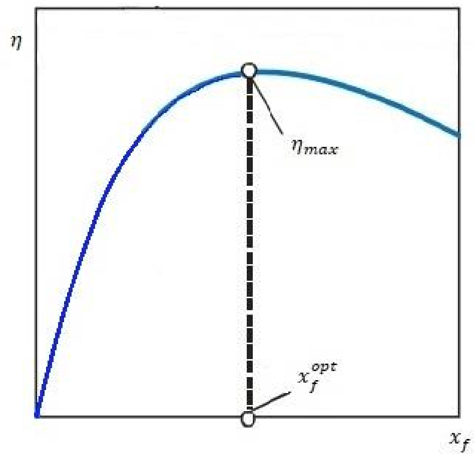

Figure 10 - Dependency of optimal velocities ratio and turbine efficiency

The optimum circumferential speed on a medium radius is calculated as shown in **eq. 9**

$$u = x_f^{opt} * C_{ad} = 0.517 * 511 = 264.3\ \frac{m}{s} \tag{9}$$

The average diameter of turboexpander is calculated as shown in **eq. 10**

$$D_a = \frac{2u}{2\pi n} = \frac{2 * 264.3}{2 * 3.14 * 117} = 0.72\ m \tag{10}$$

Where n is the rotor speed in sec^{-1}.

The available heat difference is the disposable energy that can be converted into work. The available heat difference of the nozzle array is calculated as shown in **eq. 11**

$$h_{n.g} = (1 - \theta') * h = (1 - 0{,}2) * 130.66 = 104.53\ \frac{kJ}{kg} \tag{11}$$

The loss of energy in the nozzle array is calculated as shown in **eq. 12**

$$\Delta h = (1 - \varphi^2) * h = (1 - 0.9^2) * 130.66 = 24.3\ \frac{kJ}{kg} \tag{12}$$

The absolute velocity out of nozzle array is calculated as shown in **eq. 13**

$$C_a = \varphi_{s.c}\sqrt{2000 * h_{n.g.}} = 0.9 * \sqrt{2000 * 104.53} = 411.5\ \frac{m}{s} \tag{13}$$

where $\varphi_{s.c}$ is the velocity coefficient and for turboexpanders can be assumed as 0.9 [28]

According to the tables [29, p. 11], the sound velocity is equal to $a = 486.6\frac{m}{s}$. Then the Mach number can be calculated by **eq. 14**

$$M = \frac{C_a}{a} = \frac{411.5}{486.6} = 0.845 \quad (14)$$

The gas velocity behind the nozzle array is calculated as shown in **eq. 15**

$$C_b = \sqrt{1-\theta'} * C_{ad} = \sqrt{1-0.2} * 511.2 = 457.33 \ \frac{m}{s} \quad (15)$$

The nozzle array blade height is calculated as shown in **eq. 16**

$$l_n = \frac{G}{C_a \pi D_a \sin(\alpha_1)\, \rho_w} = \frac{41.86}{457.33 * 3.14 * 0.72 * sin13 * 36.4} = 5\ mm \quad (16)$$

According to the angle of the flow exit α_1=13° and taking into account that the gas flow is subsonic, a profile «C-90-12A» with a chord b_1=50 mm is chosen. According to the table of "Atlas profiles" [30] the relative step f_{opt}=0.76 is taken. The different nozzle array profiles are shown in Table 4

Table 4 - Atlas of nozzle array profiles of axial turbines[8]

Profile name	$\boldsymbol{\alpha_1}$	$\boldsymbol{f_{opt}}$	$\boldsymbol{M}$
C-9012A	10—14	0,72—0,87	0,60—0.85
C-9015A	13—17	0,70—0,85	0.50—0.85
P-3021A	19—24	0,58—0,68	0,60—0.90

Gas enthalpy behind the nozzle arrays during isentropic expansion is calculated as shown in **eq. 17**

$$i_2 = i_0 - h_{n.g.} = 1735.7 - 104.53 = 1631.17 \ \frac{kJ}{kg} \quad (17)$$

According to the value of enthalpy i_2= $1631.17\frac{kJ}{kg}$ and with the use of tables [29] the specific volume of gas behind the nozzle grid at isentropic expansion is determined $v_1 = 0.024\frac{m^3}{kg}$.

In the nozzle array during the expansion of the working medium thermal energy is converted into kinetic energy, as a result of which the medium behind the nozzle array gets the speed C_{ad}, the direction of which in relation to the front of the grid is determined by an angle α_1.Then, when the flow is turned and the working medium is further expanded, its kinetic energy is

[8] M.E. Filippov, G.A. Lazarev, "Atlas of profiles of grates of axial turbines", 1965, p. 5

transformed into mechanical energy. Profiles of nozzle arrays (upper) and rotating blades (lower) are shown in Figure 11.

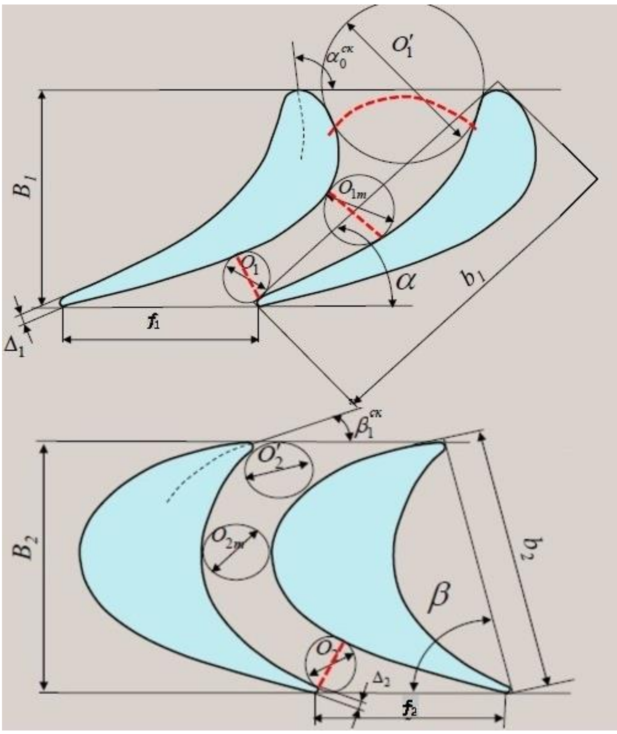

Figure 11 - Profiles of nozzle arrays and rotating blades[9]

where B_1 and B_2 are profile widths,Δ_1, Δ_2 are the thickness of the input edge, b_1 and b_2 are profile chords, α and β are setting angles (the chord of each blade element is an angle with the plane of rotation of the screw, called the setting angle.),f_1 and f_2 are relative steps, α_0^{ck} and β_1^{ck} are so called skeletal angles (the angles between the tangent to the midline of the profile at the inlet (outlet) to the grid and the direction of the circumferential velocity. The channel formed by neighboring profiles is divided into initial O_1', O_2', middle O_{1m}, O_{2m} and final O_1, O_2 sections.

The number of nozzle blades is calculated as shown in **eq. 18**

$$z_c = \frac{\pi D_a}{b_1 f_{opt}} = \frac{3.14 * 0.72}{0.05 * 0.76} = 60 \tag{18}$$

The Reynolds number is calculated as shown in **eq. 19**

$$Re = \frac{C_b b_1}{\mu_g v_1} = \frac{457.33 * 0.05}{13.43 * 10^{-6} * 0.024} = 71 * 10^6 \tag{19}$$

[9] L. A. Belyaev “Thermal and nuclear turbines power stations, 2009, p 48 Energetika," [Online]. Available: http://energetika.in.ua/ru/books/book-4/part-1/section-3/3-2. [Accessed May 2020].

where μ_g is the dynamic viscosity of the gas behind the nozzle grid

Losses on friction can be calculated as shown in **eq. 20**

$$\zeta_f = 0.04(3 - 13\,sin(\alpha_1) + 21 sin^2\alpha_1) = 0.04 * (3 - 13 sin13° + 21 sin^2 13°) = 0.0456 \tag{20}$$

The blade end losses are calculated as shown in **eq. 21**

$$\zeta_{be} = 0.015\left(\frac{b_1}{l_n}\right)(1.5 - 2sin\alpha_1) = 0.015 * \left(\frac{0.05}{0.0084}\right) * (1.5 - 2sin13°) = 0.0937 \tag{21}$$

Taking the thickness of the output edge Δ_e= 0,8 mm, its relative thickness can be found by the **eq. 22**

$$\Delta_e = \frac{\triangle_e}{b_1 * 0.76 * sin\alpha_1} = \frac{0.8}{50 * 0.76 * sin13} = 0.094 \tag{22}$$

Edge losses are calculated as shown in **eq. 23**

$$\Delta\zeta_e = 0.15(\Delta_e - 0.1) = 0.15(0.094 - 0.1) = -0.0009 \tag{23}$$

The correction by Mach number for narrowing arrays is calculated as shown in **eq. 24**

$$\Delta\zeta_M = -0.04M^2 + 0.05M^3 = -0.04 * 0.845^2 + 0.05 * 0.845^3 = 0.0014 \tag{24}$$

The correction by Re is calculated as shown in **eq. 25**

$$\Delta\zeta_{Re} = 5.8 * 10^4 * Re^{-\frac{5}{4}} = 5.8 * 10^4 * 71 * 10^{6^{-\frac{5}{4}}} = 0.00001 \tag{25}$$

The loss coefficient in the nozzle array is calculated as shown in **eq. 26**

$$\zeta_{all} = \zeta_f + \Delta\zeta_e + \Delta\zeta_M + \Delta\zeta_{Re} = 0.0456 + 0.0937 - 0.0009 + + 0.0014 + 0.000011 = 0.14 \tag{26}$$

The accurate velocity factor for the nozzle array is calculated as shown in **eq. 27**

$$\varphi_a = \sqrt{1 - \zeta_{all}} = 0.927 \tag{27}$$

The accurate gas discharge rate from the nozzle array is calculated as shown in **eq. 28**

$$C_1 = \varphi C_b = 0.927 * 457.33 = 424.16\ \frac{m}{s} \tag{28}$$

The axial velocity is calculated as shown in **eq. 29**

$$C_{1a} = C_1 sin\alpha_1 = 424.16 * sin13° = 95.42\ \frac{m}{s} \tag{29}$$

The radial velocity is calculated as shown in **eq. 30**

$$C_{1u} = C_1 cos\alpha_1 = 424.16 * sin13° = 413.33\ \frac{m}{s} \tag{30}$$

The relative gas exit velocity from the nozzle array is calculated as shown in **eq. 31**

$$\omega_1 = (C_1^2 + u^2 - 2uC_1 cos\alpha_1)^{0.5} = (424.16^2 + 264.3^2 - 264.3 * \\ * 424.16 * cos13°)^{0.5} = 177\frac{m}{s} \tag{31}$$

The inlet angle of the flow into the rotating blades is calculated as shown in **eq. 32**

$$\beta_1 = arctg\left(\frac{C_{1a}}{C_{1u} - u}\right) = arctg\left(\frac{95.42}{413.33 - 264.}\right) = 32.5° \tag{32}$$

The theoretical relative output velocity out of rotating blades is calculated as shown in **eq. 33**:

$$\omega_{2t} = (\omega_1^2 + \theta' C_{ad})^{0.5} = (177^2 + 0.2 * 511^2) = 289\ \frac{m}{s} \tag{33}$$

According to the tables [29, p. 11], the sound velocity is equal to $a = 473\frac{m}{s}$. Then the Mach number can be calculated by **eq. 34**

$$M_2 = \frac{\omega_{2t}}{a} = \frac{289}{473} = 0.61 \tag{34}$$

The loss of kinetic flow energy in the nozzle array is calculated as shown in **eq. 35**

$$\Delta H_c = \zeta_{all} h_{n.g.} = 0.14 * 104.53 = 14.63\ \frac{kJ}{kg} \tag{35}$$

Assuming the flow coefficient, which is the ratio of actual velocity to theoretical velocity, $\acute{\upsilon}_2$=0.95, the output area of the rotating blade can be calculated by **eq. 36**

$$F_{rb} = \frac{G\nu_1}{\acute{\upsilon}_2 \omega_{2t}} = \frac{41.86 * 0.024}{0.95 * 289} = 0.0036\ m^2 \tag{36}$$

The cover of the working blades should be 3-5 mm and can be calculated by the following **eq.37**

$$\Delta l = 1.8 + 0.06 * l_n = 1.8 + 0.06 * 5 = 2.1\ mm \tag{37}$$

Then the height of the working blade can be calculated by the following **eq. 38**

$$l_2 = l_n + \Delta l = 5 + 2{,}1 = 7{,}1\ mm \tag{38}$$

The efficient angle of the flow outlet from the rotating blades is calculated as shown in **eq. 39**

$$\beta_{2eff} = \arcsin\left(\frac{F_{rb}}{\pi D_a l_2}\right) = arcsin\left(\frac{0.0036}{3.14 * 0.72 * 0.0071}\right) = 13° \quad (39)$$

According to the angles $\beta_1 = 32.5°$ and $\beta_{2eff} = 13°$ a profile «P-26-17A» from the table of “Atlas profiles” [30] will be chosen. The relative step $f_2 = 0.61$ a chord $b_2 = 50\ \mathrm{mm}$ will be taken.

The number of the rotating blades is calculated as shown in **eq. 40**

$$z_2 = \frac{\pi D_a}{b_2 f_{opt}} = \frac{3.14 * 0.72}{0.05 * 0.61} = 75 \quad (40)$$

Friction losses in the rotating blades are calculated as shown in **eq. 41**

$$\begin{aligned}\zeta_{fr}^{rb} &= 0.08(1.841 - 1.584sin\Delta\beta + 0.62sin^2\Delta\beta) \\ &= 0.08(1.841 - 1.584sin134.5° + 0.62sin^2 134.5°) = 0.082\end{aligned} \quad (41)$$

Where rotation angle the flow $\Delta\beta = 180 - (\beta_1 + \beta_{2eff}) = 180 - (32.5 + 13) = 134.5°$

Rotating blade end losses are calculated as shown in **eq. 42**

$$\begin{aligned}\zeta_e^{rb} &= 0.026\frac{b_2}{l_2}(1.87 - 1.15sin134.5°) \\ &= 0.025 * \frac{30}{7.1}(1.87 - 1.15sin134.5°) = 0.11\end{aligned} \quad (42)$$

Taking the thickness of the output edge [30] $\Delta_{\mathrm{kr}} = 0{,}5\ \mathrm{mm}$, it’s relative thickness can be found by the **eq.43**

$$\Delta'_{kr} = \frac{\Delta_{kr}}{(b_2 f_2 sin\beta_{2eff})} = \frac{0.5}{30 * 0.61 * sin13°} = 0.121\ mm \quad (43)$$

The edge loss factor is calculated as shown in **eq. 44**

$$\Delta\zeta_{elf}^{rb} = 0.15(\Delta_{\mathrm{kr}} - 0{,}1) = 0.15 * (0.121 - 0.1) = 0.0031 \quad (44)$$

Losses related to the flow regime in the rotated blades are determined by the influence of M and Re, as well as by the difference between the angle of flow inlet and the optimal one.

The correction by M for the rotated blades is calculated as shown in **eq. 45**

$$\Delta\zeta_M^{rb} = -0.04{M_2}^2 + 0.05{M_2}^3 = -0.04 * 0.61^2 + 0.05 * 0.61^3 = 0.0263 \quad (45)$$

The Reynolds number is calculated as shown in **eq. 46**

$$Re = \frac{\omega_{2t} b_2}{\mu_g \nu_2} = \frac{289 * 0.05}{13.43 * 10^{-6} * 0.024} = 44.8 * 10^6 \tag{46}$$

The correction by Re is calculated as shown in **eq. 47**

$$\Delta\zeta_{Re}^{rb} = 5.8 * 10^4 * Re^{-\frac{5}{4}} = 5.8 * 10^4 * 44.8 * 10^{6^{-\frac{5}{4}}} = 1.5 * 10^{-5} \tag{47}$$

The loss factor for the rotated blades is calculated as shown in **eq. 48**

$$\zeta_{lr}^{rb} = \zeta_{fr}^{rb} + \Delta\zeta_{elf}^{rb} + \Delta\zeta_M^{rb} + \Delta\zeta_{Re}^{rb} = 0.082 + 0.11 + 0.0263 + 1.5 * 10^{-5} = 0.2 \tag{48}$$

The accurate velocity coefficient for the rotated blades is calculated as shown in **eq. 49**

$$\psi = (1 - \zeta_{lr}^{rb})^{0.5} = (1 - 0.22)^{0.5} = 0.88 \tag{49}$$

In order to calculate the absolute rotated blades output velocity, it is necessary to plot velocity diagram. Graphical representation of the velocity diagram is shown in Figure 12

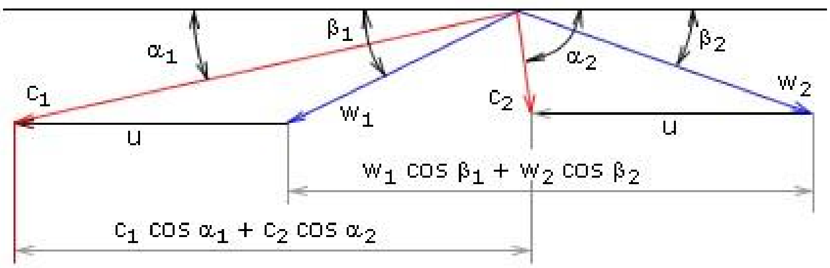

Figure 12 - Velocity diagram

The actual output velocity from the rotated blades is calculated as shown in **eq. 50**

$$\omega_{rb} = \psi \omega_{2t} = 0.88 * 289 = 254.3 \frac{m}{s} \tag{50}$$

The angle of flow outlet from the rotated blades is calculated as shown in **eq. 51**

$$\beta_2 = arcsin\left(\frac{\mu_2}{\psi} sin\beta_{2eff}\right) = arcsin\left(\frac{0.95}{0.88} sin13°\right) = 14° \tag{51}$$

The axial velocity of the rotated blades is calculated as shown in **eq. 52**

$$\omega_{2a} = \omega_{rb} sin\beta_2 = 254.3 * sin14° = 61.52 \frac{m}{s} \tag{52}$$

The radial velocity of the rotated blades is calculated as shown in **eq. 53**

$$\omega_{2u} = \omega_{rb} cos\beta_2 = 254.3 * cos14° = 246.74 \frac{m}{s} \tag{53}$$

The absolute rotated blades output velocity is calculated as shown in **eq. 54**

$$c_2 = \left(\omega_{rb}^2 + u^2 - 2u\omega_{rb} cos\beta_2\right)^{0.5} = \\ = (254.3^2 + 264.3^2 - 2 * 264.3 * 254.3 * cos14 = 62.77 \frac{m}{s} \tag{54}$$

The angle of flow outlet from the rotated blades is calculated as shown in **eq. 55**

$$\alpha_2 = arctg \frac{\omega_{2a}}{\omega_{2u} - u} = arctg \frac{61.52}{246.74 - 264.3} = 74° \tag{55}$$

The loss of kinetic energy in the rotated blades is calculated as shown in **eq. 56**

$$\Delta H_{ke} = \frac{\omega_{2t}^2 \zeta_{lr}}{2} = \frac{289^2 * 0.22}{2} = 9140 \frac{J}{kg} = 9.14 \frac{kJ}{kg} \tag{56}$$

The loss with output velocity is calculated as shown in **eq. 57**

$$\Delta H_{ov} = \frac{c_2^2}{2} = \frac{62.77^2}{2} = 1970 \frac{J}{kg} = 1.97 \ \frac{kJ}{kg} \tag{57}$$

The useful work of turboexpander blades is calculated as shown in **eq. 58**

$$H_u = h - \Delta H_c - \Delta H_{ke} - \Delta H_{ov} = 130.66 - 14.63 - 9.14 - 1.97 = 104.9 \ \frac{kJ}{kg} \tag{58}$$

The relative turboexpander blade efficiency is calculated as shown in **eq. 59**

$$\eta_t = \frac{H_u}{h} = \frac{104.9}{130.66} = 0.8 \tag{59}$$

The turboexpander power can be calculated as shown in **eq. 60**

$$N_t = Gh\eta_t = 41.86 * 130.66 * 0.8 = 4375.54 \ kW \tag{60}$$

The useful work of a turboexpander is calculated as shown in **eq. 61**

$$H_i = h\eta_t = 130.66 * 0.8 = 104.53 \frac{kJ}{kg} \tag{61}$$

The gas enthalpy outside the turboexpander is calculated as shown in **eq. 62**

$$i_2 = i_0 - H_i = 1735.7 - 104.53 = 1631.17 \frac{kJ}{kg} \quad (62)$$

With the help of tables [24, p. 109] for pressure $p_2 = 3.2\ MPa$ and enthalpy $i_2 = 1631.17$ the temperature after turboexpander will be $T_2 = 313\ K$

3.2 Thermodynamic calculation of the gas turbine unit

In order to choose the optimal compression ratio in the gas turbine cycle, the thermal scheme is calculated with the given coefficients. Calculations are carried out according to the method described in [31, pp. 19-21], and the results of the calculation are summarized in Table 6. In this calculation the two-shaft scheme of a gas turbine unit for driving an electric generator was chosen. Input data for the gas turbine unit calculation is provided in Table 5.

Table 5 - Initial data for GTU calculation[10]

Parameter	**Nomenclature**	**Value**
Atmospheric air pressure	P_{air}	$101315\ Pa$
Temperature of combustion products before the turbine	T_{out}	$1200\ K$
Ambient air temperature	T_{air}	$288\ K$
Efficiency of the high-pressure turbine	η_{high}	0.87
Efficiency of the low-pressure turbine	η_{low}	0.88
Efficiency of the compressor	η_{comp}	0.87
Efficiency of the combustion chamber	$\eta_{c.c}$	0.99
Mechanical efficiency	η_m	0,98
Cooling air losses	q_c	0.05
Air losses through seals	q_s	0.02
Flow coefficient	τ	0.95
Specific heat capacity of air in the compressor	C_{pk}	1.01 $\frac{kJ}{kg*K}$
Specific heat capacity of combustion products in the turbine	C_{pt}	1.15 $\frac{kJ}{kg*K}$

[10] O. V. Komarov and V. L. Blinov, "Thermal and gas dynamic calculations of gas turbine installations," 2018, pp. 147-148.

Specific heat capacity of air before the combustion chamber	C_{pb}	1.03 $\frac{kJ}{kg*K}$
Specific heat capacity of flue gases in the combustion chamber	C_{ch}	1.12 $\frac{kJ}{kg*K}$
Heat capacity ratio for air	k_k	1.4
Hat capacity ratio for flue gas	k_t	1.333
Hydraulic resistance along the path	ζ	0.1

The compression ratio in compressors is the ratio of the output pressure to the inlet pressure. For the first iteration we assume that compression ratio is equal to $\pi_k = 6$

Compressor performance is calculated as shown in **eq. 63.**

$$\overline{H} = \pi_k^{\frac{k_k-1}{k_k}} - 1 = 6^{\frac{1.4-1}{1,4}} - 1 = 0.67 \tag{63}$$

The specific compression work is calculated as shown in **eq. 64**

$$H_k = \frac{C_{pk}T_{air}\overline{H}}{\eta_{comp}} = \frac{1.01 * 288 * 0.67}{0.87} = 223.51\frac{kJ}{kg} \tag{64}$$

The air temperature after compressor is calculated as shown in **eq. 65**

$$T_k = T_{air} + \frac{H_k}{C_{pk}} = 288 + \frac{223.51}{1.01} = 509.3\,K \tag{65}$$

The total degree of expansion in turbines is calculated as shown in **eq. 66**

$$\pi_{T\Sigma} = \pi_k(1-\zeta) = 6 * (1 - 0.1) = 5.4 \tag{66}$$

The specific work of the expansion in the high-pressure turbine is calculated as shown in **eq. 67**

$$H_{t1} = \frac{H_k}{\tau\eta_{м}} = \frac{223.51}{0.98 * 0.95} = 240.1\frac{kJ}{kg} \tag{67}$$

The combustion product temperature outside the high-pressure turbine is calculated as shown in **eq. 68**

$$T_{t1} = T_{t2} = T_{out} - \frac{H_{t1}}{C_{pt}} = 1200 - \frac{240.1}{1.15} = 991.2\,K \tag{68}$$

The expansion ratio of combustion products in the high-pressure turbine is calculated as shown in **eq. 69**

$$\pi_{t1} = \left[1 - \frac{H_{t1}}{C_{pt}\eta_{high}T_{out}}\right]^{\frac{k_t-1}{k_t}} = \left[1 - \frac{240.1}{1.15 * 0.87 * 1200}\right]^{\frac{1.333-1}{1.333}} = 2.44 \tag{69}$$

The expansion ratio of combustion products in the low-pressure turbine is calculated as shown in **eq. 70**

$$\pi_{t2} = \frac{\pi_{T\Sigma}}{\pi_{t1}} = \frac{5.4}{2.44} = 2.21 \tag{70}$$

The specific work of the expansion in the low-pressure turbine is calculated as shown in **eq. 71**

$$H_{t2} = C_{pt}T_{t2}\left(1 - \pi_{t2}^{\frac{1-k_k}{k_k}}\right)\eta_{low} = 1.15 * 991.2 * \left(1 - 2.21^{\frac{1-1.4}{1.4}}\right) * \ 0.88 = 203.45\frac{kJ}{kg} \tag{71}$$

The specific effective work is calculated as shown in **eq. 72**

$$H_e = H_{t2}\tau\eta_m = 203.45 * 0.95 * 0.98 = 199.38\frac{kJ}{kg} \tag{72}$$

The temperature of combustion products behind the low-pressure turbine is calculated as shown in **eq. 73**

$$T_t = T_{t2} - \frac{H_e}{C_{pt}} = 1200 - \frac{199.38}{1.15} = 817.86\ K \tag{73}$$

The amount of the air heat entering the combustion chamber is calculated as shown in **eq. 74**

$$Q_a = C_{pb}T_k(1 - q_c - q_s) = 1.03 * 509.3 * (1 - 0.05 - 0.02) = 490.5\frac{kJ}{kg} \tag{74}$$

The amount of heat transferred to the flue gases in combustion chamber is calculated as shown in **eq. 75**

$$Q_{c.h} = C_{ch}T_{out}(1 - q_c) - Q_a = 1.12 * 1200 * (1 - 0.05) - 490.5 = 786.32\frac{kJ}{kg} \tag{75}$$

Absolute thermal efficiency is calculated as shown in **eq. 76**

$$\eta_e = \frac{H_e\eta_{c.c}}{Q_{c.h}} = \frac{199.4 * 0.99}{786.32} = 0.25 \tag{76}$$

The following calculations for other values of π_k were performed in Microsoft Excel. The results are summarized in Table 6.

Table 6 - Selecting a calculated value for the compression ratio

№	Parameter	Value					
1	π_k	6	7	8	8.6	10	14
2	$\overline{H}$	0,67	0.74	0.81	0.85	0.93	1.13
3	H_k	223.51	248.63	271.3	283.95	311.17	376.31
4	T_k	509.3	534.17	556.62	569.14	569.1	660.6
5	$\pi_{T\Sigma}$	5.4	6.3	7.2	7.74	9	12
6	H_{t1}	240.1	267.1	291.4	304.99	334.24	404.2
7	T_{t1}	991.2	967.8	946.6	934.8	909.4	848.5
8	π_{t1}	2.44	2.74	3.04	3.23	3.7	5.17
9	π_{t2}	2.21	2.3	2.37	2.39	2.44	2.44
10	H_{t2}	203.45	207.53	208.94	208.88	206.86	192.91
11	H_e	199.38	203.4	204.76	204.7	202.73	189.05
12	T_t	817.86	1023.15	1021.94	1022	1023.7	1035.6
13	Q_a	490.5	514.43	536.05	548.11	574.07	636.2
14	$Q_{c.h}$	786.32	762.4	740.75	728.7	702.73	640.62
15	η_e	0.25	0.263	0.272	0.277	0.284	0.291

According to the calculated data the dependence $H_e = f(\pi_k)$ and $\eta_e = f(\pi_k)$ are plotted to select the correct compression ratio. At the intersection of the lines in Figure 13 the optimal compression ratio is found.

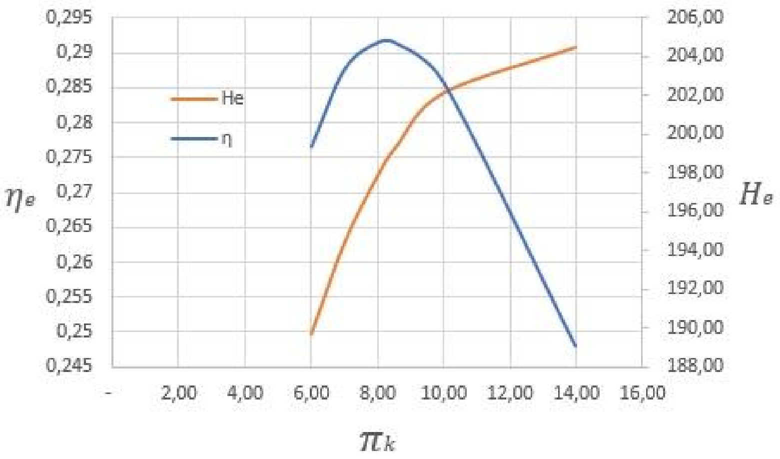

Figure 13 - Optimal compression ratio

Based on the calculated results, the nominal value $\pi_k = 10.3$ is chosen. According to the new value of the compression ratio power of the gas turbine unit is calculated, taking into account the true values of heat capacity and adiabatic values.

Specific compression work is calculated as shown in **eq. 77**

$$H_k = \frac{C_{pk} T_{air} \pi_k^{\frac{k_k - 1}{k_k}} - 1}{\eta_{comp}} = \frac{1.01 * 288 * 10{,}3^{\frac{1{,}4-1}{1{,}4}} - 1}{0.87} = 316.65 \frac{kJ}{kg} \tag{77}$$

The temperature of the air outside the compressor is calculated as shown in **eq. 78**

$$T_k = T_{air} + \frac{H_k}{C_{pk}} = 288 + \frac{316.65}{1.01} 601.5\ K \tag{78}$$

The average air compression process temperature in the compressor is calculated as shown in **eq. 79**

$$T_a = \frac{T_{air} + T_k}{2} = \frac{288 + 601.5}{2} = 444.75\ K = 174.75°\mathrm{C} \tag{79}$$

Thermophysical properties of air are specified by the average temperature of the compression process at equivalence ratio (EQR) equal to $E_1 = \infty$. The Specific heat capacity and heat capacity ratio are determined according to the Figure 14

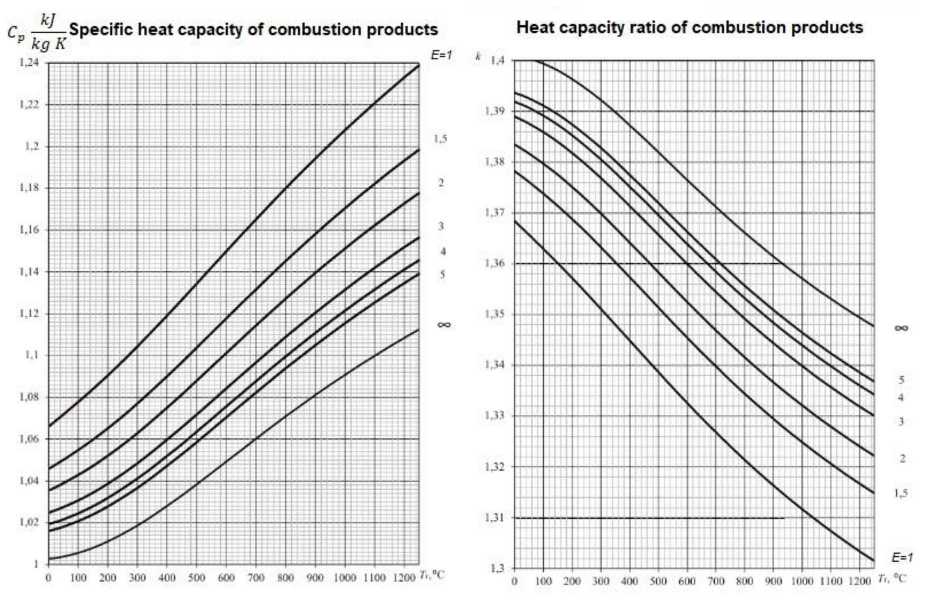

Figure 14 - Thermophysical properties of combustion products[11]

[11] O. V. Komarov and V. L. Blinov, "Thermal and gas dynamic calculations of gas turbine installations," 2018, pp. 147-148.

The value of specific heat capacity is $C'_{pk} = 1.015 \frac{kJ}{kg*K}$ and the value of the heat capacity ratio is $k' = 1.397$.

The ratio of fuel to air is commonly characterized using the so-called equivalence ratio (EQR), which is the ratio of fuel to air normalized by the ratio of fuel to air needed for complete stoichiometric combustion. An EQR of 0 represents pure air, values below one characterize conditions with excess air and values of greater that one represents conditions under which complete combustion is not possible due to the lack of oxygen. Pure fuel has an EQR equal to infinity.

The value of the specific compression work is calculated as shown in **eq. 80**

$$H'_k = \frac{C'_{pk} T_{air} \pi_k^{\frac{k'-1}{k'}} - 1}{\eta_{comp}} = \frac{1.015 * 288 * 10{,}3^{\frac{1{,}397-1}{1{,}397}} - 1}{0.87} = 315.88 \frac{kJ}{kg} \tag{80}$$

The value of the air temperature outside the compressor is calculated as shown in **eq. 81**

$$T'_k = T_{air} + \frac{H_k}{C'_{pk}} = 288 + \frac{316.65}{1.015} = 599.97\ K \tag{81}$$

The value of the average air compression process temperature in the compressor is calculated as shown in **eq. 82**

$$T'_a = \frac{T_{air} + T'_k}{2} = \frac{288 + 599.97}{2} = 443.98\ K = 170.98℃ \tag{82}$$

Due to the small change in the average temperature of the air compression process in the compressor, further clarification of the thermophysical parameters of the air is not required.

The total degree of expansion in turbines is calculated as shown in **eq. 83**

$$\pi_{T\Sigma} = \pi_k(1 - \zeta) = 10.3 * (1 - 0.1) = 9.27 \tag{83}$$

The specific work of the expansion in the high-pressure turbine is calculated as shown in **eq. 84**

$$H_{t1} = \frac{H'_k}{\tau \eta_{м}} = \frac{315.88}{0.98 * 0.95} = 339.3 \tag{84}$$

The combustion product temperature outside the high-pressure turbine is calculated as shown in **eq. 85**

$$T_{t1} = T_{out} - \frac{H_{t1}}{C_{pt}} = 1200 - \frac{339.3}{1.15} = 905\ K \quad (85)$$

The equivalence ratio for hydrocarbon fuel combustion is determined according to Figure 15 and is equal to $E_2 = 4.2$.

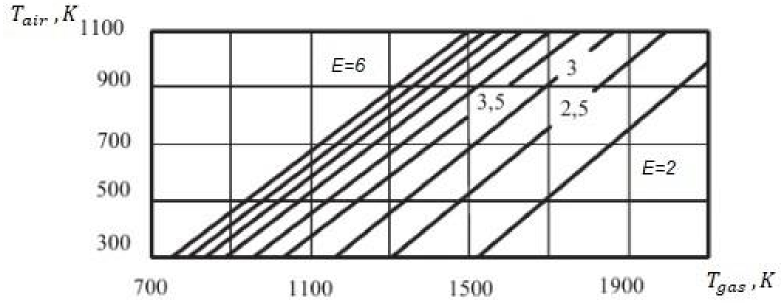

Figure 15 - Equivalent ratio for hydrocarbon fuel combustion at constant pressure[12]

The average temperature of the combustion product expansion process in the high-pressure turbine is calculated as shown in **eq. 86**

$$T_a = \frac{T_{out} + T_{t1}}{2} = \frac{1200 + 905}{2} = 1052.5\ K = 779.5℃ \quad (86)$$

The thermophysical properties of the combustion products are specified at the average temperature of the expansion process and the air ratio factor equal to $E_2 = 4.2$.

The value of the combustion products specific heat capacity is $C''_{pt} = 1.098 \frac{kJ}{kg*K}$ and the heat capacity ratio is $k'' = 1.356$.

The temperature of the combustion products behind the high-pressure turbine and the average temperature of the expansion process in the low-pressure turbine is calculated as shown in **eq. 87** and **eq.88**

$$T'_{t1} = T_{out} - \frac{H_{t1}}{C''_{pt}} = 1200 - \frac{339.3}{1.098} = 891\ K \quad (87)$$

$$T'_a = \frac{T_{out} + T'_{t1}}{2} = \frac{1200 + 891}{2} = 1045.5\ K = 772.5℃ \quad (88)$$

[12] O. V. Komarov and V. L. Blinov, "Thermal and gas dynamic calculations of gas turbine installations," 2018, pp. 147-148.

The expansion ratio of combustion products in the high-pressure turbine is calculated as shown in **eq. 89**

$$\pi_{t1} = \left[1 - \frac{H_{t1}}{C''_{pt}\eta_{high}T_{out}}\right]^{\frac{k''-1}{k''}} = \left[1 - \frac{339.3}{1.098 * 0.87 * 1200}\right]^{\frac{1.356-1}{1.136}} = 3.8 \tag{89}$$

The expansion ratio of combustion products in a low-pressure turbine is calculated as shown in **eq. 90**

$$\pi_{t2} = \frac{\pi_T}{\pi_{t1}} = \frac{9.27}{3.8} = 2.44 \tag{90}$$

The specific work of the expansion in the low-pressure turbine is calculated as shown in **eq. 91**

$$H_{t2} = C''_{pt}T'_{t1}\left(1 - \pi_{t2}^{\frac{1-k''}{k''}}\right)\eta_{low} = 1.098 * 891 * \left(1 - 2.44^{\frac{1-1.356}{1.356}}\right) * 0.88 = 192.39\frac{kJ}{kg} \tag{91}$$

The temperature of the combustion products behind the low-pressure turbine is calculated as shown in **eq. 92**

$$T_t = T_{t2} - \frac{H_{t2}}{C'_{pt}} = 1200 - \frac{192.39}{1.098} = 550.81\ K \tag{92}$$

The average temperature of the combustion product expansion process in a low-pressure turbine is calculated as shown in **eq. 93**

$$T'_a = \frac{T_{out} + T_t}{2} = \frac{1200 + 550.81}{2} = 875.4\ K = 602.4℃ \tag{93}$$

The value of the specific heat capacity at $T'_a = 602.4℃$ is $C''_{pt.a} = 1.074\ \frac{kJ}{kg*K}$ and heat capacity ratio is $k''_a = 1.364$. Since the changes in the thermal properties of combustion products are insignificant it makes no sense to specify the corresponding thermal parameters.

The specific effective work is calculated as shown in **eq. 94**

$$H_e = H_{t2}\tau\eta_m = 192.39 * 0.95 * 0.98 = 179.12\frac{kJ}{kg} \tag{94}$$

The specific heat capacity of the air which is specified at the temperature of $T'_k = 599.97K$ and the equivalence ratio equal to $E_1 = \infty$ is equal to $C^k_{pb} = 1.05\ \frac{kJ}{kg*K}$.

The amount of the air heat entering the combustion chamber is calculated as shown in **eq. 95**

$$Q_a = C_{pb}^k T_k'(1 - q_c - q_s) = 1.05 * 599.97 * (1 - 0.05 - 0.02) = 585.87 \frac{kJ}{kg} \quad (95)$$

The specific heat capacity of the air which is specified at the temperature of $T_{out} = 1200\ K$ and the air ratio factor equal to $E_2 = 4.2$ is $C_{pb}^{out} = 1.14\ \frac{kJ}{kgK}$.

The amount of heat supplied to the combustion chamber is calculated as shown in **eq. 96**

$$Q_{c.h} = \frac{C_{pb}^{out} T_{out}(1 - q_c)}{\eta_{c.c}} - Q_a = \frac{1.14 * 1200 * (1 - 0.05)}{0.99} - 585.87 = 726.85 \frac{kJ}{kg} \quad (96)$$

The absolute thermal efficiency is calculated as shown in **eq. 97**

$$\eta_e = \frac{H_e}{Q_{c.h}} = \frac{179{,}12}{726{,}85} = 0.246 \quad (97)$$

The power of the gas turbine unit is calculated as shown in **eq. 98**

$$N_e = GH_e = 41.86 * 179.12 = 7.5\ MW \quad (98)$$

3.3 Thermodynamic calculation of the combined cycle power plant

To calculate the heat recovery steam generator (HRSG) unit, the following assumptions might be applied [32, p. 28]:

1. Temperature drop in the outlet of superheater is $\Delta t_{ss} = 25°\text{C}$
2. Temperature drop in the inlet of economizer is $\Delta t_{ec} = 3.5°\text{C}$
3. Temperature drop in the inlet of vaporizer is $\Delta t_{vap} = 8°\text{C}$
4. Underheating of the condensate to the saturation temperature in deaerator is $\Delta t_d = 8°\text{C}$
5. Coefficient of hydraulic resistance is $\varsigma = 0.93$
6. Coefficient that takes into account the loss of water pressure in the path from the economizer to the drum $\varsigma_d = 1.046$
7. Hydraulic resistance of the economizer $\Delta P_{ec} = 0.17\ MPa$
8. Coefficient of hydraulic resistance of block valves $\varsigma_{b.v} = 0.93$

According to the thermal scheme, the HRSG consists of four heating surfaces: the steam superheater (SH), the vaporizer (VP), the economizer (EC) and the gas condensate heater (GCH). The purpose of the calculation is to determine the steam capacity of the HRSG and gas temperatures in the characteristic sections of the gas-air circuit. The gas and air path of the utilizer boiler is broken down by five sections limiting the listed heating surfaces:

1. Before the steam superheater

2. Before the vaporizer
3. After the economizer
4. Before the condensate gas heater
5. Behind the condensate gas heater

The main components of the heat recovery steam generator are shown on the Figure 16

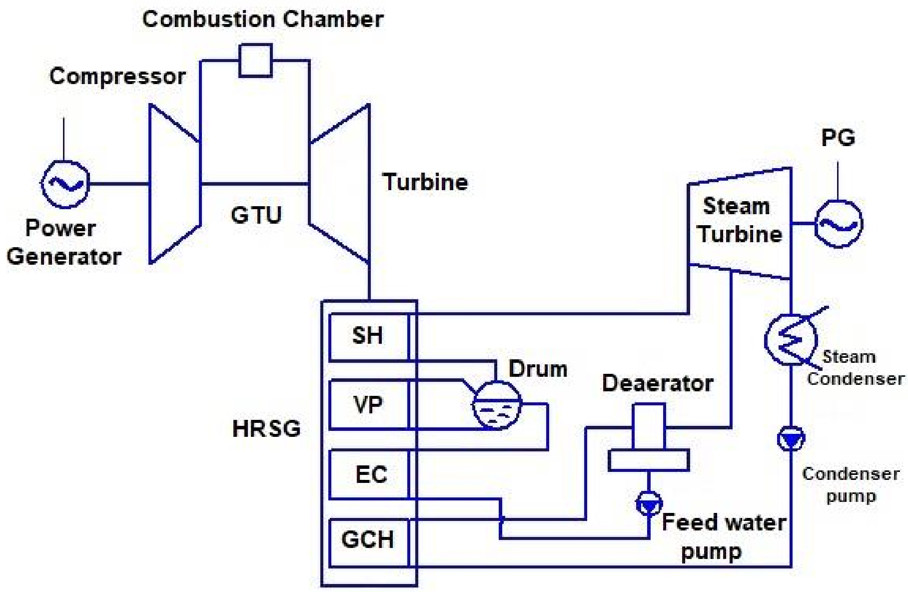

Figure 16 - Combined cycle gas turbine unit with the HRSG

For further calculation it is necessary to specify the steam pressure at the outlet of the superheater. Assume that the pressure P_{sh}^{out} is equal to $4\ MPa$.

The steam pressure in the drum is calculated as shown in **eq. 99**

$$P_d = \frac{P_{sh}^{out}}{\varsigma} = \frac{4}{0.93} = 4.3\ MPa \quad (99)$$

The temperature and enthalpy of boiling water at pressure $P_d = 4.3\ MPa$ are equal to $T_d = 250.7℃$ and $i_d' = 1108\ \frac{kJ}{kg}$. The enthalpy of dry saturated steam is $i_d'' = 2798.3\ \frac{kJ}{kg}$ [33]

Gas parameters in sections 1 and 3 are calculated as shown in **eq. 100-102**

$$i_g' = C_{pb}' T_t = 1.14 * 550.81 = 627.92\ \frac{kJ}{kg} \quad (100)$$

$$T_g^3 = T_d + \Delta t_{vap} = 250.7 + 8 = 258.7℃ \quad (101)$$

$$i_g^3 = C_{pb}' T_g^3 = 1.14 * 258.7 = 295\ \frac{kJ}{kg} \quad (102)$$

The Rankine cycle is theoretical thermodynamic cycle of a steam machine. The flow diagram in coordinates t-s of the steam power plant is demonstrated in the Figure 17

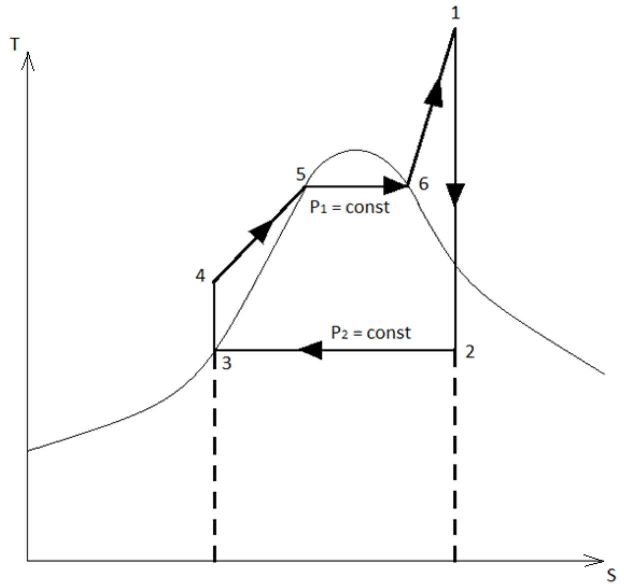

Figure 17 - The Rankine cycle in the t-s diagram

From state 1 to state 2 steam expands adiabatically in a steam turbine and from state 2 to state 3 the process of heat rejection from the condensed steam can be seen. The process 3-4 is an adiabatic compression of condensate by centrifugal pump while 4-5 is the process of heat transfer to the feed water in the economizer up to boiling point. From state 5 to 6 is the process of evaporation in the vaporizer and the process 6-1 is the superheating of steam.

The steam parameters behind the superheater are calculated as shown in **eq. 103 and 104**

$$T_{sh}^{out} = T_t - \Delta t_{ss} = 550.81 - 25 = 525.81℃ \quad (103)$$

$$P_{sh}^{out} = P_d * \varsigma = 4.3 * 0.93 = 4\ MPa \quad (104)$$

$i_{sh}^{out} = 3277\ \frac{kJ}{kg}$ [12]

The water temperature and enthalpy after the economizer are calculated as shown in **eq. 105 and 106** [34]

$$T_{ec}^{out} = T_d - \Delta t_{ec} = 250.7 - 3.5 = 247.2℃ \quad (105)$$

$$i_{ec}^{out} = 1072\ \frac{kJ}{kg} \quad (106)$$

Assume that the pressure in the deaerator $P_{de} is\ 0.7\ MPa$, the saturation temperature in the deaerator $T_{de} = 165$ ℃, the saturation enthalpy in the deaerator $i'_{de} = 697 \frac{kJ}{kg}$ [34]

The water temperature at the economizer inlet is calculated as shown in **eq. 107**

$$T_{ec}^{in} = \frac{i'_{de}}{C_w} = \frac{697}{4.186} = 166.5℃ \tag{107}$$

where C_w is a heat capacity of water

The steam flow rate from the superheater is calculated as shown in **eq. 108**

$$G_{st} = G\frac{i'_g - i_g^3}{i_{sh}^{out} - i_{ec}^{out}} = 41.86\frac{627.92 - 295}{3277 - 1072} = 6.32\frac{kg}{s} \tag{108}$$

The enthalpy of gases at the outlet of the economizer is calculated as shown in **eq. 109**

$$i_{gas}^{ec} = i_g^3 - \frac{G_{st}(i_{ec}^{out} - i'_{de})}{G} = 295 - \frac{6.32 * (1072 - 697)}{41.86} = 238.4\frac{kJ}{kg} \tag{109}$$

The economizer output gas temperature is calculated as shown in **eq. 110**

$$T_{ec}^{g} = \frac{i_{gas}^{ec}}{C'_{pb}} = \frac{238.4}{1.14} = 209℃ \tag{110}$$

When steam goes from the HRSG to the turbine its pressure decreases due to the hydraulic resistance of the pipeline from the superheater to the block valves.

The steam pressure before the turbine is calculated as shown in **eq. 111**

$$P_s^t = P_{sh}^{out}\,\varsigma_{b.v} = 4 * 0.93 = 3.72\ MPa \tag{111}$$

Steam parameters in the steam extraction chamber:

1. The steam pressure in the extraction chamber is equal to $P_{bh} = 1\ MPa$
2. The steam enthalpy in the extraction chamber determined by an $i - s$ chart is equal to $i_{bh} = 3009\frac{kJ}{kg}$

The temperature and enthalpy of the main condensate at the deaerator inlet are calculated as shown in **eq. 112,113**

$$T_{de}^{in} = T_{de} - \Delta t_d = 165 - 8 = 157℃ \tag{112}$$

$$i_{de}^{in} = T_{de}^{in}C_w = 157 * 4.186 = 657.2\frac{kJ}{kg} \tag{113}$$

The steam flow rate from the turbine extraction chamber to deaerator is calculated as shown in **eq. 114**

$$G_{dr} = G_{st}\frac{i'_{de} - i^{in}_{de}}{i_{ex} - i^{in}_{de}} = 6.32 * \frac{697 - 657.2}{3009 - 657.2} = 0.11\frac{kg}{s} \tag{114}$$

The flow rate of the main condensate entering the deaerator from the gas condensate heater is calculated as shown in **eq. 115**

$$G_{cond} = G_{st} - G_{dr} = 6.32 - 0.11 = 6.21\frac{kg}{s} \tag{115}$$

The steam flow rate for leaks through turbine rotor and valve seals is calculated as shown in **eq. 116**

$$G_l = 0.01 * G_{st} = 0.01 * 6.32 = 0.0632\frac{kg}{s} \tag{116}$$

Assume that the pressure in the condenser is equal to $P'_c = 0.005\,MPa$, the saturation temperature in the condenser is $T^s_c = 32.9$°C. The steam enthalpy in the condenser at the end of the expansion process is equal to $i_{cond} = 2278\frac{kJ}{kg}$

The steam turbine power is calculated as shown in **eq. 117**

$$N_{ST} = (i^{out}_{sh} - i_{ex})(G_{st} - G_l) + (i_{ex} - i_{cond})(G_{st} - G_l - G_{dr}) = (3277 - 3009) * (6.32 - 0.0632) + (3009 - 2278) * (6.32 - 0.0632 - 0.11) = 6.17\,MW \tag{117}$$

The combined cycle gas turbine power is calculated as shown in **eq. 118**

$$N_{CCGT} = N_e + N_{ST} = 7.5 + 6.17 = 13.67\,MW \tag{118}$$

3.4 Power comparison of three electric generating units

As a result of the thermodynamic calculations of three different power generating units it was obtained that the combined cycle gas turbine has maximum capacity. However, in view of impossibility to operate the given installation in the conditions of the extreme north it is offered to use combination of GTU with turboexpanders. Calculation results are presented in Table 7.

Table 7 - Comparison of the power

Electric generating unit	Power
Turboexpander	$4.37\,MW$
Gas turbine unit	$7.5\,MW$
Combined cycle power plant	$13.67\,MW$

4 Economic evaluation for the use of captive power plants

In case the decision to modernize the compressor station of the main gas pipelines is made, the issue of choosing the method of electric power supply to the facility comes up. If the object is supplied with electricity from a centralized source, its reliability is checked. In case these requirements are not met, the captive power plants are chosen as the source of power supply.

If the sources of centralized electricity supply still have sufficient reliability, it is necessary to economically substantiate the need to reconstruct the energy supply system. Before proceeding with implementation of the project it is required to economically justify the reason of transition to the alternative scheme of power supply of the enterprise. It is necessary to determine the total capital costs, including commissioning, and to compare them with the payback period of investments. In the gas industry the payback period is accepted for 8 years.

The potential investment projects are based on how much money will be invested and how much will eventually come from these investments. In the case of a compressor station, there is a question of determining an objective evaluation of the efficiency of a project to modernize the electricity supply system, i.e. the difference between the initial investment and the returnable cash flows given to date.

The process of estimating future cash flows is called discounted cash flow analysis (DCF). DCF analysis is based on the concept of the time value of money, which assumes that the amount of money today is of greater value than the amount to be received after some time.

As most financial decisions involve the estimation of projected cash flows, DCF analysis is very important. The basic idea is that any unit of money today has a greater value than the same unit of money that should or can be received after a certain period of time, because it can be invested in financial and property assets with the prospect of future additional income.

In order to calculate the efficiency of the invested money, it is necessary first to determine the cost of capital investment. The main expenditure item related to capital expenditures consists of the following components:

1. Employee salary
2. Pension and insurance contributions
3. Equipment and materials costs
4. Maintenance expenses
5. Payments for environmental pollution with sulphur oxide and other impurities
6. Taxes on gas and electricity

The first two factors can be neglected due to their relatively small value. Repair and maintenance are the next largest investments in terms of costs. Most often, the manufacturer specifies the service life of the equipment until the next major overhaul. As for commissioning, all costs are covered by the customer. The cost of the equipment and the specific capital investment can be obtained by making an inquiry directly to the manufacturer.

Air pollution charges are calculated by the specific amount of emissions per year. Components that enter the atmosphere together with the exhaust gases include:

1. Nitrogen oxide (NO_x)
2. Nitrogen dioxide (NO_2)
3. Carbon oxide (CO)
4. Carbon dioxide (CO_2)
5. Sulfur oxide (SO_x)

The calculation of payment for the negative environmental impact of associated petroleum gas flaring products within the established norms is shown in Table 8.

Table 8 - Calculation of payment for the negative environmental impact from associated petroleum gas flaring[13]

№	Name of contaminant	Emissions fee standard $/ton
1	Nitrogen dioxide	2
2	Nitrogen oxide	1.34
3	Carbon oxide	0.02

The treatment of combustion gases from nitrogen oxides is technically difficult and, in most cases, economically unprofitable. The most appropriate technology is the implementation of nitrogen oxide reduction at the stage of fuel combustion, which involves the organization of the furnace process at the lowest possible temperature in the combustion zone.

Dissociation of carbon dioxide is only possible at temperatures above $2000\ K$. Unfortunately, measures aimed at suppressing carbon monoxide formation lead to higher nitrogen oxide concentrations and vice versa. Therefore, each type of incinerator should be evaluated for the emissions of individual pollutants.

For a $1\ MW$ unit capacity, on average, $1.4\ \frac{kg}{h}$ of air pollutant emissions are generated. Fees for these emissions will be around $\$600$ per year [35], which is very low on a compressor station scale. However, a crucial factor in deciding whether a technology can be used is the amount of emissions that should not exceed the maximum allowable concentration.

In a potential investment project, cash flows will occur at different points in time. To make a useful comparison of different flows, all of them must be converted at a common point in time

[13] "Energybase.ru," [Online]. Available: https://energybase.ru/news/articles/calculation-of-fines-for-burning-associated-petroleum-gas-2020-04-16. [Accessed April 2020].

and usually at present. This process is called discounting of cash flows (DCF) and is shown in the Figure 18.

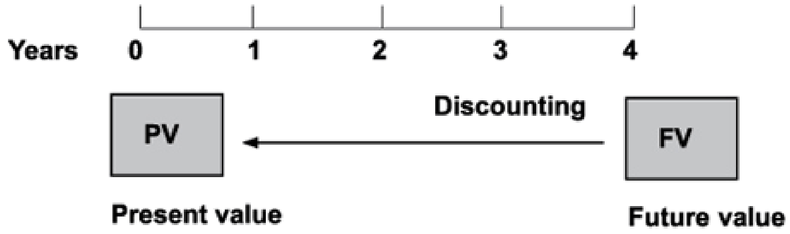

Figure 18 - Process of cash flows discounting[14]

The discounting of cash flows is based on an important economic law of diminishing value of money. In other words, over time, money loses its value compared to the current value, so it is necessary to take the current valuation moment as a starting point and all future cash receipts (profits/losses) lead to the present. For this purpose, a discount factor is used.

The discount factor is used to bring future income to current value by multiplying the discount factor and payment flows. The **eq.119** for calculating the discount factor is shown below:

$$K_d = \frac{1}{(1+r)^y} \tag{119}$$

where r is the interest rate or discount rate and y is the time in years before the future cash flow occurs.

The discounted cash flow is calculated as shown in **eq.120**

$$DCF = \sum_{y=1}^{n} \frac{CF_i}{(1+r)^y} \tag{120}$$

where DCF is discounted cash flow, CF_i is cash flow for the certain period, n is the number of time periods for which the cash flows occur.

The economic efficiency of the compressor station modernization project is evaluated using the net present value (NPV) method. The difference between cash inflows and expenditures (investments) incurred to date is called net present value (NPV). [36] It represents the sum of all discounted inflows and outflows to date of the project. The main purpose of this indicator is

[14] "Finversity.ru," [Online]. Available: https://finversity.ru/finansovaya_biblioteka/finansovyiy-menedzhment/investitsionnaya-otsenka/dcf-diskontirovannyie-denezhnyie-potoki/, . [Accessed April 2020].

to give a clear understanding of whether it is worth investing money in a particular investment project. NPV is calculated as shown in **eq 121**

$$NPV = \sum_{y=1}^{n} \frac{CF_i}{(1+r)^y} - IC \tag{121}$$

where NPV is net present value, IC – invested capital

The weighted average cost of capital (WACC) model is used in investment analysis as a discount rate in the calculation of performance indicators of an investment project. The essence of WACC is to estimate the value of equity and debt capital of a company. [37]. WACC is calculated as shown in **eq.122**

$$WACC = \frac{E}{S+A} R_e + \frac{D}{S+A} (1-\delta) R_d \tag{122}$$

where A is the total debt, S is the total shareholder's equity, δ is a corporate tax rate and in Russia equal to 20%, R_e is cost of equity and R_d is cost of debt.

The WACC method is calculated according to the balance sheet of the "Surgutneftegas" company. [38] Cost of equity is calculated as shown in **eq.123**

$$R_e = \frac{Net\ Profit}{Equity} = \frac{\$5406381906{,}67}{\$57384461053{,}33} = 9.42\% \tag{123}$$

At the next stage, it is necessary to calculate the cost of debt, which is a fee for the use of borrowed funds, in other words, the percentage that the organization pays for the funds borrowed. Cost of dept is calculated as shown in **eq.124**

$$R_d = \frac{Interest\ paid}{Total\ debt} = \frac{70044281{,}6}{3331357987} = 2.1\% \tag{124}$$

The weighted average cost of capital is calculated as shown in **eq.125**

$$\begin{aligned} r = WACC = {} & \frac{573844 * 10^5}{573844 * 10^5 + 33313 * 10^5} * 9.42 \\ & + \frac{333313 * 10^5}{573844 * 10^5 + 33313 * 10^5} * (1-0.2) * 2.1 = 9\% \end{aligned} \tag{125}$$

It should be noted that the total power consumption at the compressor station is about 10 MW. In order to provide all electricity consumers with the necessary amount of power, it was suggested to use a turbine expander in combination with a gas turbine unit. For this purpose, the model of turboexpander unit "EGU-5000" (expander-generator unit) [39] was chosen. The market value is about \$ 420,000. As it was said before, to prevent wear of turboexpander blades it is necessary to heat the supplied gas up to $80-100$℃. Therefore the "Kelvion REKUGAVO" heat exchanger [40] was chosen. The cost of this heat exchanger is \$25,000.

The model “GTA-8000” [41] was chosen as a gas turbine unit. The cost of this unit is about 2.5 million dollars. Thus, capital costs, including commissioning works, will be about 3 million dollars.

Combined operation of the turboexpander and the gas turbine unit will generate 11.87 MWh of electricity. Hence, the difference between electricity consumption and electricity production is 1.87 MWh. The cost of 1 kWh of electricity in Russia is $0.08. Therefore, the amount of money that can be earned per year is calculated as shown in **eq.126**

$$CF_1 = 1870 * 24 * 365 * 0.08 = \$1{,}310{,}496 \tag{126}$$

Further calculations of the discounted cash flow are made in MS Excel and the results are shown in Table 9.

Table 9 - Calculation of the discounted cash flow

Period (year)	Cash flow	Discounted cash flow
1	$1,310,496	$1,171,131
2	$1,310,496	$1,046,587
3	$1,310,496	$935,288.2
4	$1,310,496	$835,825
5	$1,310,496	$746,939.2
6	$1,310,496	$667,506
7	$1,310,496	$596,520.1
8	$1,310,496	$533,083.2

In order to obtain a more accurate result of the net present value, an inflation correction j of 3% should be introduced. Hence, net present value is calculated as shown in **eq 127**

$$NPV = \sum_{t=1}^{n} \frac{CF_i}{(1+r+j)^y} - IC = \sum_{t=1}^{8} \frac{\$1{,}310{,}496}{(1+9+3)^8} - \$3{,}000{,}000 = \$3{,}532{,}881 \tag{127}$$

$NPV > 0$ and that means the project is interesting to invest in and requires further analysis. Therefore, the Profitability Index (PI) will be calculated. PI is defined as the ratio of net present value to capital costs. The economic sense of this ratio is the estimation of additional value for each invested dollar. [42] PI is calculated as shown in **eq.128**

$$PI = \frac{NPV}{CC} = \frac{\$3532881}{\$3000000} = 1.177 \tag{128}$$

This means that over an eight-year period, the investment will return money that exceeds the investment by 17.7%.

5 Conclusion

The modernization of compressor station is a very complex, resource-intensive and costly process. The data analysis has shown that it is reasonable to switch compressor stations of gas main pipelines from centralized power supply to alternative sources. The reason for that is the fact that the majority of power transmission lines in Russia are outdated, and electricity prices both for legal entities and individuals are steadily growing. Thus, the majority of companies, dependent on electricity cannot operate in a normal mode due to frequent power cuts. One hour of downtime of the compressor station leads to losses of several thousand dollars, and in the monthly or even annual equivalent, the amount is estimated in millions.

To solve this problem, various options of power supply schemes for compressor stations including a turboexpander, a gas turbine unit and a combined cycle gas turbine unit were considered. As a result of calculations CCGT showed the highest capacity, but due to the territorial location of the compressor station, where most of the year the temperature is far below zero, its operation is problematic. In the CCGT calculation, processed water was used as the working fluid. For future researchers it is recommended to perform a technical and economic analysis, where the refrigerant is used as a working fluid. This will make it possible to use the CCGT in the extreme north and generate as much electricity as possible.

However, the combined use of a turboexpander and a gas turbine unit generates a capacity close to that of a combined cycle gas turbine and is 11.87 MW. Given the fact that the compressor plant consumes about 10 MW, its net power output is 1.87 MW. Economic analysis has shown that the investment of three million dollars will pay off in 8 years and will bring in an additional 17.7% of the invested amount.

On the one hand, utilization of associated petroleum gas allows to generate electricity, but on the other hand, the exhaust gases emitted during its combustion significantly pollute the atmosphere. It is also recommended for future researchers to review existing technologies to reduce emissions of carbon dioxide, carbon oxide and other harmful substances. In addition, technical and economic calculations to prove their feasibility are required. It is also necessary to carry out the analysis of electric power losses both on power lines, and on power stations of own needs. These measures will provide more accurate data for further studies.

To sum up, it should be said that measures aimed at improving energy efficiency of oil and gas companies make it possible to use hydrocarbons more rationally, which on the one hand leads to higher incomes of both organizations and employees, and on the other hand improves the environmental condition of the planet.

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Abbreviations

CCGT	Combined cycle gas turbine
CF	Cash flow
CPP	Captive power plant
CS	Compressor station
EC	Economizer
EQR	Equivalence ratio
FGTU	Fuel gas treatment unit
GCH	Gas condensate heater
GDC	Gas distribution centers
GTA	Gas turbine aggregate
GTU	Gas turbine unit
HRSG	Heat recovery steam generator
IC	Invested capital
NGV	Nozzle guide vanes
NPV	Net present value
PI	Productivity index
SH	Superheater
TEG	Triethylene glycol
VP	Evaporator
WACC	Weighted average cost of capital

Nomenclature

A	total dept [-]
a	sound velocity $[\frac{m}{s}]$
B_1	profile width of nozzle arrays [mm]
B_2	profile width of rotating blades [mm]
b_1	profile chord of nozzle arrays [mm]
b_2	profile chord of rotating blades [mm]
C_1	accurate gas discharge rate $\left[\frac{\mathrm{m}}{\mathrm{s}}\right]$
C_{1a}	axial velocity $\left[\frac{\mathrm{m}}{\mathrm{s}}\right]$
C_{1u}	radial velocity $\left[\frac{\mathrm{m}}{\mathrm{s}}\right]$
C_a	absolute velocity out of nozzle array $\left[\frac{\mathrm{m}}{\mathrm{s}}\right]$
C_{ad}	velocity corresponding to an adiabatic drop $[\frac{\mathrm{m}}{\mathrm{s}}]$
C_b	gas velocity behind the nozzle array $\left[\frac{\mathrm{m}}{\mathrm{s}}\right]$
C_{ch}	specific heat capacity of combustion products in the combustion chamber $[\frac{\mathrm{kJ}}{\mathrm{kg*K}}]$
C_{ch}	specific heat capacity of combustion products in the combustion chamber $[\frac{\mathrm{kJ}}{\mathrm{kg*K}}]$
C_i	gas velocity at the turboexpander inlet $[\frac{\mathrm{m}}{\mathrm{s}}]$
C_{pb}	specific heat capacity of air before the combustion chamber $[\frac{\mathrm{kJ}}{\mathrm{kg*K}}]$
C_{pb}^{k}	specific heat capacity of the air which is specified at the temperature of T_k' $[\frac{\mathrm{kJ}}{\mathrm{kg*K}}]$
C_{pb}^{out}	specific heat capacity of the air which is specified at the temperature of T_{out} $[\frac{\mathrm{kJ}}{\mathrm{kg*K}}]$
C_{pk}	specific heat capacity of air in the compressor $[\frac{\mathrm{kJ}}{\mathrm{kg*K}}]$
C_{pk}'	specific heat capacity at equivalence ratio equal to $E_1[\frac{\mathrm{kJ}}{\mathrm{kg*K}}]$
C_{pt}	specific heat capacity of combustion products in the turbine $[\frac{\mathrm{kJ}}{\mathrm{kg*K}}]$
$C_{pt.a}''$	actual specific heat capacity at equivalence ratio equal to E_2 $[\frac{\mathrm{kJ}}{\mathrm{kg*K}}]$
C_{pt}''	specific heat capacity at equivalence ratio equal to E_2 $[\frac{\mathrm{kJ}}{\mathrm{kg*K}}]$
C_w	heat capacity of the water $[\frac{\mathrm{kJ}}{\mathrm{kg*K}}]$
CF_i	cash flow [$]
c_2	rotated blades output velocity $[\frac{m}{s}]$
D	diameter of the gas supply pipeline [m]
D_a	average diameter of turboexpander [m]
DCF	discounted cash flow [$]
E_1	equivalence ratio equals to infinity [-]
E_2	equivalence ratio for hydrocarbon fuel combustion [-]
F_i	supply pipeline cross-section area $[m^2]$

f_{opt} optimal relative step [mm]

G gas flowrate [$\frac{\mathrm{kg}}{\mathrm{s}}$]

G_{cond} flow rate of the main condensate entering the deaerator [$\frac{\mathrm{kg}}{\mathrm{s}}$]

G_{dr} steam flow rate to deaerator from the turbine extraction chamber [$\frac{\mathrm{kg}}{\mathrm{s}}$]

G_l steam flow rate for leaks through turbine rotor and valve seals [$\frac{\mathrm{kg}}{\mathrm{s}}$]

G_{st} steam flow rate from the superheater [$\frac{\mathrm{kg}}{\mathrm{s}}$]

$\overline{H}$ compressor performance [-]

H_e specific effective work [$\frac{\mathrm{kJ}}{\mathrm{kg}}$]

H_i net work of the turboexpander [$\frac{\mathrm{kJ}}{\mathrm{kg}}$]

H_k specific compression work [$\frac{\mathrm{kJ}}{\mathrm{kg}}$]

H_k' specific compression work [$\frac{\mathrm{kJ}}{\mathrm{kg}}$]

H_{t1} specific work of the expansion in the high-pressure turbine [$\frac{\mathrm{kJ}}{\mathrm{kg}}$]

H_{t2} specific work of the expansion in the low-pressure turbine [$\frac{\mathrm{kJ}}{\mathrm{kg}}$]

H_u useful work of turboexpander blades [$\frac{\mathrm{kJ}}{\mathrm{kg}}$]

ΔH_c loss of kinetic flow energy in the nozzle array [$\frac{\mathrm{kJ}}{\mathrm{kg}}$]

ΔH_{ke} loss of kinetic energy in the rotated blades [$\frac{kJ}{kg}$]

ΔH_{ov} loss with output velocity [$\frac{\mathrm{kJ}}{\mathrm{kg}}$]

h adiabatic drop in the turboexpander [$\frac{\mathrm{kJ}}{\mathrm{kg}}$]

h_{dr} adiabatic drop [$\frac{kJ}{kg}$]

$h_{n.g}$ the available heat difference [$\frac{kJ}{kg}$]

Δh loss of energy in the nozzle array [$\frac{\mathrm{kJ}}{\mathrm{kg}}$]

IC invested capital [$]

i_o gas enthalpy before turboexpander [$\frac{\mathrm{kJ}}{\mathrm{kg}}$]

i_2 gas enthalpy behind the nozzle arrays during isentropic expansion [$\frac{\mathrm{kJ}}{\mathrm{kg}}$]

i_{bh} steam enthalpy in the extraction chamber [$\frac{\mathrm{kJ}}{\mathrm{kg}}$]

i_{cond} steam enthalpy in the condenser at the end of the expansion process [$\frac{\mathrm{kJ}}{\mathrm{kg}}$]

i_d' enthalpy of boiling water [$\frac{\mathrm{kJ}}{\mathrm{kg}}$]

i_d'' enthalpy of dry saturated steam [$\frac{\mathrm{kJ}}{\mathrm{kg}}$]

i_{de}' saturation enthalpy in the deaerator [$\frac{\mathrm{kJ}}{\mathrm{kg}}$]

i_{de}^{in} enthalpy of the main condensate at the deaerator inlet [$\frac{\mathrm{kJ}}{\mathrm{kg}}$]

i_{ec}^{out} water enthalpy after the economizer $[\frac{\mathrm{kJ}}{\mathrm{kg}}]$

i_g' enthalpy of gas before the steam superheater $[\frac{\mathrm{kJ}}{\mathrm{kg}}]$

i_g^3 enthalpy of gas after economizer $[\frac{\mathrm{kJ}}{\mathrm{kg}}]$

i_{gas}^{ec} enthalpy of gases at the outlet of the economizer $[\frac{\mathrm{kJ}}{\mathrm{kg}}]$

i_{sh}^{out} steam enthalpy behind the superheater $[\frac{\mathrm{kJ}}{\mathrm{kg}}]$

j inflation correction [%]

K_d discount factor [-]

k heat capacity ratio [-]

k' heat capacity ratio at equivalence ratio equal to E_1 $[\frac{\mathrm{kJ}}{\mathrm{kg*K}}]$

k'' heat capacity ratio at equivalence ratio equal to E_2 $[\frac{\mathrm{kJ}}{\mathrm{kg*K}}]$

k_a'' actual heat capacity ratio at equivalence ratio equal to E_2 $[\frac{\mathrm{kJ}}{\mathrm{kg*K}}]$

k_k heat capacity ratio for air [-]

k_t heat capacity ratio for flue gas [-]

l_2 height of the working blades [mm]

l_n nozzle array blade height [mm]

Δl cover of the rotating blades [mm]

M Mach number [-]

N_{ST} steam turbine power [MW]

N_e power of the gas turbine unit [MW]

N_t turboexpander power [kW]

NPV net present value [$]

n rotor speed [rpm]

O_1 final channel of nozzle arrays [mm]

O_2 final channel of rotating blades [mm]

O_1' initial channel of nozzle arrays [mm]

O_2' initial channel of rotating blades [mm]

O_{1m} middle channel of nozzle arrays [mm]

O_{2m} middle channel of rotating blades [mm]

P_{air} atmospheric air pressure [Pa]

P_{bh} steam pressure in the extraction chamber [MPa]

P_c' pressure in the condenser [MPa]

P_d steam pressure in the drum [MPa]

P_{de} pressure in the deaerator [MPa]

P_{in} inlet pressure of the turboexpander [Pa]

$P_{n.c}$ pressure under standard conditions [Pa]

P_{out} outlet pressure of the turboexpander [Pa]

P_{sh}^{out} steam pressure behind the superheater [MPa]

P_s^t steam pressure before the turbine [MPa]

PI	productivity index [-]
ΔP_{ec}	hydraulic resistance of the economizer [MPa]
Q_a	amount of the air heat entering the combustion chamber $[\frac{\mathrm{kJ}}{\mathrm{kg}}]$
$Q_{c.h}$	amount of heat transferred to the flue gases in combustion chamber $[\frac{\mathrm{kJ}}{\mathrm{kg}}]$
q_c	cooling air losses [-]
q_s	air losses through seals [-]
R_d	cost of debt [$]
R_e	cost of equity [$]
Re	Reynolds number [-]
r	discount rate [%]
S	total shareholder's equity [-]
s	gas entropy before turboexpander $[\frac{\mathrm{kJ}}{\mathrm{kg*K}}]$
T_2	output temperature of the turboexpander
T_a	average air compression process temperature in the compressor [K]
T'_a	average air compression process temperature in the compressor [K]
T_{air}	ambient air temperature [K]
T_c^s	temperature in the condenser [℃]
T_{de}	temperature in the deaerator [MPa]
T_{de}^{in}	temperature of the main condensate at the deaerator inlet [℃]
T_{ec}^{g}	economizer output gas temperature [℃]
T_{ec}^{in}	water temperature at the economizer inlet [℃]
T_{ec}^{in}	water temperature in the economizer [K]
T_{ec}^{out}	water temperature after the economizer [℃]
T_g^3	gas temperature after economizer [℃]
T_{in}	inlet gas temperature of the turboexpander [K]
T_k	air temperature after compressor [K]
T'_k	air temperature outside the compressor [K]
$T_{n.c}$	temperature under standard conditions [K]
T_{out}	temperature of combustion products before the turbine [K]
T_{sh}^{out}	steam temperature behind the superheater [℃]
T_t	temperature of combustion products behind the low-pressure turbine [K]
T_{t1}	combustion product temperature outside the high-pressure turbine [K]
t_1	relative step of nozzle arrays [mm]
t_2	relative step of rotating blades [mm]
Δt_d	underheating of the condensate to the saturation temperature in deaerator [℃]
Δt_{ec}	temperature drop in the inlet of economizer [℃]
Δt_{ss}	temperature drop in the outlet of superheater [℃]
Δt_{vap}	temperature drop in the inlet of vaporizer [℃]
u	optimum circumferential velocity $\left[\frac{\mathrm{m}}{\mathrm{s}}\right]$

V_0' volumetric gas discharge under working conditions $\left[\frac{\mathrm{m}^3}{\mathrm{s}}\right]$

V_o volumetric gas discharge under standard conditions [$\frac{\mathrm{m}^3}{\mathrm{s}}$]

$WACC$ weighted average cost of capital [%]

x_f^{opt} optimal ratio of velocities [-]

y time [year]

z_2 number of rotating blades [piece]

z_c number of nozzle blades [piece]

Greek nomenclature

α_0^{ck} skeletal angle of nozzle arrays [°]
α_1 the angle of flow outlet from the nozzle grating [°]
α_2 angle of flow outlet from the rotated blades [°]
β_1 inlet angle of the flow into the rotating blades [°]
β_1^{ck} skeletal angle of rotating blades [°]
β_2 angle of flow outlet from the rotated blades [°]
β_{2eff} efficient angle of the flow outlet from the rotating blades [°]
$\Delta\beta$ rotation angle of the flow [°]
ζ_{all} loss coefficient in the nozzle array [-]
ζ_{be} blade end losses [-]
ζ_e^{rb} end losses of rotating blades [-]
ζ_f loss coefficient in the nozzle array [-]
ζ_{fr}^{rb} friction losses in the rotating blades [-]
ζ_{lr}^{rb} loss factor for the rotated blades [-]
$\Delta\zeta_M$ correction by Mach number for nozzle arrays [-]
$\Delta\zeta_M^{rb}$ correction by M for the rotated blades [-]
$\Delta\zeta_{Re}$ correction by Re number [-]
$\Delta\zeta_{Re}^{rb}$ correction for the Reynolds number [-]
$\Delta\zeta_e$ edge losses [-]
$\Delta\zeta_{elf}^{rb}$ edge loss factor [-]
η_{high} efficiency of the high-pressure turbine [-]
$\eta_{\text{м}}$ mechanical efficiency [-]
$\eta_{c.c}$ efficiency of the combustion chamber [-]
η_{comp} efficiency of the compressor [-]
η_e absolute thermal efficiency [-]
η_{low} efficiency of the low-pressure turbine [-]
η_t relative turboexpander blade efficiency [-]
θ' degree of reactivity on the middle radius [-]
ν_1 specific volume of gas behind the nozzle grid [$\frac{m^3}{kg}$]
$\pi_{T\Sigma}$ total degree of expansion in turbines [-]
π_k compression ratio [-]
π_{t1} expansion ratio of combustion products in the high-pressure turbine [-]
π_{t2} expansion ratio of combustion products in the low-pressure turbine [-]
ρ_i gas density under standard conditions [$\frac{\text{kg}}{\text{m}^3}$]
ρ_w gas density under working conditions [$\frac{\text{kg}}{\text{m}^3}$]
ς coefficient of hydraulic resistance [-]
$\varsigma_{b.v}$ coefficient of hydraulic resistance of block valves [-]

ς_d coefficient that takes into account the loss of water pressure in the path from the economizer to the drum [-]

φ_1 velocity coefficient in the nozzle array [-]

φ_a accurate velocity factor for the nozzle array [-]

$\varphi_{s.c}$ velocity coefficient and for turboexpanders [-]

ω_1 relative gas exit velocity from the nozzle array $\left[\frac{\mathrm{m}}{\mathrm{s}}\right]$

ω_{2a} axial velocity of the rotated blades $\left[\frac{\mathrm{m}}{\mathrm{s}}\right]$

ω_{2t} theoretical relative output velocity out of rotating blades [-]

ω_{2u} radial velocity of the rotated blades $\left[\frac{\mathrm{m}}{\mathrm{s}}\right]$

ω_{rb} actual output velocity from the rotated blades $[\frac{\mathrm{m}}{\mathrm{s}}]$

δ corporate tax rate [%]

ζ hydraulic resistance along the path [-]

$\nu°$ specific volume under standard conditions $[\frac{m^3}{kg}]$

τ flow coefficient [-]

ψ accurate velocity coefficient for the rotated blades [-]

Δ_1 thickness of the nozzle arrays' input edge [mm]

Δ_2 thickness of the rotating blades' input edge [mm]

Δ_e relative thickness [mm]

Δ'_{kr} relative thickness of the output edge [mm]

Δ_{kr} thickness of the output edge [mm]

$\acute{\upsilon}_2$ flow coefficient [-]