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НАЦІОНАЛЬНИЙ АВІАЦІЙНИЙ УНІВЕРСИТЕТ
АЕРОКОСМІЧНИЙ ФАКУЛЬТЕТ
Кафедра авіаційних двигунів

ДОПУСТИТИ ДО ЗАХИСТУ
Завідувач кафедри
д-р техн.наук, проф..
____________ М.С. Кулик
“____” _________ 2020р.

ДИПЛОМНА РОБОТА
ВИПУСКНИКА ОСВІТНЬОГО СТУПЕНЯ « МАГІСТР »
ЗА ОСВІТНЬО-ПРОФЕСІЙНОЮ ПРОГРАМОЮ
“ ГАЗОТУРБІННІ УСТАНОВКИ І КОМПРЕСОРНІ СТАНЦІЇ ”
(Пояснювальна записка)

Тема: Дослідження методів ефективності роботи компресорної станції при підвищені тиску газу в магістральному газопроводі

Виконав: **Логінов А.А**
Керівник: докт. техн. наук, доцент **Ясиніцький Е.П.**

Консультанти з окремих розділів пояснювальної записки:
Охорона праці: к.б.н., доцент ________________________Коваленко В.В.
Охорона навколишнього середовища:
канд. техн. наук, доц. ____________________________Черняк Л.М.

Нормоконтролер : ___________________ /_____________________ /

КИЇВ 2020

MINISTRY OF EDUCATION AND SCIENCE OF UKRAINE
NATIONAL AVIATION UNIVERSITY
AEROSPACE FACULTY
AEROENGINES DEPARTMENT

PERMISSION FOR DEFENCE

Head of the Department
Doctor of Sciences (Engineering), prof. ___________ M.S. Kulyk
"_____" ______ 2020

MASTER'S THESIS
ON THE EDUCATIONAL PROFESSIONAL PROGRAM
"GAS TURBINE PLANTS AND COMPRESSOR STATIONS"

(EXPLANOTARY NOTE)

Theme: "Research of efficiency compressor station work methods at gas pressure increase in main gas pipeline"

Performed by: Lohinov A.A
Supervisor:
Doctor of Sciences (Engineering), Associate Professor___________
Yasenitski E.P.

Advisers:
Labor precaution :
Ph.D., Associate Professor______________________________Kovalenko V.V.

Environmental protection :
Ph.D. (Engineering), Assoc. Prof.__________________________Chernyak L. M.

Standards Inspector _________________________ /______________________/

Kyiv 2020

NATIONAL AVIATION UNIVERSITY
Faculty: Aerospace Faculty
Department: Aeroengine department
Educational degree: Master
Specialty: 142 "Power machinery"
Educational Professional Program: «Gas Turbine Plants and Compressor Stations»

APPROVED BY
Head of the Department
__________M. Kulyk
" _"_ _ _2019

Graduation Work Assignment
_____ Lohinov Antont Andrievich_____
(surname , name and patronic of graduating student)

1. **Theme: "Research of efficiency compressor station work methods at gas pressure increase in main gas pipeline"**
Approved by the Rector's order of September 26, 2019 № 2187/ст
2. **The project to be performed from 29.10.2019 to 02.02.2020**
3. **Initial data for the project: Gas turbine plant should be calculated for standard ambient conditions $p_н$ = 101,325 Pa; $T_н$ = 288 K**
4. **The content of the explanatory note (the list of problems to be considered): Introduction; Analytical part; Project part; Special part; Labor precautions; Environmental protection; General conclusion .**
5. **The list of mandatory graphic materials: presentation of results. Microsoft office Power Point, should be used to provide graphic support and presentation.**

6. Time and Work Schedule
№
Stages of Graduation Project Completion Stage Completion Dates Remarks
1. **Literature review of materials concerning theme of diploma work. Analytical part 15.10.19-31.10. 19**
2. **Project part 1.11.19-12.11.19**
3. **Special part12.11-16.11.19**

4. Labor precaution 15.12.18- 20.12. 19
5. Environmental protection 20.12. 18 - 30.12.19
6. Arrangement of graphical part of diploma work14.01. 18 -20.01.20
7. Arrangement of the explanatory note 20. 01.19- 25.01.20

7. Advisers on individual sections

Section	Adviser	Date, Signature	
		Assignment Delivered	Assignment Accepted
Labor precaution	Ph.D., Associate Professor Kovalenko V.V.		
Environmental protection	Ph.D. (Engineering), Assoc. Prof. Chernyak L.M.		

8. Assignment issue date: «_ _» _________ 2020

Diploma work supervisor: ____________________________ K.V. Doroshenko
(supervisor signature)
Assignment is accepted for execution: ___________________ A.A Lohinov
(graduate student's signature) (date)

ABSTRACT

Explanatory note to the diploma work "Research of efficiency compressor station work methods at gas pressure increase in main gas pipeline": 111 pages, 28 figures, and 6 tables 22 references.

The object of study is the gas compressor station at the main gas pipeline.
Subject of study - methods of efficiency evaluation at higher pressure in main gas pipeline.
The aim is to evaluate the optimal working conditions and research the possible usage at higher transit amount due to increased demand.
Thesis materials are recommended for use in planning the future operation of Ukrainian compressor transportation to stay as the main market player.

Key words: GAS TURBINE PLANT (GTP), THERMODYNAMIC CALCULATION, GASDYNAMIC CALCULATION, MATHEMATICAL MODEL (MM), COMPRESSOR STATION,OPERATIONAL PARAMETERS (OP), MULTITHREAD PIPELINE (MTP).

CONTENT

INTRODUCTION

Today, despite growing green energy tendency, gas as energy supply is still among the leaders, and it's extraction and consuming only increases without stop, and gas itself will be valid as energy resource for at least the next few decades.

Nowadays, according to the internal and external political, and economy factors, gas transportation system of Ukraine requeries and emergency re-management of main problems which should be displayed, and resolved in the next master plan Ukrainian Gas Transmission System (UGTS) Priority Objects Modernization and Reconstruction.

The main problems can be determined as:

- Low GTS efficiency.
- Total profitability of Gas Transit routes.
- High level of environmental pollution.
- The high risks during gas delivery to European consumers.
- Total percent of GTS equipment fatigue.

To get this major problem resolved, we need to, firstly begin a massive re-calculating of major GCS to get the actual information about equipment status, to get full view of repair and replacement procedures which must be done.

Still, to get resolved other problems, some specified actions must be done, neither than replace, or rebuild current transportation system. The most perspective way in reaching this aim is to perform modernization of UGTS in order to increase the pressure after single compressor stations to achieve a better base for further optimization, automatization, and ensure the gas supplying security.

Will be considered consider methods to achieve the work of compressor stations at elevated pressures of the main gas pipeline.

CHAPTER 1. OVERVIEW OF THE UKRAINE GAS TRANSPORTATION SYSTEM

1.1 Overview

Natural gas transportation system of Ukraine is a couple of various gas pipelines of a range of different varieties for import and transit of gas to and from Ukraine. Ukrainian GTS is one of the largest in the world and is second largest in Europe. On one side Ukrainian gas transportation system is linked to Russia where the gas is extracted and sent to a number of European countries such as Poland Romania Moldova Hungary and Slovakia. The system itself is owned by the government of Ukraine and is ran by Ukrtransgaz. Some parts of local gas distribution is owned by private companies.

1.2 Technical description

The natural gas transportation system of Ukraine consists of 38,550 km (23,950 mi) of pipelines, counting 22,160 km (13,770 mi) of trunk pipelines and 16,390 km (10,180 mi) of branch pipelines. In addition, the system consists of 72 compressor stations with 702 compressors, having a total capacity of 5,442.9 MW, and 13 underground gas storage facilities with an active storage capacity of 30.9 billion m^3 (1.09 trillion cubic feet). As of 2009, the system had import capacity of 288 billion m^3 (10.2 trillion cubic feet) and export capacity of 178 billion m^3 (6.3 trillion cubic feet) per year.

Before 2012, gas entered to Ukraine only from entry points on borders with Russia and Belarus. Most of the gas transit went to Slovakia and further to other countries in Central and Western Europe. Smaller amount of natural gas was transported to Hungary, Poland, Romania, and Moldova. In 2012–2014, some entry/exit points with Poland, Hungary, and Slovakia were modified to allow also reverse gas flow from these countries to Ukraine.

The value of the Ukrainian transmission system is estimated at US$9–25 billion. In 2004, the Ukrainian Centre for Economic and Political Studies estimated its value at $12–13 billion.

1.3 History

The development of Ukrainian gas pipeline system started in Galicia, then part of Poland. The first gas pipeline was Boryslav–Drohobych pipeline in 1912. In 1924, after discovery of the Dashava gas field the Dashava–Stryi–Drohobych gas pipeline was constructed. In 1928, the Dashava–Lviv and in 1937, the Dashava–Tarnów pipelines were built. After Soviet annexation of Eastern Galicia, the Dashava–Tarnów pipeline became the first cross-border pipeline of the USSR. The Opory–Boryslav and Opory–Lviv pipelines were built in 1940–1941.

The current Ukrainian transportation system was built as a part of the joined gas supply system of the former USSR. In 1940–1960s, it was mainly constructed to use the Galician gas in other regions of the USSR. In 1948, the Dashava–Kyiv pipeline which was the largest pipeline that time in Europe, was launched. In 1951, Dashava–Kyiv pipeline was prolonged to Bryansk and Moscow. In 1955, construction of the Dashava–Minsk pipeline started, which later was prolonged to Vilnius and Riga. It was completed in 1960. After discovery of the Shebelinka gas field in 1956, the Shebelinka–Kharkiv pipeline and the Shebelinka–Dnipropetrovsk–Kryvyi Rih–Odessa pipeline with branches to Zaporizhia, Mykolaiv and Kherson were completed in 1966. This southern corridor was prolonged to Moldova and later to Southeast Europe between 1974

and 1978. The Shebelinka–Kyiv pipeline with branches
to Poltava and Kremenchuk was completed in 1969. In 1970–1974, it was prolonged to the Western border. Also the Shebelinka–Belgorod–Kursk–Bryansk pipelines was built. In 1964, the first underground gas storage in Ukraine, the Olyshevske gas storage, was commissioned.

Figure 1.1- Gas transportation system of Ukraine.

In 1970–1980s, the Ukrainian gas transportation system was developed as a gas export route to Europe. The first large export pipeline, the Dolyna–Uzhhorod–Western pipeline, opened in 1967. It was the first stage of the Bratstvo (Brotherhood) pipeline system. In 1978, the Soyuz pipeline (Orenburg–Western border pipeline) was built as the first Soviet natural gas export pipeline. It followed by the Urengoy–Pomary–Uzhhorod pipeline in 1983 (now also named as Bratstvo or Brotherhood pipeline) and the Progress pipeline (Yamburg–Western border pipeline) in 1988. Between 1986 and 2001, the Yelets–Kremenchuk–Ananyiv–Tiraspol–Izmail route was constructed.

1.4 Natural Gas Pipelines working principles

Natural gas passes through a series of compressors, creating pressure drops - gas flows from a high-pressure region to a relatively low-pressure region. Compressors are driven by electric or gas engines, which compress or compress the incoming gas and displace it at a higher pressure. It is expected that the compressor stations for large power lines will be much larger than the compressors used to direct gas through small distribution lines to our homes. Some collection systems do not require compressors because the pressure of the gas from the wells is sufficient to pass gas through the collection lines. Natural gas is compressed in power lines to a pressure that Is typically between 500 and 1,400 psi. Power line compressor stations are typically built every 50–100 miles along the length of the power line, so that pressure can be increased as needed to maintain gas flow. Some pipelines are bidirectional, which means that gas can come from both ends of the line. Depending on where the gas is removed and where the compressors create a pressure differential, gas can flow in both directions. One example is the William Northwest Pipeline, which runs here in Bellingham. It receives gas from Canada in the north and from the Rocky Mountains region in the south. These bidirectional pipelines offer customers greater flexibility in both delivery and price. Many gas pipeline networks are "looped," which simply means that two or more lines are parallel to each other, usually in the same way. Grinding provides increased gas storage in the system to meet peak load requirements. Gas pipeline operators monitor all problems and control the gas flow through the pipeline using a monitoring, control and data acquisition system (SCADA). SCADA is a piping computer system that records information such as flow rate, operating status, pressure, and temperature. This information allows conveyor operators to know what is happening along the conveyor and allows them to respond more quickly to normal operations, device malfunctions, and discharges. Some SCADA systems also provide the ability to remotely control specific devices, including compressors and valves. This allows operators at the control center to control flow rates in the pipeline and isolate certain sections of the pipeline.

The City Gate is the place where the power grid goes into the low pressure distribution system, which delivers natural gas directly to households and companies. At the city gates, gas pressure decreases, and a perfume (usually mercaptan) is added to the gas, which gives a characteristic smell of rotten eggs so that leaks can be detected. While power lines can operate at pressures above 1000 psi, distribution systems operate at significantly lower pressures. Some pipelines (2 to 24 inches in diameter) in the distribution system can run up to 200 psi, but the small supply lines that deliver gas to individual households are usually well below 10 psi. As soon as gas is delivered to the local gas supplier at the city gates, the gas company's control center monitors the flow and pressure at various points in its system. Operators must ensure that gas reaches each consumer with sufficient flow and pressure to start the equipment. They also ensure that the pressure for each segment of the system remains below the maximum pressure. When gas flows through the system, regulators adjust the flow from higher to lower pressure. If the regulator determines that the pressure has dropped below the set point, it opens accordingly so that more gas can flow out. Conversely, if the pressure rises above a predetermined value, the controller closes for adjustment. As an additional safety function, safety valves are installed on the pipes to release gas when the pipe is under pressure and the regulators do not work properly.

1.5 Construction and aggregates of compressor stations (CS)

Gas pipeline has a number of compressor stations built along the way in order to increase the pressure and conditions of the gas to optimal for the type of the pipes in order to keep the highest speed and efficiency of gas transfer. There are 2 types of compressor used at CS reciprocating compressors and axial gas turbine compressors. Reciprocating compress are widely used at underground storage facilities. Gas turbine driven compressors are responsible for about 72% of all power of compressor stations. Due to technical and financial researches the optimal conditions and power modes are considered to be a 700-1400mm pipes and 6, 3, 10, 16, 25 kW. Axial compressors are used for compression of gas and allow both parallel and series set up. And construction allows for fast repair and replacement of rotors.

Technological scheme 1.1 displays a setup of a CS with reciprocating driven compressors. Gas enters the station from the pipeline 1, cleaned by dust collectors 2 and proceeds to gas collector 3 and to reciprocating driven compressors 6. After compression gas goes to compressed gas collectors 5 and if needed to cooler 7 or drying station 8. When gas reached optimal condition to be pumped through the pipeline it undergoes odorization process at odorization station 9 and goes to the main pipeline through the measuring sector 10.

Oil filters 4 are installed on the station. In this example of reciprocating driven compressor station compressors are connected via parallel setup which allows for each compressor to be separated into reserve if needed.

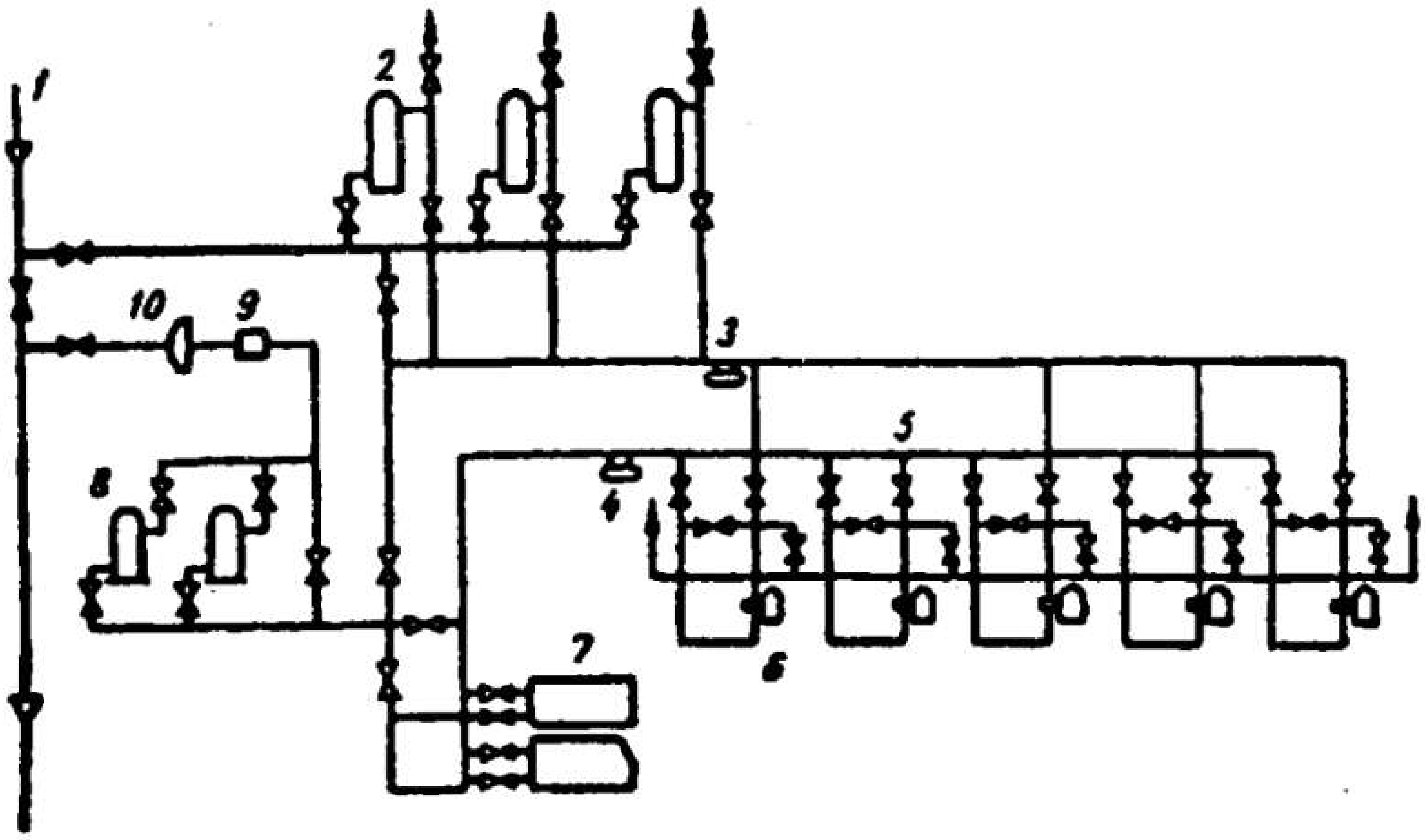


Figure 1.2- Reciprocating driven compressor station machinery.

Technological scheme 1.2 works on the principle of two stage compression. Gas from the main pipeline is delivered from the main gas pipeline through valve #7 and pass through filters to be cleaned from mechanical impurities. After the air filters it goes to oil filters and to the input collector. In the pipeline of the CS valves #1 and #2 are installed for shutting off the GTUs. Bypass valve #3 to swap between input and output of the compressors, valves to fill the inside line #4. For line cleaning #5. After compression gas is passed back to the main gas pipeline through reverse valves #6 and #8. Valve #6a is for automatic control of the outer line.

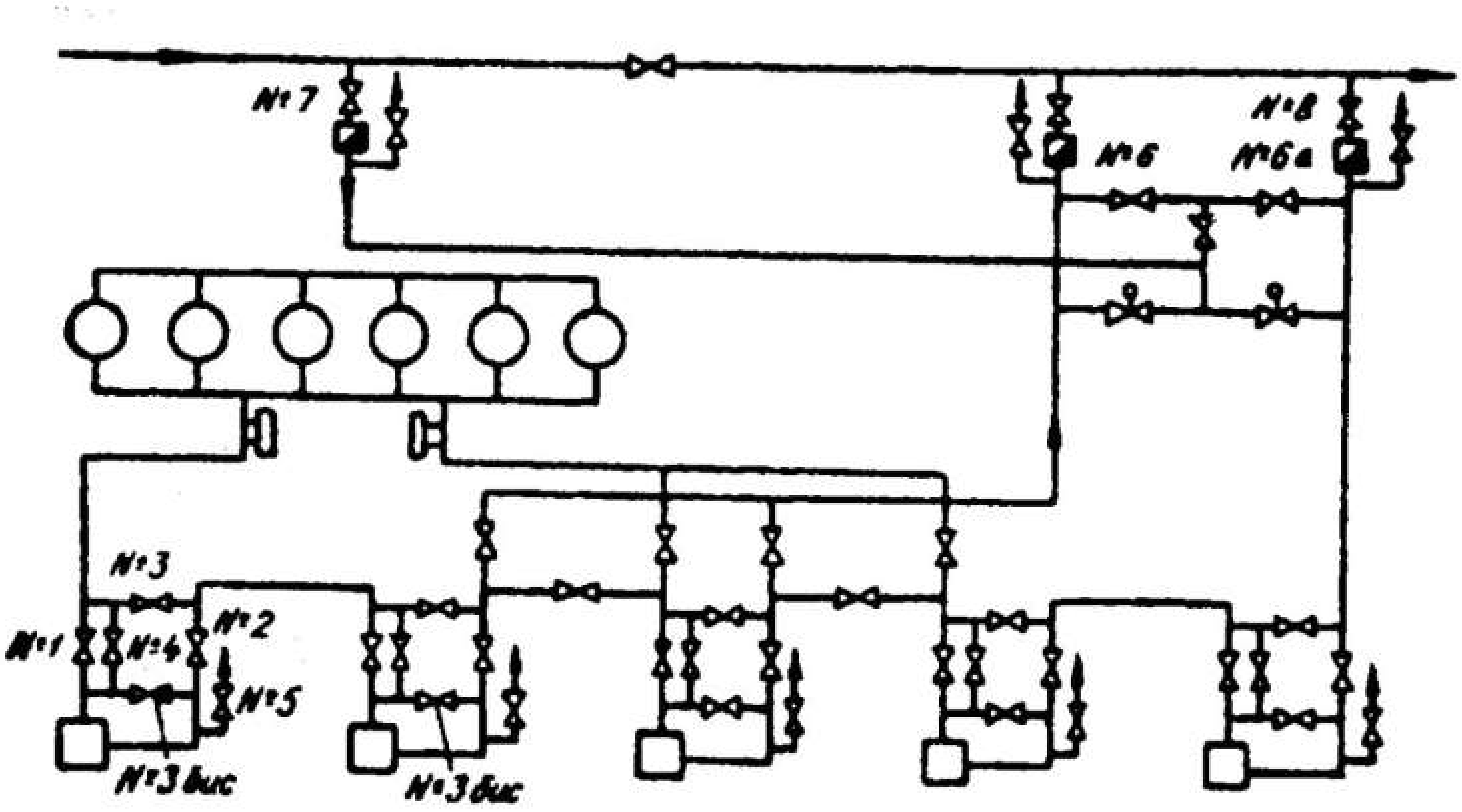


Figure 1.3 - Machinery of CS with gas turbine driven compressors.

Compressor stations of main pipeline are divided into main (MCS) and intermittent. Appliances of compressor stations are divided into technological and auxiliary.

First group is responsible for cleaning of gas from mechanical impurities, gas compression and cooling. Second group is responsible for reduction of gas pressure for fuel purposes and for CS needs. Power station of CS, heat management system, fuel and lubricant storage, repair workshop,

Gas cooling is required due to a number of different reasons. First of all during the compression process heat is exerted and is stopped within the gas, this causes reduction in efficiency of CS, increase of power needed and fuel consumption for its operation. In addition increased temperature of the gas can lead to changes in isolation integrity and its deterioration. Amount of heat supplied during the gas compression at CS is determined by the capacity of said CS, gas temperature at the inlet valve to the CS, compression ratio, ambient conditions and politropic value of compressor

During gas water cooling following apparatus are used: shelled tube, irrigation and pipe in pipe kind of apparatus. These systems not only include heat transfer elements but water-cooling elements, communication, pumps, control and monitoring systems. During gas air cooling various types of air cooling apparatus are used. Technological scheme of CS depends on the type of equipment used at CS. It includes gas lines of technological, fueling, launching, impulse and domestic gas. Technological gas communications supervise transportation of gas around the CS. They can include gas filtration systems, gas coolers, and oil filters.

1.6 Compressor station equipment exploitation.

Reciprocating driven compressor station operation. The operating mode of the compressor unit must be as close as possible to the calculated parameters. The main parameters CS operation: gas pressure at the inlet and outlet and CS throughput.

The deviation of the operating mode of the compressor unit may have influence on the following parameters: rate and hourly fuel consumption, speed and average indicator pressure (for the power unit), rotational speed and hourly output, gas pressure at the inlet and outlet and average indicator pressure (for the compressor part). The start-up of the reciprocating driven compressor station depends mainly on the following factors: air in cylinders; purity of starting air pipelines; state of starting system, ignition and fuel supply systems; water temperature in cooling system; lubricating oil temperature; state of automated systems; qualifications of staff. When starting the reciprocating driven compressor station it is necessary to ensure that there is no gas pressure in the manifolds.

Before giving fuel to the power cylinders must be purged to remove explosive mixture. To ensure a reliable start of the mining and metallurgical complex it is necessary: heat up the unit before starting the circulation of water in the cooling system; heat up lubricating oil; maintain of necessary pressure in the air tanks for starting (1.7 MPa).

When fuel is burned in the engine cylinders due to friction, a large amount of heat is released, which must be removed with a cooling system that provides: heat removal from parts and nodes, lubricating oil cooling, air or gas cooling, in the CS.

In GMK, lubricating oil is designed to create a liquid rubbing in pairs and removal of excess heat. reliability and durability of the unit highly depends on properly serviced and reliable lubrication system

The CS fuel supply system includes: gas supply elements, installations, instruments for regulating and monitoring the state of gas in the power system, installations, units and devices for the preparation and supply of gaseous combustible mixture in cylinders. The following requirements are applied to the power system: fuel supply and its mixing with air to the end of compression should ensure the creation of a uniform gas mixture throughout combustion chamber volume; the amount of fuel entering the cylinders per each cycle should correspond to the amount of air filling cylinder; the amount of gas inlet flowing into different cylinders unit for one cycle during the entire period of operation in this mode, should be the same in size and composition; when changing load the supply of the required amount of fuel should change automatically.

Operation of gas turbines. Successful operation of a gas turbine depends on its design. instructions, quality of installation or revision. The reliability of the machine, power and efficiency are more dependent on the quality of assembly of gas turbine units. When assembling, it is necessary to strictly follow designed clearances. With an increase in the gap along the flowing parts rapidly decreases efficiency and power of the unit.

During the operation of gas turbines, most of the nodes are exposed to high temperatures. To compensate for temperature elongations provided linear compensators, and - free movement of electric cops of the unit along the supports. The level of thermal conditions of the units is provided by air cooling system.

Before starting procedures, it is necessary to undergo thorough stress testing of pipelines, check of fittings, control units cranes, cable communications testing, electrical inspection, instrumentation, control panels and automation. In preparation for launch from cold conditions, an external inspection and inspection of the new and auxiliary equipment. After a general check, the oil system threads and circulating water supply check oil pumps and oil supply to bearings. It is necessary to check for leaks in the oil system and piping.

The sequence of operations during startup of the unit and its launch to minimum load conditions is determined by design features of supercharger turbines, accessories and shutoff valves in accordance with the manufacturer's instructions. Great value for normal and trouble-free operation of the automatic control system and protection of bearings and other components are high oil quality and cleanliness.

Basic principles of operation of the unit and automation system for all types of gas turbines remain unchanged and differ in constructive design, but, despite this, for each type of gas turbine there are following rules and instructions.

When servicing the unit, any indication of the device that does not comply with valid parameters, serves as a signal of a malfunction either an assembly unit or system. In normally working unit, the differential of the oil temperature at the inlet and outlet of the bearings must not exceed 283-286 K. It is necessary to take an oil sample to analyze its purity at the start of every working shift and systematically check the individual components of the unit for vibration.

1.7 Reasons for research

For my research paper I chose to investigate the effects of increasing overall pressure in the main gas pipeline in order to find out the effects of it on the gas transfer system of Ukraine. In my opinion this would be the most effective and the best future solution for the current state of gas transport system due to a number of reasons. To start with the amount of turbine driven pumping units operating past their designed working hours is around 70%. This means that failures are a constant possibility and liability for the system. Increasing overall pressure means that optimal conditions for achieving the needed pressure of gas for the system will be reached by lower number of machines meaning that extra machinery can be used as reserve or salvaged for parts to replace worn out parts of worn out machines causing reduction in failure rate and higher efficiency of the units due to improved conditions. This would be a way more economically affordable way of solving the problem of machines that worked past their designed service life. New engines can cost around 2.5 million dollars and up depending on required characteristics, taking in consideration that each CS has at least 10 such machines the price of modernist on would be around 25$ million which sounds impossible considering the number of such station we have running along the transportation system of Ukraine. In addition to the cost of realign the machinery of the station it would also take enormous amount of down time of the CS and extra cost and time of insulation, test runs and changes in infrastructure of the CS as well as time and effort required to teach ad educate personal of the CS to work with the new equipment.

In addition to the techies problems with our pipeline there are a number of political factors that hinder the future of Ukraine as the main gas transit country between Russia and Europe due to a number of projects by the Russian government such as TurkStream and Nord stream. There are the new gas pipelines that specifically go around Ukraine to deliver gas to Europe as seen in figures 1.4 and 1.5

Figure 1.4 - Nord stream gas pipeline.

Nord Stream 1 is supplied via the Gryazovets-Vyborg gas pipeline. It is part of the uniform gas transmission network in Russia and connects the existing network in Gryazovets with the coastal compressor station in Vyborg. The length of this pipeline is 917 kilometers, the pipe diameter is 1,420 millimeters and the operating pressure is 100 normal atmospheres (10 MPa), which are provided by six compressor stations. The Gryazovets-Vyborg pipeline also supplies gas to the northwestern region of Russia (St. Petersburg and the Leningrad region) in parallel to the Northern Lights pipeline branch. The pipeline is operated by Gazprom Transgaz St. Petersburg.

To supply Nord Stream 2, a new pipeline with a length of 866 kilometers and three compressor stations were built and five existing compressor stations were expanded. The supply line starts in Gryazovets and follows the existing Northern Lights pipeline route. In Volkhov, the gas pipeline turns south and leads to the Slavyansk compressor station in the Ust-Luga region.

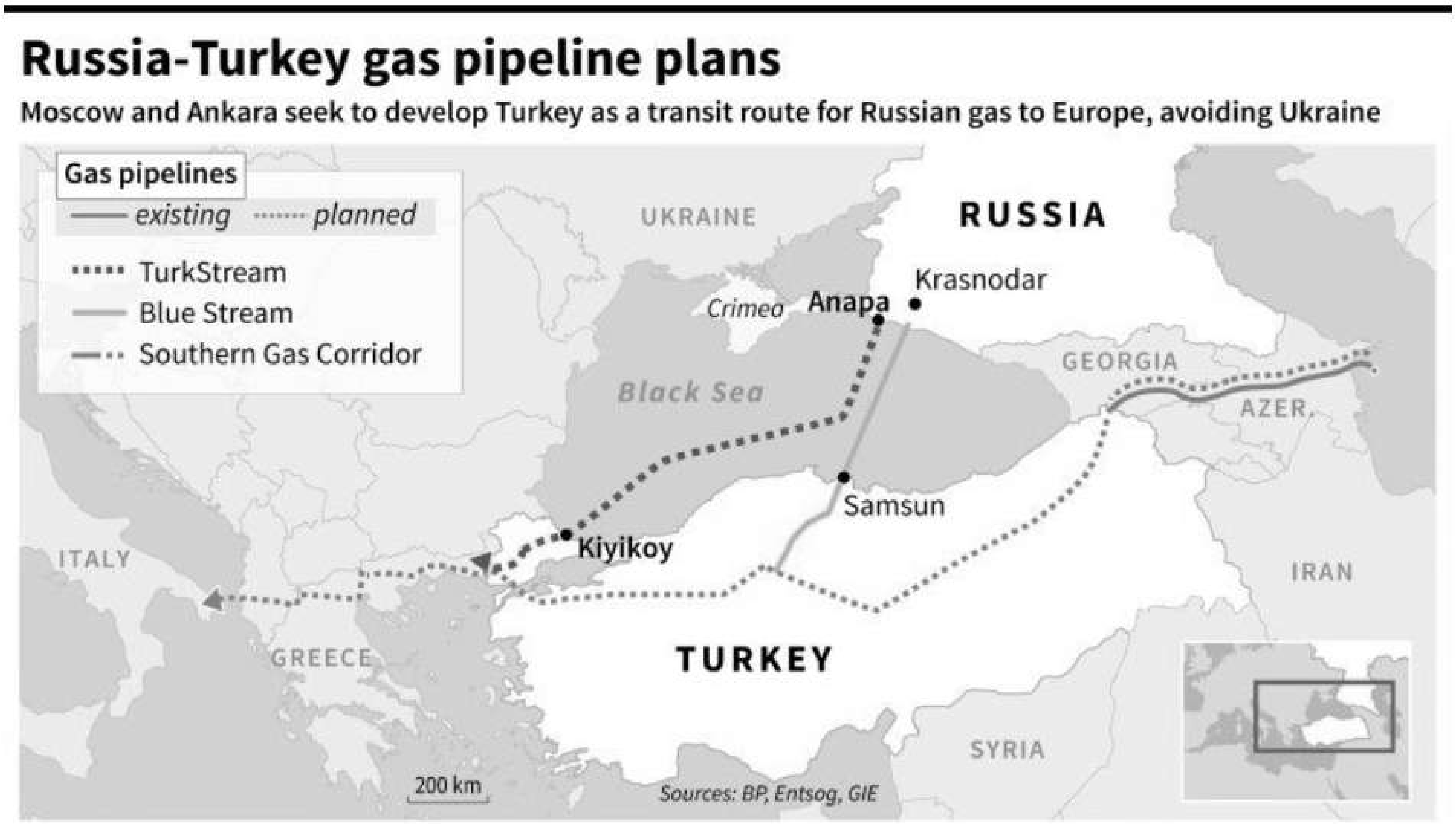


Figure 1.5 - Turk stream gas pipeline

Recently on 8th of January Turk stream was inaugurated this is another pipeline that goes around Ukraine. The cost of the pipeline is estimated at 11.4 billion euros. The pipeline has two pipes with a total throughput of 31.5 billion m3 / year (1.11 trillion cubic feet / year) of natural gas. The first line supplies Turkey and the second line transports natural gas to Southeast and Central Europe. Both lines use pipes with an outer diameter of 810 mm (32 inches), which are manufactured by the German company Europipe GmbH, the Russian Vyksa Metallurgical Plant OMK and the Russian Izhora Pipe Plant Severstal, as well as the consortium Marubeni, Itochu and Sumitomo from Japan. The pipes have a wall thickness of 39 mm and a concrete coating of 80 mm. The internal pipe pressure is 300 bar (4,400 psi). The pipeline will be laid at a depth of up to 2,200 m. Due to these reasons it is safe to assume that the demand of Ukrainian gas transportation system will be lowered due to other more modern and state of art gas pipelines that have higher transit rate and lower chance of failure which will ensure the that all the gas will be delivered in time and without any hiccups. Another reason is that is that the world nowadays is moving more towards usage of renewable energy and many researches are being funded in this direction. Use of renewable energy is also encouraged by the governments and International community. In 2009, the Renewables Directive mandated EU member states to ensure that by 2020, twenty percent of European energy consumption comes from renewable sources. As a result, member states created national-level support schemes for renewables. This causes difficulty when energy flows across national borders, and despite legal uncertainties caused by the present situation, subsidy schemes mostly remain in national silos. To avoid confusion, we should move towards EU-wide subsidies. As a result of all these factors and predictions it is safe to say that the demand for Ukrainian gas transportation system will be falling every year meaning that there would be no need to keep it operational at the current level.

1.8 Ukraine as gas storage

Another viable direction of Ukrainian GTS development id as a storage facility for further distribution around Europe. This is a viable possibility due to a fact that Ukraine has the biggest gas storage Capacities in Europe. One of the problems with Ukrainian GTS is that we cant supply a needed amount of gas to the final consumer in peak times with the flow rates we provide. But if we would be depositing gas in the storage facilities during low demand time we would have enough gas to gradually transfer it to the European consumers without wasting as many resources on domestic purposes.

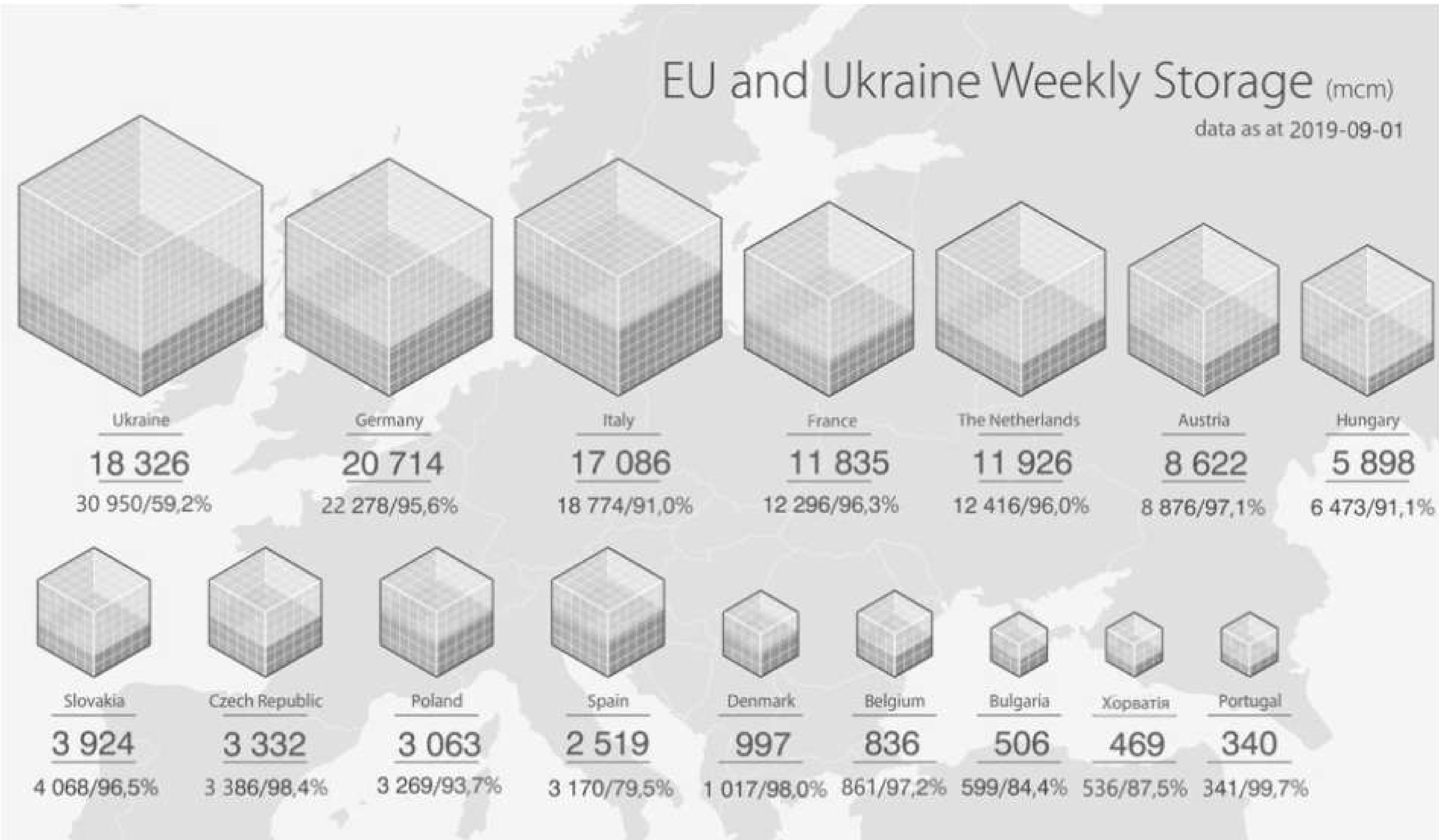


Figure 1.6 - Gas Balances in European Underground Storages

From figure 1.6 we can see that Ukraine has the largest gas storage facilities in Europe by a large margin and the least used by far. In addition to this we can see from figure 1.7 that the largest storages are right at the boarder with the Europe.

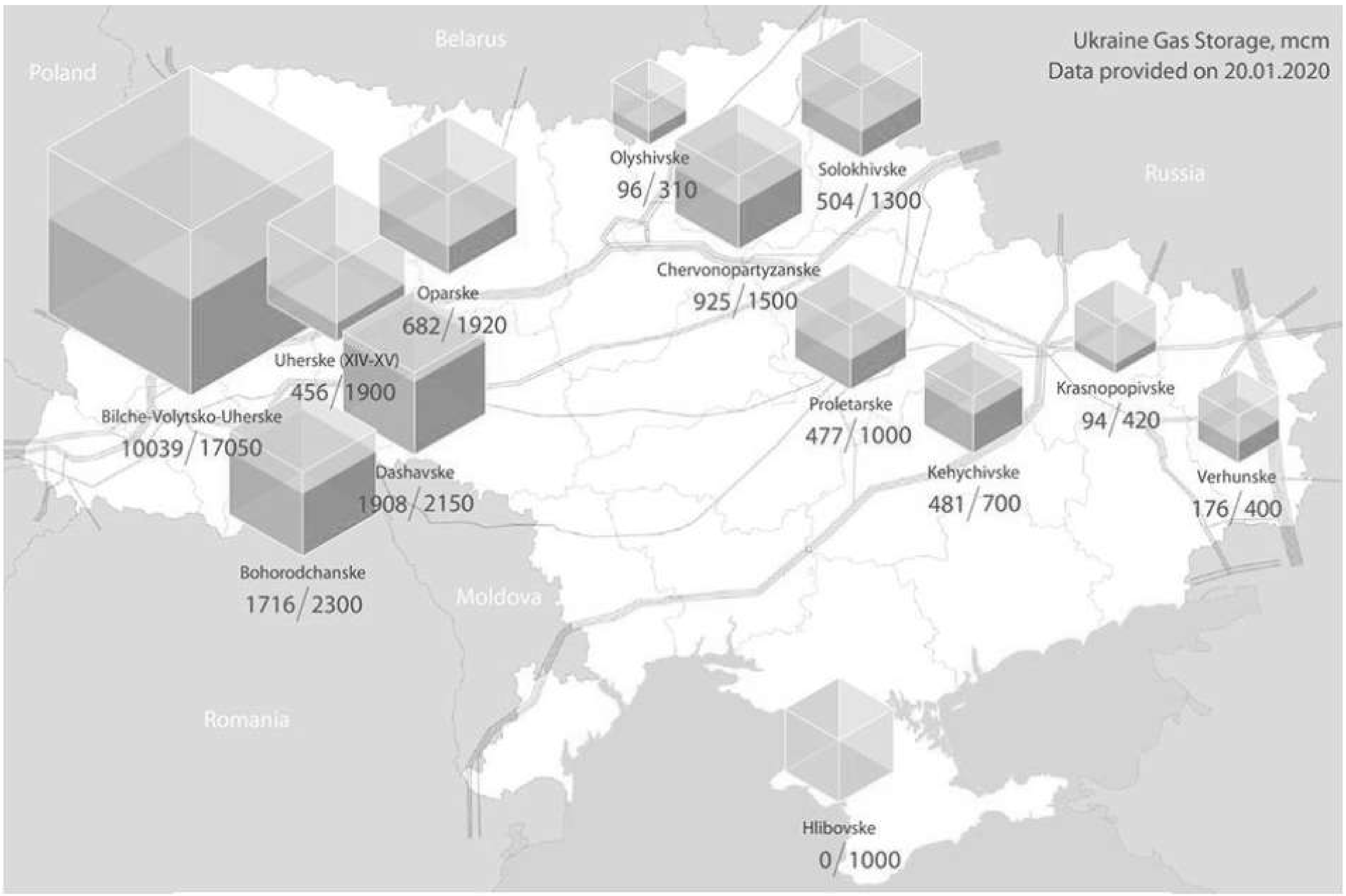


Figure 1.7 - Gas Balances in Ukrainian Underground Storages

1.9 current state of gas transportation of Ukraine

In recent years, in the gas transportation system of Ukraine has serious problems. The first of these is the fact that over 70% of the gas turbine driven compressor unit is almost worn out (100 thousand. Hours). Between 2008 and 2015, the planned replacement of 204 GTE: 2008 - 20 2009 - 23 2010 - 25 2011 - 26 of 2012 - 28 2013 - 29 2014 th - 26 and 2015 - 27 engines. Another problem is the low efficiency (efficiency) old GTE, which is only 18-25%. Today, most key indicators maintained GTE lower foreign engines of similar capacity which have a full resource of 150 thousand. Hours and efficiency at the level of 34-38%. Out of this situation is to prompt the development of a new generation of Ukrainian gas turbine engine, intended for use on the GTS Ukraine. The most important qualities of GTE in order of ranking is the safety and reliability for long term use, high efficiency (efficiency), the relative simplicity of construction and operation, low cost life-cycle. Industrial gas turbine engines for the gas transportation system from the very beginning should be created specific to their operation on gas pipelines. This high performance can be achieved through the use of advanced turbine technologies used in aviation and marine gas turbine engines for military purposes.

In addition to problems with the power machinery we have a problem with the main pipeline. Overall our system consists of 38 550 kilometres of pipelines including 22 160 kilometres of trunk pipelines and 16,390 kilometres of branch pipelines. Most of which have worked way over their designed exploration period which effects its integrity and operation capacity and highly increases chances of failures which can cause delays and even ecological disasters. To prevent this the system is equipped with a number of safety system along the way of the pipeline which include measuring stations along the pipeline in order to detect abnormal pressure which may indicate failure or breach in the pipeline. As well as measuring the sunset visual observation of the pipeline can be useful to detect gas leaks, in winer it is possible to detect gas leaks due to patches of melted snow along the pipeline and in summer grass discoloration will indicate a leak.

Ukraine has significantly increased gas purchases in Europe in 2019 Naftogaz has not procured natural gas from Russia for more than 1,500 days.

Ukrainian gas companies have stepped up gas purchases from Poland, Hungary and Slovakia in 2019 by 34% year-on-year to 14.2 billion m3. The most significant increase in pumping this energy source from Slovakia was 42%, to 9.1 billion m3.

Gas imports to Ukraine from Poland last year doubled, rising to 1.4 billion m3, and from Hungary - up 7% to 3.7 billion m3.

However, for the sake of Naftogaz or other importers, Ukraine has not been buying natural gas from Gazprom for more than 1,500 days (since November 26, 2015). Ukraine only accepts Russian gas from Gazprom for transportation to Europe, and receives volumes that have passed through Ukraine's gas transmission system in reverse from neighboring EU countries (Slovakia, Poland and Hungary).

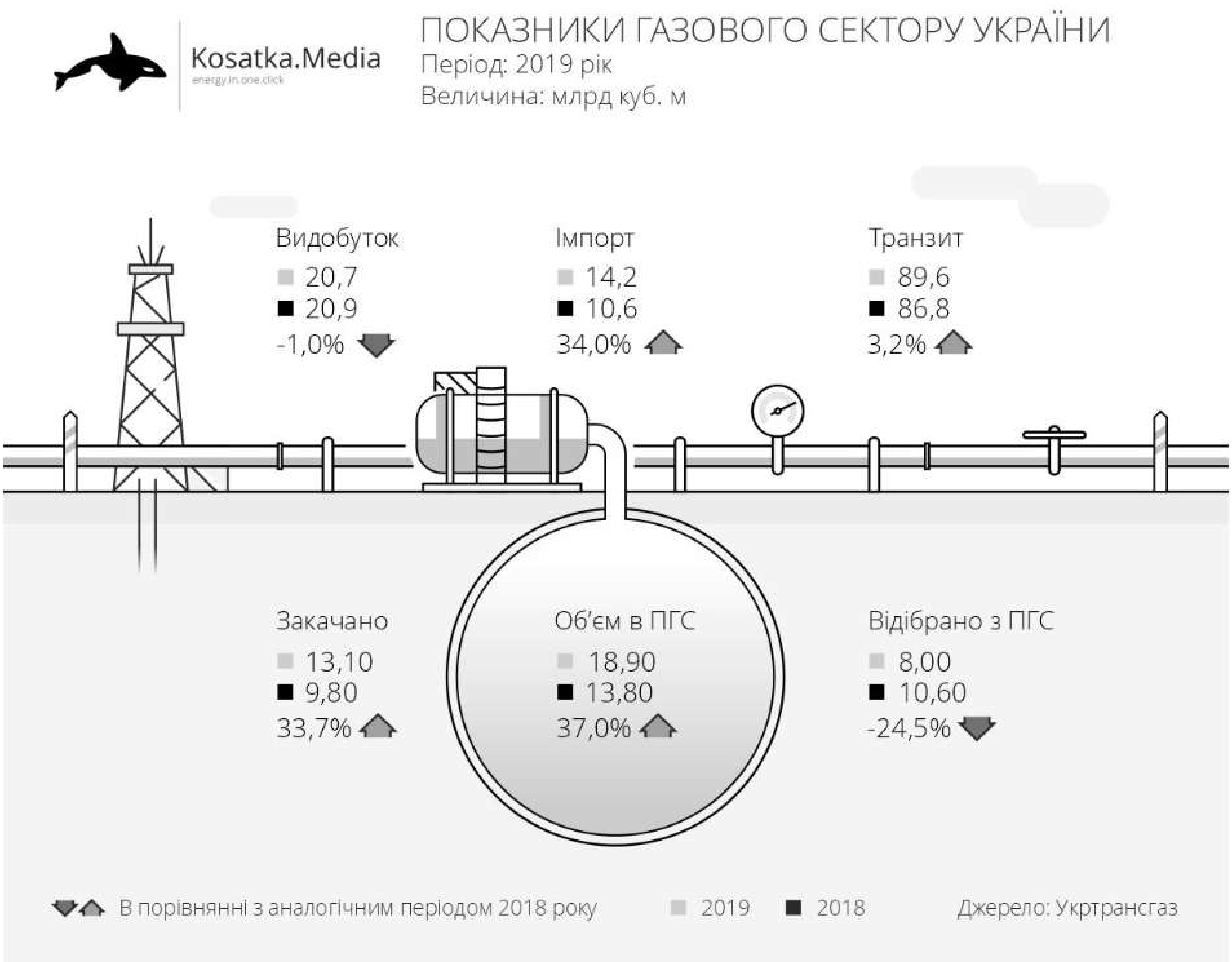
At the end of December and the first days of January, there were reports that some Ukrainian companies would be able to buy natural gas in Russia to provide their businesses without resale rights, but no agreements were reached and agreements were reached.

The volumes of gas purchased by Ukrainian importers in the EU countries were mainly used for pumping into underground gas storage and for use in the heating season. A total of 13.1 billion m3 of gas was pumped into the UGS of Ukraine in 2019, which is 33.7% more than in 2018.

At the same time, gas consumption decreased by 7% to 26.4 billion m3 last year, and gas production and processing costs amounted to 3.4 billion m3, 8% less than in 2018. The decline in consumption in the country is due primarily to the relatively warm weather in the autumn and winter months of late 2019 and, to a lesser extent, energy-saving and energy-efficiency measures, which also bear fruit.

Natural gas transit from Russia through the territory of Ukraine in 2019 has increased by 3% compared to 2018, to 89.6 billion m3. European gas consumers have significantly increased their purchases of energy from Gazprom in order to fully fill gas storage facilities in the face of a gas standoff, which could end in a transit through Ukraine through January 1, 2020, but fortunately for all parties, this has been avoided.

Gas production by Ukrainian gas companies (both public and private) decreased last year by 1% or 200 million m3 to 20.7 billion m3. First of all, the volume of energy production was reduced by the largest producer - the state-owned company Ukrgazvydobuvannia.



Conclusion

1. The Ukrainian Gas Transport System is able to transport gas volumes of 120 bcm per annum. On average, 80% of Russian gas exports to Europe pass via the Ukrainian system. This equates to around 20% of the EU's total gas demand. Ukraine uses the same pipeline system to transport gas for domestic use. These facts demonstrate that the impeccable functionality of the Ukrainian system is in the common interest of Naftogaz Ukraine, Gazprom, as well as Ukrainian and European consumers.
2. Ukraine transits roughly the same quantity of gas destined for the EU as it consumes in its national market (around 300 mcm / day in winter). During winter, the capacity of the Ukrainian system is insufficient to cope with both the immense domestic demand in Ukraine and with transit volumes to Europe. To solve this problem, Gazprom and Ukraine perform a gas swap.
3. In recent years, GTS of Ukraine has faced with serious problems, the main of which is connected with the fact that more than 70 % GTU with gas turbine drive is almost worn out (100 thousand hours), and some has worked for more than 150 thousand hours. Another problem is the low efficiency of obsolete GTPs, which is only 18...27 %. Modern level of engine manufacture allows to increase the efficiency of the engine therefore engines that are currently in service can be upgraded with a view to increase their efficiency and the output pressure. topic shows the possibility of UGTS to stay as a serious market player, with using of modern technologies, and ensure it's reliable work and profit for the country.

CHAPTER 2. METHODS OF ENSURING AN EFFECTIVE WORKFLOW ON PROMISING GAS PIPELINES

2.1 Gas Compressor Station Complex Issues and their Optimization.

Nowadays, according to the internal and external political, and economy factors, gas transportation system of Ukraine requeries and emergency re-management of main problems which should be displayed, and resolved in the net master plan Ukrainian Gas Transmission System (UGTS) Priority Objects Modernization and Reconstruction.

The main problems can be determined as:

- Low GTS efficiency.
- Total profitability of Gas Transit routes.
- High level of environmental pollution.
- The high risks during gas delivery to European consumers.
- Total percent of GTS equipment fatigue.

To get this major problem resolved, we need to, firstly begin a massive re-calculating of major GCS to get the actual information about equipment status, to get full view of repair and replacement procedures which must be done.

Still, to get resolved other problems, some specified actions must be done, neither than replace, or rebuild current transportation system.

Such methods are: analytical methods of optimization of operational parameters of the main gas pipelines, Optimization of Fuel Consumption of Compressor Station engines, and station optimization analysis.

2.1.1 Capital Cost: First Cost and Installation Cost

Capital cost for a project consists of first cost and installation cost. First cost includes not only the cost for the driver and compressor, and their skid or foundation, but also the necessary systems that are required for operating them, including filters, coolers, instruments, and valves, and, if reciprocating compressors are used, pulsation bottles. Capital spares, operational spares, and start-up and commissioning spares also have to be considered.

Although not intuitively obvious, this is also the area that is affected by driver derates due to site ambient temperature and site elevation: the power demand of the compressor has to be met at site conditions, not at ISO or NEMA conditions.

Installation cost includes all labor and equipment cost to install the equipment on site. It is determined by component weights, as well as the amount of operations necessary to bring the shipped components to working condition.

2.1.2 Maintenance Cost

Maintenance cost includes the parts and labor to keep the equipment running at or above a certain power level. This includes routine maintenance (like change of lube oil and spark plugs in gas engines) and overhauls. Maintenance events can be schedule or condition based. A cost related to maintenance effort is the cost due to the unavailability of the equipment (see below). Maintenance affects availability in two ways. Many, but not all, maintenance events require the shut down of the equipment, thus reducing its availability. Scheduled maintenance has usually less of an effect than unscheduled events. For example, a scheduled overhaul of a gas turbine may keep the equipment out of operation for only a few days if an engine exchange program is available.

On the other hand, insufficient or improper maintenance negatively affects the availability due to more rapid performance degradation and higher chance of unplanned shutdowns.

2.1.3 Efficiency, Operating Range, and Fuel Cost

The performance parameters of the compressor and its driver that are important for the economic evaluation are efficiency and operating range. Efficiency ultimately means the cost of fuel consumed to bring a certain amount of gas from a suction pressure to a discharge pressure. In technical terms, this would be a package with a high thermal efficiency (or low heat rate) for the driver and a high isentropic efficiency of the compressor, including all parasitic losses (such as devices to dampen pulsations, but also pressure drop due to filtration requirements) combined with low mechanical losses. This factor determines the fuel cost of the unit, operating at given operating conditions.

Operating range describes the range of possible operating conditions in terms of flow and head at an acceptable efficiency, within the power capability of the driver. Of particular importance are the means of controlling the compressor (e.g., speed control for centrifugal machines, or cylinder deactivation, clearance control, and others for a reciprocating machine) and the relationship between head and flow of the system the compressor feeds into. Operating range often determines the capability to take advantage of opportunities to sell more gas. It should be noted that there is no real low flow limit for *stations*, due to the conceptual capability for station recycle or to shutting down units. Unit shutdown, in turn, has to be considered with regards to starting reliability of the units in question, as well as the impact on maintenance. In this study, operating range and the upside potential are not specifically considered, mostly because no data regarding frequency and value of these upside potentials was available. Upside potential can be realized, if the equipment capability can be used to produce more gas, thus taking advantage of additional market opportunities.

The cost of the fuel gas is not automatically the same as the market price of the transported gas. It depends, among other things, on how fuel usage and transport tariff are related. The cost of fuel is also impacted by whether the operator owns the gas in the pipeline (which makes fuel cost an internal operating cost) or the operator ships someone else's gas. In some instances, the fuel cost might be considered virtually nil. Thus, the ratio between fuel cost and maintenance cost can vary widely. In most installations, however, the fuel cost may account for more than two-thirds of the annual operating cost.

The value of operational flexibility is somewhat hard to quantify, but many pipeline systems operate at conditions that were not foreseen during the project stage. Operational flexibility will result in lower fuel cost under fluctuating operating conditions and in added revenue if it allows to ship more gas than originally envisioned.

2.1.4 Emissions

Any natural-gas-powered combustion engine will produce a number of undesirable combustion products. NOx is the result of the reaction between Nitrogen (in the fuel or combustion air) and oxygen and requires high local temperatures to form. Lean premix gas turbines and lean premix engines reduce the NOx production. Catalytic exhaust gas treatments, such as SCR's, can remove a big portion of NOx in the exhaust gas, but also add ammonia to the exhaust gas stream.

Products of incomplete combustion include VOC's, CO, methane, and formaldehyde. Fuel bound sulfur will form SOx in the combustion process.

The combustion products above are usually regulated. For the economic analysis, the cost of bringing the equipment to meet local or federal limits has to be considered.

CO_2 is the product of burning any type of hydrocarbons. CO_2, and some other gases, such as methane, are considered greenhouse gases. Typically, all greenhouse gases are lumped together into a CO_2 equivalent. In this context, it must be noted that methane is considered about 20 times as potent a greenhouse gas as CO_2. Thus, 1 kg of methane would be considered as about 20 kg CO_2 equivalent. In some countries, CO_2 production is taxed, and there is a possibility that other countries, including the USA, adopt regulations that would give CO_2 avoidance an economic value. In this case, the amount of CO_2 or methane that is released to the atmosphere has to be considered as a cost in the economic evaluation.

It further needs to be considered that the engine exhaust is not the only source of emissions in a compressor station related to the compression equipment. There are also sources of methane leaks in the compression equipment that may have to be considered. In this case, one has to distinguish leakage that is easily captured and can thus be fed to a flare and leakage that cannot be captured easily. Further, it may have to be evaluated how frequently the station has to be blown down. For example, whether the compression equipment can be maintained and started from a pressurized hold determines the amount of unwanted station methane emissions.

Lastly, other consumables may have to be considered. The frequency and cost of lube oil changes, as well as lube oil replacements due to lube oil consumption, generate costs on various levels: first, the replacement cost for lube oil, second, if the lube oil is used in the combustion process, the resulting emissions, and third, if the lube oil enters the pipeline, the cost due to the pipe contamination, including possibly the increased maintenance cost of downstream equipment.

2.3.5 Availability

Availability is the ratio between the hours per year when the equipment is supposed to operate and actually can operate and the hours per year where the equipment is supposed to operate. Availability therefore takes into account the entire equipment down time, both due to planned and unplanned maintenance events. In other words, if the operator needs the equipment for 8760 hours per year, and the equipment requires 3 shutdowns, lasting 2 days each for scheduled maintenance, and in addition also had to be shut down for 3 days due to a equipment failure, the availability of the equipment would be 97.5%. Other than MTBF, which describes the frequency of unplanned failures, the availability has a direct impact on the capability of an installation to earn money. Besides the type of equipment, the quality of the maintenance program and the measures taken to deal with environmental conditions (air, fuel, etc.) have a significant effect on the availability (and reliability) of the equipment.

The cost associated with availability is the fact that the station may not be able to perform its full duty for certain amount of time, thus not earning money. The loss of income can be due to the reduction of the pipeline throughput, the unavailability of associated products (oil on an oil platform, condensates in a gas plant), or due to penalties assessed because contractual commitments are not met. The value associated with the lost production is not necessarily the market value of the lost production. It may also be the loss of income from transportation tariffs, the cost from penalties for not being able to satisfy delivery contracts, or the cost for lost opportunity (i.e., due to the requirement to keep spare power to compensate for poor availability, instead of being able to use the spare power to increase contractually agreed deliveries).

Station availability can be improved by installing spare units, but this is an additional first cost factor.

2.2 Analytical methods of optimization of operational parameters of the main gas pipelines

The task of optimization of main gas pipelines is multiparametric. It is reduced to the analysis of nonlinear equations systems. The obtained analytical correlations allow carrying out the analysis of optimal solutions, depending on the definition of the task and the imposed conditions.

2.2.1 Modeling of compressor station operation modes

The model of the compressor station (CS) is formed on the basis of the technological scheme structure model and its functional objects models, which directly affect the value of the operational parameters. The structure model is represented as a graph in which objects that have a length are represented by edges, and all others are vertices. In the main object (gas-pumping unit (GPU)) two main components are distinguished (drive and centrifugal supercharger (CeS)). In this work, gas turbine type drives are considered. It is well-known [12] that the gas parameters at the input and output of the CeS are related via a set of empirical dependencies

$$\varepsilon = \varphi_1\left(q|_{pr}, \left[\frac{n}{n_n}\right]_{pr}\right), \quad \eta_{pol} = \varphi_2\left(|q|_{pr}\right), \quad \frac{N_i}{\gamma_n}\left(\frac{n_n}{n}\right)^3 = \varphi_3\left(|q|_{pr}\right).$$

$$N_e^p = N_e^n K_{Ne}\left(1 - K_t\frac{t_0 - t_0^n}{t_0 + 273}\right)\frac{p_a}{0.1033}, \quad \left[\frac{n}{n_n}\right]_{pr} = \frac{n}{n_n}\sqrt{\frac{z_{pr}R_{pr}T_{pr}}{zRT}}, \quad [q]_{pr} = \frac{n_n}{n}q.$$

The amount of fuel gas is based on the formula

$$q_p = q_p^n K_t\left(0.75\frac{N_e}{N_e^n} + 0.25\sqrt{\frac{t_0 + 273}{t_0^n + 273}\frac{p_a}{0.1033}}\right),$$

Where:

$$q_p^n = \frac{860 N_e^n}{\eta_e^n Q_n 10^3}, \quad N_e = N_i : (\eta_m K_N),$$

n is the CeS rotation speed, q is the CeS gas consumption, η_{pol} is the CeS polytrophic efficiency, q_p, $q_p{}^n$ is the real and nominal consumption of fuel gas, ε is the pressure differential, N_e^n is the nominal power of the gas turbine unit (GTU); K_{Ne} is the coefficient of technical state of gas turbine unit (GTU); K_t is the coefficient, which takes into account the influence of temperature of atmospheric air; t_0 is the GTU entry air temperature;t_0^n is the nominal GTU entry air temperature; p_a is the absolute pressure of atmospheric air, depending on altitude above sea level H; t_0 is the GTU entry air temperature (°C), N_i is the internal CeS power; Q_n is the nominal lower specific volumetric heat of fuel combustion, $\eta_e^n$is the nominal GTU efficiency, η_m is the mechanical efficiency, K_N is the technical state by power, z_{pr}, R_{pr}, T_{pr} are gas parameters at which the characteristics of the supercharger are experimentally determined; γ_c is the specific weight of gas at standard conditions (P = 0.1033MPa; T = 293K); n_n is the nominal CeS rotation speed, $\phi_k\,(k = 1 - 3)$ are empirically determined functions.

The given characteristics allow considering: the deviation of gas parameters at the entry of the supercharger (z_{2E},R,T_{2E}) from their reduced values (z_{pr},R_{pr},T_{pr}), the deviation of the actual speed n of the supercharger from its nominal $n_n$value.

The value of polytrope is from the ratio$T_{out}z_{out} - T_{in}z_{in}\varepsilon^{\frac{k-1}{k\cdot\eta_{pol}}}$ and specified using, with known

ones ε and n, the formula:

$$\varepsilon^{\frac{k-1}{k\eta_{pol}}} = \left(\frac{n}{n_n}\right)^2 \frac{z_{pr}T_{pr}R_{pr}}{z_{in}T_{in}R}\left(\varepsilon_n^{\frac{k-1}{k\eta_{pol}}} - 1\right) + 1 .$$

To calculate the internal power of the centrifugal supercharger, we use the known formula:

$$N_i - \left(\frac{n}{n_n}\right)^3 \rho\frac{mz_{pr}T_{pr}R_{pr}q_{pr}}{(m \quad 1)\eta_{pol}60}\left(\varepsilon_n^{\frac{m-1}{m}} - 1\right),$$

where m is a polytrophic coefficient.

There is a set of technological constraints on the following: the position of the working points on the characteristics of CeS to ensure the operation of the GPU without surging; maximum volumetric CeS efficiency; frequency of the CeS shaft rotation ($n_{\min}$ 6 n 6 $n_{\max}$); maximum GTU power of GPU; maximum CeS exit pressure, which is determined by the durability of pipelines at the exit of CeS; maximum temperature at exit of CeS, determined by the insulation of pipelines; minimum pressure value at the exit of each CeS; conditions related to the set of GPU operation stability level (distance from the surging zone); conditions of consistency of the scheme of CeS connection with the entry and exit loops.

2.2.2 Polytropic work of gas compression at compressor stations at the conditions of fixed accumulated gas volume in the system.

We consider one-line gas main pipeline with three operating compressor stations, which are connected by two sections of pipeline (Figure 2.1).

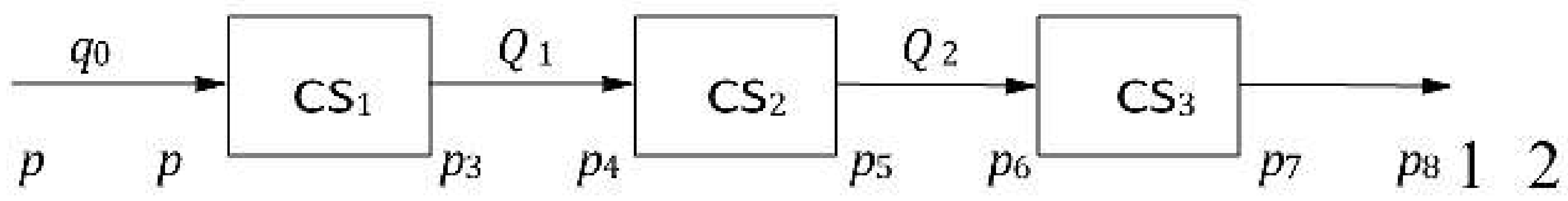


Figure 2.1 - Linear pipeline scheme.

We will conduct a study of the effect of the volumes of accumulated gas at the sections of the gas pipeline and the distribution of fuel gas consumption between the CS on the total energy costs with the following restrictions and given input data. It is believed that the indicators of:

— volumes of accumulated gas at the sections of the gas pipeline is constant, i.e.

$$Q = Q_1 + Q_2,\ \ Q_1 = \text{const},\ \ Q_2 = \text{const}; \qquad (1)$$

— entry p_2 and exit p_7 pressures are constant;

— volumetric gas consumption q_0 at the system entry is constant;

— exit volumetric consumption q_v is changing depending on the fuel gas volumetric consumption;

— indicator of CS work is determined by the pressure differentials ε_i, i = 1,2,3, summary polytropic gas compression work.

Task formulation. Find the following values of gas compression coefficients ε_i,i = 1,2,3 at compressor stations, which achieve a minimum total polytropical operation of gas compression $N_{\min}$ and follow the imposed restrictions.

Task solution. Under the defined gas flow mode the values of the pressure at the entries and exits of pipelines sections and compressor stations are connected by correlations:

(2)

$$p_3 = \varepsilon_1 p_2;$$

(3)

$$p_3^2 - p_4^2 = \varkappa_1;$$

$$p4 = \varepsilon 2p5; \qquad (4)$$

$$p_5^2 - p_6^2 = \varkappa_2;$$

$$p7 = \varepsilon 3p6.$$

(5)

(6)

Here the pipeline parameter κ_i, defined according to the formula

$$\varkappa_i = \frac{\lambda_i z_i R T_i}{2D_i}\left[\frac{4\rho_0\left(q_0 - \sum_{i-1}^{3}\Delta q_i\right)}{\pi D_i^2}\right]^2, \tag{7}$$

where i is the pipeline section number,

Note that for $i = 1$ — $\Delta q_1 = 0$. Polytropic gas compression work at compressor station is calculated by the formula

$$N = \frac{g\rho_0 q}{102}\frac{k}{k-1} z R T\left(\varepsilon^{(k-1)/k\eta} - 1\right). \tag{8}$$

Correlations (2)–(6) provide the opportunity to connect the values of entry and exit pressures between each other

$$p_7^2 = \varepsilon_1^2\varepsilon_2^2\varepsilon_3^2 p_2^2 - \varkappa_1\varepsilon_2^2\varepsilon_3^2 - \varkappa_2\varepsilon_3^2. \tag{9}$$

If we introduce the indication

$$l_{1i} = \frac{2\pi l_i D_i^2}{12 z_i R T},$$

then the volumes of gas accumulated at the sections of pipeline are calculated by the formulas

$$Q_1 = l_{11}\left(\varepsilon_1 p_2 + \frac{\varepsilon_1^2 p_2^2 - \varkappa_1}{\varepsilon_1 p_2 + \sqrt{\varepsilon_1^2 p_2^2 - \varkappa_1}}\right),$$

((10)

$$Q_2 = l_{12}\left(\varepsilon_2\sqrt{\varepsilon_1^2 p_2^2 - \varkappa_1} + \frac{\varepsilon_2^2\left(\varepsilon_1^2 p_2^2 - \varkappa_1\right) - \varkappa_2}{\varepsilon_2\sqrt{\varepsilon_1^2 p_2^2 - \varkappa_1} + \sqrt{\varepsilon_2^2\left(\varepsilon_1^2 p_2^2 - \varkappa_1\right) - \varkappa_2}}\right). \tag{11}$$

Herewith the summary polytropic gas compression work of compressor stations of given system is defined as follows:

$$N = \rho_1 R\left(\theta_1 q_1\frac{T_0\left(\varepsilon_1^{\nu_1} - 1\right)}{1 + f_0 p_2} + \theta_2 q_2\frac{T_2\left(\varepsilon_2^{\nu_2} - 1\right)}{1 + f_2\sqrt{\varepsilon_1^2 p_2^2 - \varkappa_1}} + \theta_3 q_3\frac{T_4\left(\varepsilon_3^{\nu_3} - 1\right)}{1 + f_4\sqrt{\varepsilon_2^2\left(\varepsilon_1^2 p_2^2 - \varkappa_1\right) - \varkappa_2}}\right). \tag{12}$$

In the last formula we have introduced the indication

$$\theta = \frac{k}{k-1}, \quad \nu = \frac{1}{\theta\eta}, \quad \rho_1 = \frac{g\rho_0}{102}.$$

The variable parameters in this task are the coefficients of gas compression at three compressor stations, and the constants are the volume of accumulated gas in the gas pipeline sections. It is obvious that in order to study the effect of compression coefficients on the energy costs of gas transportation, it is desirable to have an analytical solution of the formulated problem. Since the problem is nonlinear and it is not always possible to obtain an analytic solution, therefore, in our case, we confine ourselves to the first two units of the expansion in formulas (10) and (11). Then the following formulas are obtained for determining the volume of accumulated gas in the both gas pipeline sections

$$Q_1 = l_{11}\left(\varepsilon_1 p_2 + \frac{\varepsilon_1^2 p_2^2 - \varkappa_1}{\varepsilon_1 p_2 + \sqrt{\varepsilon_1^2 p_2^2 - \varkappa_1}}\right).$$

After certain transformations using asymptotic expansions in the first approximation we obtain that

$$Q_1 = \frac{3}{2} l_{11}\varepsilon_1 p_2\left(1 - \frac{5}{12}\frac{\varkappa_1}{\varepsilon_1^2 p_2^2}\right).$$

In the same way, to determine the volume of accumulated gas in the second section of the gas pipeline, the following formula is obtained

$$Q_2 = \frac{3}{2} l_{12}\varepsilon_2\varepsilon_1 p_2\left(1 - \frac{\varkappa_1}{2\varepsilon_1^2 p_2^2} - \frac{\varkappa_2}{4\varepsilon_1^2 p_2^2 \varepsilon_2^2}\right).$$

Then the total volume of accumulated gas at the sections of the gas pipeline is calculated by the formula

$$Q = \frac{3}{2}\varepsilon_1 p_2\left[l_{11}\left(1 - \frac{5}{12}\frac{\varkappa_1}{\varepsilon_1^2 p_2^2}\right) + l_{12}\varepsilon_2\left(1 - \frac{\varkappa_1}{2\varepsilon_1^2 p_2^2} - \frac{\varkappa_2}{4\varepsilon_1^2 p_2^2 \varepsilon_2^2}\right)\right],$$

or in the first approximation

$$Q = \frac{3}{2}\varepsilon_1 p_2\left(l_{11} + l_{12}\varepsilon_2\right),$$

where

$$\varepsilon_1 = \frac{2Q}{3p_2\left(l_{11} + \varepsilon_2 l_{12}\right)}. \qquad (13)$$

From the formula (9) we define the gas pressure differential for the third CS

$$p_7^2 = \varepsilon_3^2\left(\varepsilon_1^2\varepsilon_2^2 p_2^2 - \varkappa_1\varepsilon_2^2 - \varkappa_2\right),$$

where

$$\varepsilon_3 = \frac{p_7}{\sqrt{\varepsilon_1^2\varepsilon_2^2 p_2^2 - \varkappa_1\varepsilon_2^2 - \varkappa_1}}. \qquad (14)$$

Substituting the values ε_1 and ε_3 in the formula (12), we obtain the dependence of the summary polytropical gas compression work by compressor stations only on the pressure differential of the second CS, that is, one-parameter optimization task is obtained

$$N(\varepsilon_2) = \rho_1 R \left\{ \theta_1 q_1 \frac{T_0}{1 + f_0 p_2} \left(\left(\frac{b}{l_{11} + \varepsilon_2 l_{12}} \right)^{\nu_1} - 1 \right) + \theta_2 q_2 \frac{T_2 \left(\varepsilon_2^{\nu_2} - 1 \right)}{1 + f_2 \sqrt{p_2^2 \left(\frac{b}{l_{11} + \varepsilon_2 l_{12}} \right)^2 - \varkappa_1}} \right.$$

$$\left. + \theta_3 q_3 \frac{T_4}{1 + f_4 \sqrt{\varepsilon_2^2 \left(\left(\frac{b}{l_{11} + \varepsilon_2 l_{12}} \right)^2 p_2^2 - \varkappa_1 \right) - \varkappa_2}} \left[\frac{p_7^{\nu_3}}{\left(\varepsilon_2^2 p_2^2 \left(\frac{b}{l_{11} + \varepsilon_2 l_{12}} \right)^2 - \varkappa_1 \varepsilon_2^2 - \varkappa_1 \right)^{\nu_3/2}} - 1 \right] \right\}. \quad (15)$$

Here $b = 2Q/3p_2$.

The following algorithm for the calculation of above system optimal parameters is proposed.

We calculate the indicator of pressure differential ε_2 from the condition of function minimum $N(\varepsilon_2)$ (formula (15)).

By the formulas (13) and (14) we calculate the values of first and second CS pressure differentials.

From the conducted analysis it follows that the two imposed restrictions exclude consideration of the total two CS capacity minimizing — the first and third CS. Similarly, other CSs can be excluded.

Since the CS power, and thus the volumes of the used fuel gas, are nonlinear with respect to the pressure differentials, then a non-uniform solution should be expected. In the process of analysis of the solution (local extremes) we select the one that provides the minimum of fuel gas.

From the conducted study, it follows that if the number of CSs n is greater than the number of constraints m, then $n - m$ values of the pressure differentials are based on the minimization of the total capacity function from these coefficients.

The solution obtained in this way is approximate, since asymptotic formulas were used to determine the total volume of accumulated gas in two sections of the pipeline. However, to analyze the impact of gas pressure differentials on energy costs, this is enough. Obviously, in order to determine the values of gas pressure differentials more accurately for energy costs optimization, it is necessary to formulate an appropriate optimization task in a nonlinear formulation.

2.2.3 Calculation of extreme parameters of main gas pipeline operation with gas turbine GPU at CS.

In the same way as in the previous paragraph, the main gas pipeline, which consists of three operating CSs and sections of the gas pipeline between them (Figure 2.1), is considered.

If p_i is the pressure value at i-th point, and a_{ij} is calculated by the formula

$$a_{ij} - \frac{\lambda_{ij} z_{ij} R T_{ij} l_{ij}}{D_{ij}} \left(\frac{\rho_0}{S_{ij}}\right)^2, \tag{16}$$

then in the case of steady gas flow mode we have the correlation

$$p_i^2 - p_j^2 = a_{ij} q_0^2. \tag{17}$$

Considering the formula (17) and gas pressure differential dependency ε CS on the entry p_{vx} and exit p_{vyx} pressures ($\varepsilon = p_{vyx}/p_{vx}$), then following dependencies take place for the studied system:

$$
\begin{aligned}
&p_3 = \varepsilon_1 p_2,\\
&p_4^2 = \varepsilon_1^2 \left[p_1^2 - a_{12}(q_0 - q_{p1})^2\right] - a_{34}(q_0 - q_{p1} - q_{p2})^2,\\
&p_5 = \varepsilon_2 p_4,\\
&p_6^2 = \varepsilon_2^2 \left\{\varepsilon_1^2 \left[p_1^2 - a_{12}(q_0 - q_{p1})^2\right] - a_{34}(q_0 - q_{p1} - q_{p2})^2\right\} - a_{56}(q_0 - q_{p1} - q_{p2} - q_{p3})^2,\\
&p_7 = \varepsilon_3 p_6.
\end{aligned}
$$

Then we obtain the following connection between the values of starting and ending pressures and gas pressure differentials

$$
\begin{aligned}
p_8^2 = \varepsilon_3^2 &\left\{\varepsilon_2^2 \left\{\varepsilon_1^2 \left[p_1^2 - a_{12}(q_0 - q_{p1})^2\right] - a_{34}(q_0 - q_{p1} - q_{p2})^2\right\} - a_{56}(q_0 - q_{p1} - q_{p2} - q_{p3})^2\right\}\\
&- a_{78}(q_0 - q_{p1} - q_{p2} - q_{p3})^2.
\end{aligned} \tag{18}
$$

CS power, required for the gas compression of volume q_p, is calculated by the formula

$$N = \xi \frac{zR}{m} q_p T_1 \left[\left(\frac{p_2}{p_1} \right)^{m/\eta_{pol}} - 1 \right]$$

$$N = \xi \frac{zR}{m} q_p T_1 \left(\varepsilon^{m/\eta_{pol}} - 1 \right). \quad (19)$$

or

Here q_p is the gas consumption at CS (mcm/day), T_1 is the gas temperature at CS entry, m is the polytrope indicator, calculated by the formula

$$m = \left(1 - \lg \frac{T_2}{T_1} \Big/ \lg \frac{p_2}{p_1} \right)^{-1}, \quad (20)$$

p_1 is the CS entry pressure, p_2 is the CS exit pressure, T_2 is the gas temperature at CS exit, T_1 is the gas pressure at CS entry, η_{pol} is the polytropic efficiency, which is defined as follows:

$$\eta_{pol} = \frac{m}{m-1} \frac{k-1}{k},$$

where k is the adiabatic process indicator, ξ is the dimensional coefficient.

Gas quantity, used for gas compression at CS in the volume q_p, is calculated by the formula

$$q_n = 0.02064 \frac{\xi z R T_1 \left(\varepsilon^{m/\eta_{pol}} - 1 \right)}{1.16 m \eta_{gtu} Q_n} q_p \left[\frac{3}{4} + 0.025 \frac{p_a}{1.033\, K_3} \sqrt{\frac{T_1}{288}} \right], \quad (21)$$

where Q_n is the gas net calorific value (J/m^3), η_{gtu} is the efficiency of compressor gas turbine drive, p_a is the atmospheric pressure (MPa), K_3 is the engine load coefficient. The formulas (19)–(21) determine the amount of fuel gas required to maintain the output pressure at a given level. We introduce the indicator

$$B_i = 0.02064 \frac{\xi_i z R T_1}{1.16 m \eta_{gtu} Q_n} \left[\frac{3}{4} + 0.025 \frac{p_a}{1.033\, K_3} \sqrt{\frac{T_1}{288}} \right], \quad \alpha = \frac{m}{\eta_{pol}} = \frac{k(m-1)}{k-1}.$$

Then

$$q_{?i} = B_i q_i \left(\varepsilon_i^{\alpha_i} - 1 \right).$$

If three CSs are involved in the operation mode of GM, then fuel gas volumes at each of them will be calculated by the formulas

$$q_{p1} = \frac{B_1 q_0 \left(\varepsilon_1^{\alpha_1} - 1\right)}{1 + B_1 \left(\varepsilon_1^{\alpha_1} - 1\right)},$$
$$q_{p2} = \frac{B_2 \left(q_0 - q_{p1}\right) \left(\varepsilon_2^{\alpha_2} - 1\right)}{1 + B_2 \left(\varepsilon_2^{\alpha_2} - 1\right)},$$
$$q_{p3} = \frac{B_3 \left(q_0 - q_{p1} - q_{p2}\right) \left(\varepsilon_3^{\alpha_3} - 1\right)}{1 + B_3 \left(\varepsilon_3^{\alpha_3} - 1\right)},$$

and the total volumes of fuel gas q_{sn} will be the functions of three parameters ε_i, $i = 1,2,3$, i.e.

$$q_{pz} = \frac{B_1 q_0 \left(\varepsilon_1^{\alpha_1} - 1\right)}{1 + B_1 \left(\varepsilon_1^{\alpha_1} - 1\right)} + \frac{B_2 \left(q_0 - q_{p1}\right) \left(\varepsilon_2^{\alpha_2} - 1\right)}{1 + B_2 \left(\varepsilon_2^{\alpha_2} - 1\right)} + \frac{B_3 \left(q_0 - q_{p1} - q_{p2}\right) \left(\varepsilon_3^{\alpha_3} - 1\right)}{1 + B_3 \left(\varepsilon_3^{\alpha_3} - 1\right)}. \qquad (22)$$

Just as in previous paragraph, the pressure differentials ε_i, $i = 1,2,3$ are changing parameters. If the values of the input and output pressures are known, then formula (18) makes it possible to exclude one of the pressure differentials and substitute it into the equality (22). Therefore, the volumes of fuel gas can be considered as a function of two variables. The solution of the system of two equations with two unknowns by means of known methods allows us to determine the extreme values of the fuel gas volumes. For example, from the formula (22), we define ε_3 and substitute it into (22), then the system for finding the extremum will be

$$\left\{ \frac{\partial q_{pz}}{\partial \varepsilon_1} = 0, \quad \frac{\partial q_{pz}}{\partial \varepsilon_2} = 0 \right. .$$

Let us note that the addition of another condition, for example, for the volume of accumulated gas in the system, will allow us to exclude one of the parameters. In this case, one-parametric task is obtained to find the extrema of the fuel gas from the pressure differential.

2.3 Optimization of Fuel Consumption in Compressor Stations

Gas-transmission systems have been used for decades to transport large quantities of natural gas across long distances by means of compressor stations. Therefore, natural-gas compressor stations are located at regular intervals along the pipelines to compensate for pressure loss. These compressors consume a part of the transported gas, which results in an important fuel-consumption cost. Therefore, optimization of fuel consumption plays an important role in the operating costs of the stations. The difficulties of such optimization problems arise from several aspects. First, compressor stations are very sophisticated entities that may consist of a few dozen compressor units with different configurations, characteristics, and nonlinear behavior. Second, the set of constraints that defines feasible operating conditions in the compressors, along with the constraints in the pipes, constitutes a very complex system of nonlinear constraints.

Flores-Villarreal and Ríos-Mercado (2003) extended a study by means of an extensive computational evaluation of the generalized reduced-gradient method to obtain a fuel-cost-minimization problem. Borraz-Sánchez (2010) discusses the method of minimizing fuel cost in gas-transmission networks by dynamic programming and adaptive discretization. Goldberg and Kuo (1985) provide genetic algorithms that were used for the first time for the optimization of the pipelines. Odom and Muster (1991), members of a solarturbine manufacturing corporation, presented two diverse versions for modeling gas turbines that activate centrifugal compressors. Andrus (1994) wielded spreadsheets for the first time to simulate the steady-state gas-transmitting pipes and network, stipulating that the compressibility factor is constant for all the systems. The aforementioned methodology was executed once again by Cameron (1999) to examine both steady and transient flows in Excel software. Chapman and Abbaspour (2003) developed fuel-optimization methodologies in compressor stations in nonisothermal and transient conditions with compressor-shaft velocities as a decisionmaking variable.

Edgar and Himmelblau (1988) and Osiadacz (1994) used both nonlinear programming (NLP) and a branch-and-bound scheme for optimal design of gas-transmission systems. Wolf and Smeers (2000) used mixed-integer NLP (MINLP) for fuel-cost minimization. Kabirian and Hemmati (2007) applied an integrated nonlinear-optimization model and a heuristic random-search method for design and development of natural-gas-transmission pipeline networks. Chebouba et al. (2009) applied ant-colony optimization (ACO) for the first time for the fuel-consumption-minimization problem in natural-gas-transportation systems. For gas-pipelineoperation optimization, the ACO algorithm is a new evolutionary optimization method. Zhang et al. (2009) applied a fuzzy optimization for design of a gas pipeline.

Goslinga et al. (1994) showed that gradient-based optimization methods have been used to analyze gas-pipeline networks in the past. As the name implies, they rely on the derivative of the function being optimized with respect to all control variables that define the system. There are several gradient-based optimization methods, depending on the nature of the objective function and associated constraints (i.e., constrained and unconstrained linear programming, quadratic programming, and NLP). An extensive review of these methods and available software tools can be found in More and Wright (1993). Tsal et al. (1986) showed that dropped particularly close to the operation boundaries, often trapped in local minima and very dependent on the starting point. These methods have also been extended to transient pipeline optimization, but only on relatively smaller systems. Hawryluk et al. (2010) studied optimizations that were based on dynamic programming (e.g., based on Bellman's optimality principle). Habibvand and Behbahani (2012) showed the optimization of the fuel consumption of the compressors through manipulation of the affecting parameters of the compressors and the operating-condition parameters of the turbines and the air coolers within a gas-compression-station unit in operation phase by use of a genetic algorithm.

Jenicek and Kralik (1995) described an optimization system for local control of compressor-station operation. The compressor-station topography is arbitrary, and the configuration of compressors was assumed to be fixed, while each compressor unit was described by its individual characteristics. The optimization was performed as a steady-state optimization, and the criterion used was represented by a total of the consumed energy or a total of the price for the consumed energy. Wright et al. (1998) discussed simulated annealing as a technique for finding the optimum configuration and power settings for multiple compressors when the number of compressors is large. The simulatedannealing technique derived from statistical mechanical considerations.

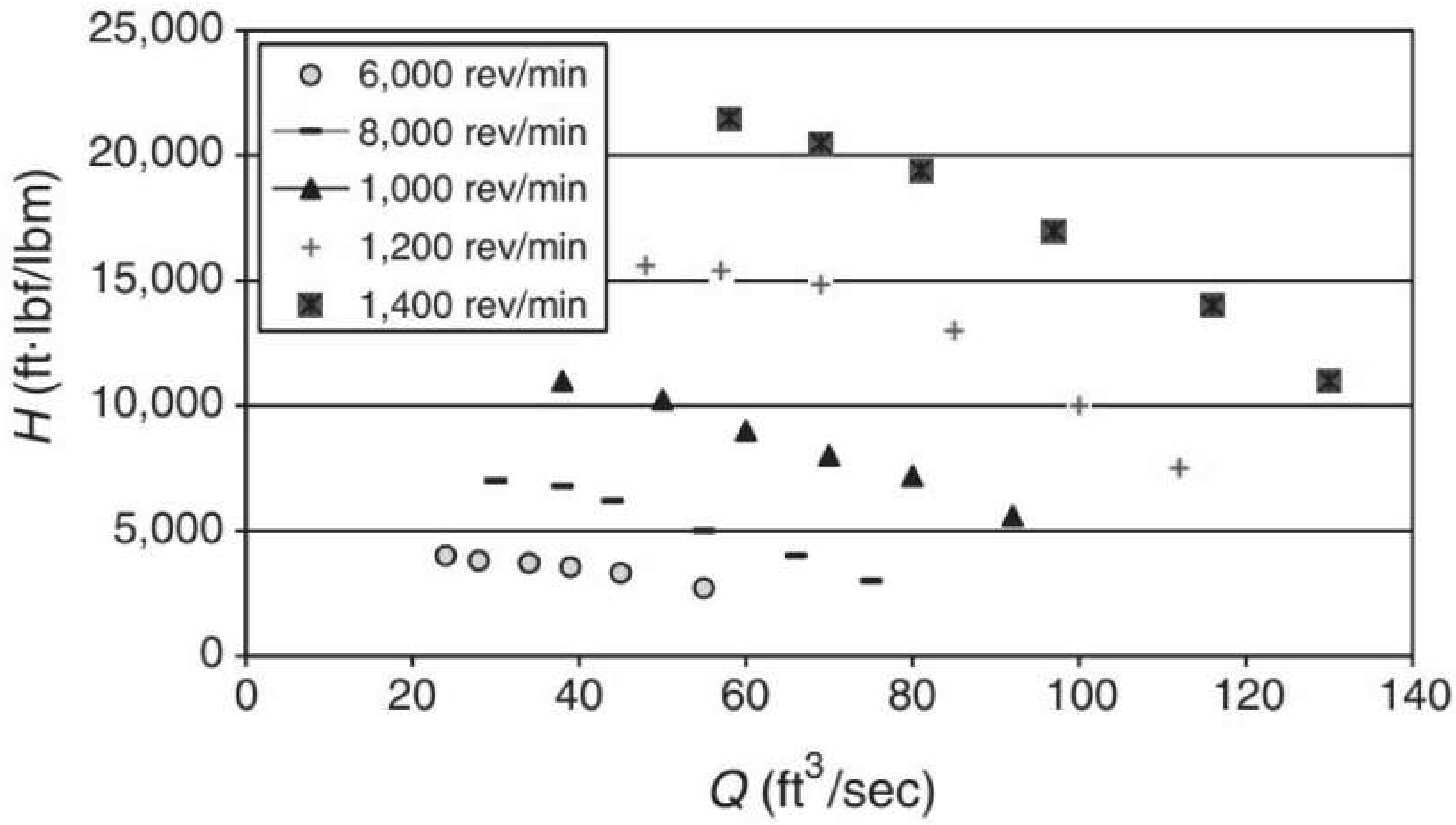


Figure 2.2 - Single-centrifugal-compressor map.

was used to find the "best-use" mode of their operation. Tests of simulated annealing against the more ubiquitous MINLP and heuristic techniques were performed.

Jin and Wojtanowicz (2010) discussed the optimization of a large gas-pipeline network—a case study in China that aimed to optimize the network to minimize its energy consumption and cost. The large size and complex geometry of the network required breaking the study down into simple components, optimizing operation of the components locally, recombining the optimized components into the network, and optimizing the network globally. This four-step approach used four different optimization methods—penalty function, pattern search, enumeration, and nonsequential dynamic programming—to solve the problem. The results showed that cost savings because of global optimization were reduced with increased throughput.

The purpose of this paper is to minimize the fuel consumption in the compressor stations with a nonlinear mathematical relationship as the objective function. Genetic algorithms are used as the optimization methodology. The genetic algorithm is a relatively new optimization method and serves well as an optimization tool. This is because the nature of the variables involved is made of continuous and discrete variables that can change several parameters simultaneously, and its use in the pertinent nonlinear problem does not make it entrapped in the local-optimum spots. Two cases of centrifugal compressor stations with different performances are studied.

2.3.1 Main methods result

Centrifugal Compressor. Habibvand and Behbahani (2012) showed that the governing quantities of a centrifugal compressor unit are inlet volumetric flow rate Qi , speed S (rev/sec), adiabatic head H (ft-lbf/lbm), and adiabatic efficiency η. It has been recognized that the relationship among these quantities can be described by the following two equations:

$$\frac{H}{S^2} = A_H + B_H\left(\frac{Q}{S}\right) + C_H\left(\frac{Q}{S}\right)^2 + D_H\left(\frac{Q}{S}\right)^3$$

And

$$\eta = A_E + B_E\left(\frac{Q}{S}\right) + C_E\left(\frac{Q}{S}\right)^2 + D_E\left(\frac{Q}{S}\right)^3$$

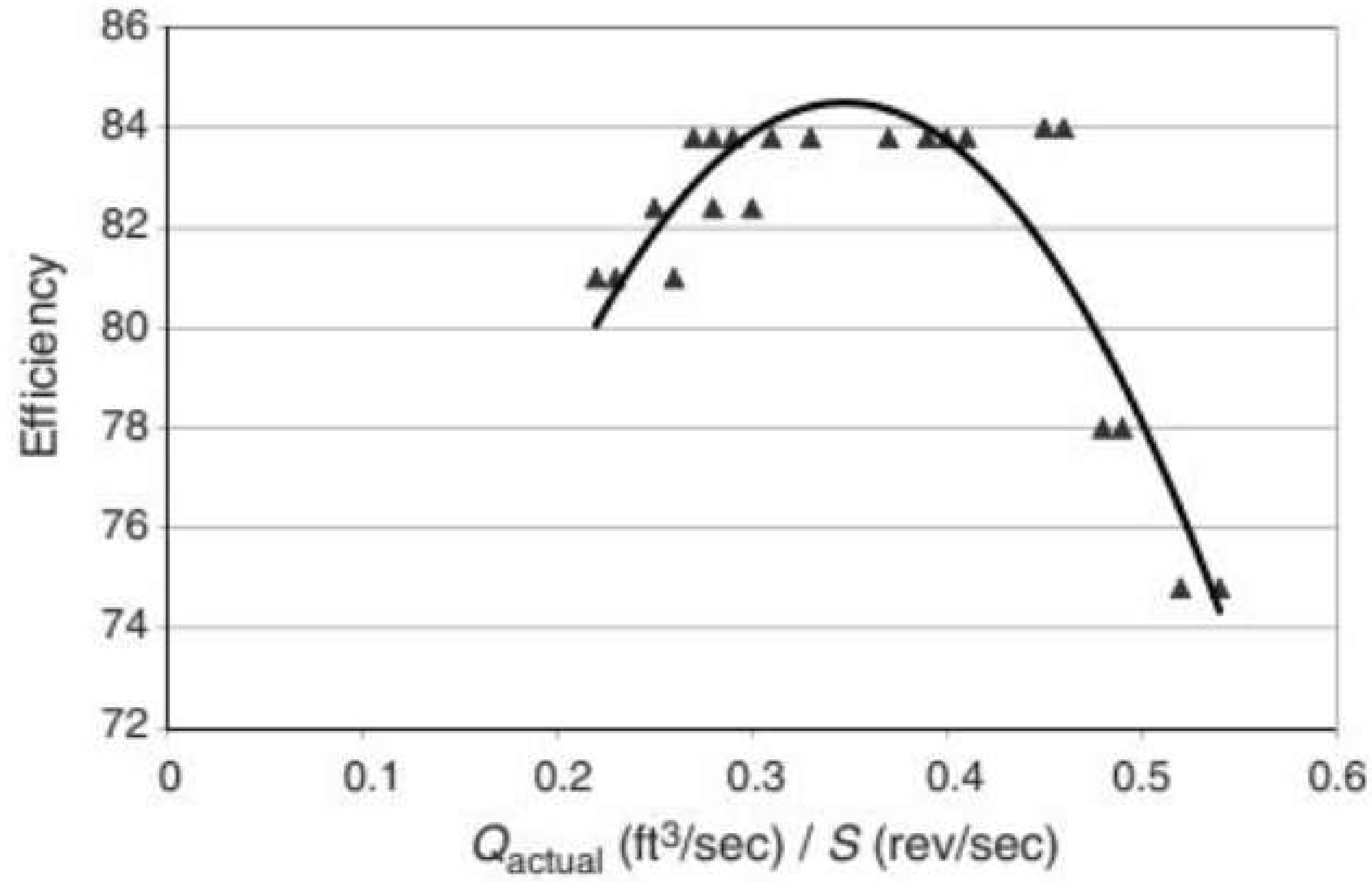


Fig. 2.3 - Compressor efficiency as a function of Q_{actual} per speed

where A_H, B_H, C_H, D_H, A_E, B_E, C_E, and D_E are constants that depend on the compressor unit and are typically estimated by applying the least-squares method to a set of collected data of the quantities S (rev/sec), H (ft-lbf /lbm), and η. Figures 2.2 and 2.3 show the set of data collected from a typical centrifugal unit. Fig. 1 represents H_i (ft-lbf/lbm) vs. Q_i (ft^3/sec) at different speed values for S_i (rev/sec) (between $S_{i,\min}$ and $S_{i,\max}$ in rev/sec) and Q_i/S_i generated by Eq. 1. A plot of Eq. 2 is illustrated in Fig. 2.2

Woldeyohannes and Majid (2011) showed that the relationship between H (ft-lbf/lbm), suction pressure P_s (psia), and discharge pressure P_d (psia) is

$$H = \left(\frac{k}{k-1}\right) ZRT\left[\left(\frac{P_d}{P_s}\right)^{\left(\frac{k-1}{k}\right)} - 1\right].$$

The power of the compressor can be computed by the following relation:

$$PWR = \frac{32.174 \times W \times H}{\eta \times \eta_{\text{mech}}} + Al$$

The power of the compressor can be computed by the following relation:

Centrifugal compressors have been bounded within choke/stonewall limits at the bottom of the map and surge and stall scope at the top of the map and at the minimum and maximum shaft-rotation speed.

Compressibility-Calculation Procedure. Mallinson et al. (1986) show that the compressibility of natural gas through pipelines and compressor stations can be estimated from the following equation:

$$Z = Z_0 + \left(Z_1 B + Z_2 A^2\right) P + \left(3Z_3 BA^2 + Z_4 A^4\right) P^2 .$$

Z_0, a, and b are constants derived from a virial series expansion (Eq. 5) for Z on the basis of the Redlich-K wong equation of state, and are dependent upon the gas temperature and the pseudocritical pressure and temperature.

$$Z = Z_0 + \left(Z_1 B + Z_2 A^2\right) P + \left(3Z_3 BA^2 + Z_4 A^4\right) P^2 .$$

A and B are constants that depend on the gas temperature and the pseudocritical pressure and temperature. The constants a and b are a = Z_1B + Z_2A^2 and b = $3Z_3BA^2$ + Z_4A^4. The values of A and B are $A2$ =VS $T1$ $C2$ 5. /P TC 2 5. and B =VS $T2$ C /P TC .

Gas Turbine. The gas turbine generates the required power for the compressor. Habibvand and Behbahani (2012) showed that the required PWR (ft-lbf/sec) of the compressor is used to compute the generating power of the gas turbine and the fuel-consumption rate. The quantity of fuel consumption (FC) of the gas turbine in scf/sec can be computed as

$$APWR = \left[1 - ARF\left(T_A - T_R\right)\right] \times RPWR \tag{8}$$

$$FAP = \frac{PWR}{APWR} \tag{9}$$

And

$$FC = PWR \Big/ \left(\eta_D \times HV\right)$$

Optimization. Objective Function. Dual-shaft gas turbines are used as drivers. When centrifugal compressors are driven by gas turbines, they may initiate diverse operational circumstances through velocity control. This is one of the most natural manners of controlling the transmission system because the centrifugal compressor and dual-shaft gas turbines can function on a wide range of velocities. Having determined the flow rate by means of a tool such as an orifice, by a Venturi nozzle, or by infrasonic equipment, the controller can alter the fuel flow in case any change in the output or input pressure transpires. Thus, the gas turbine will produce energy commensurate with the consumed fuel and will instigate the decrease of the compressor rotation. Hence, it is quite worthy to opt for the compressor rotation velocity as a decision-making variable. The consumption rate of the gas-turbine fuel relation can be applied to determine the target function, thus counterpoising the power initiated in the gas turbine with the required energy to materialize the set points and constraints of the compressor and transmission system. The relationship can be simplified on the basis of the decision-making variable.

$$\min \sum_{i=1}^{n} FC(S_i) = \min \sum_{i=1}^{n} \frac{PWR_i}{\eta_{Di} \times HV}$$

Constraints. The constraints are usually provided. They are minimum speed $S_{i,\min}$ (rev/sec), maximum speed $S_{i,\max}$ (rev/sec), surgelimit surge, and stonewall-limit stonewall. These provide the limits to the speed S_i and the ratio of Q/S:

$$S_{i,\min} \leq S_i \leq S_{i,\max}$$

and

$$surge \leq \frac{Q_i}{S_i} \leq stonewall$$

From Eq. 11 and 12, it follows that the inlet volumetric flow rate Q_i (ft^3/sec) must satisfy conditions, where $Q_i^L = S_{i,\min} \times surge$ and $Q_i^U = S_{i,\max} \times stonewall$. For each Q_i within this range, the adiabatic head H_i is bounded below by either $S_{i,\min}$ or stonewall and bounded above by either $S_{i,\max}$ or surge. Let $H_i^L(Q_i)$ and $H_i^U(Q_i)$ be the lower and upper bound functions, respectively. Then,

$$H_i^L(Q_i) \le H_i \le H_i^U(Q_i),\ Q_i^L \le Q_i \le Q_i^U$$

Conclusion to chapter 2

1. Nowadays, UGTS has a lot of issues that must immediately solved, to ensure the possibility of proper working processes. The total cost of every compressor station modernization or rebuilding can be decreased with using of new methods of construction, which can be borrowed from the world-wide companies.
2. Using this methods, major of these problems would be resolved in the recent decade, which allow us to dictate market conditions, and use it for bigger profit, which can be spent of further modernization.

CHAPTER 3. METHODS OF PRESSURE INCREASING ON GAS PIPELINES FOR TOTAL EFFICIENCY OUTPUT INCREASING

Introduction

Today, despite growing green energy tendency, gas as energy supply is still among the leaders, and it's extraction and consuming only increases without stop, and gas itself will be valid as energy resource for at least the next few decades.

We can see that for the last few years, the United States of America increase its total gas production and trying to increase its supply market over Europe. In order to compete in the market of energy resources, we need to decrease the total cost of gas transportation and increase the safety of gas supplying trough our pipelines.

The most perspective way in reaching this aim is to perform modernization of UGTS in order to increase the pressure after single compressor stations to achieve a better base for further optimization, automatization, and ensure the gas supplying security.

Below I will consider methods to achieve the work of compressor stations at elevated pressures of the main gas pipeline.

3.1 Methods of pressure increasing in the main pipeline.

3.1.1 Compressor station pressure output increasing by adding additional engines of the same capacity.

Distinguish between installed, working and standby capacity of the compressor station.

Operating productivity CS, Q_w m3 / min, is the sum of the nominal capacities of all the station compressors, with the exception of the reserve ones. Standby capacity, Q_s m3 / min, is the sum of the nominal capacities of standby compressors of a given compressor station.

Installed capacity CS Q_w, m3 / min, is the sum of the nominal capacities of all compressor machines installed at the station, including standby ones:

$$Q_i = \sum_{1}^{m} Q_{K_i} = Q_w + Q_s$$

where m- is the number of compressors at the compressor station Q_k, is the nominal capacity of one (i-th) compressor, m3 / min. The methods for selecting the indicated capacities, as well as the number, types and sizes of compressors installed at the station, depend on the nature of consumers and the schedule of compressed air consumption from the compressor station.

At compressor stations supplying a large number of small consumers with air with a pressure not higher than 1-1.2 MPa, units of the same type are usually installed. Let us be given a daily schedule of compressed air consumption in% of Qm.d. We consider that Qs = Qm.d. At the compressor station, the same type of units (both working and standby) were accepted for installation.

Let us construct a dependency diagram Qs and the number of working machines installed on the CS $-n_w$ (see Figure 3.1):

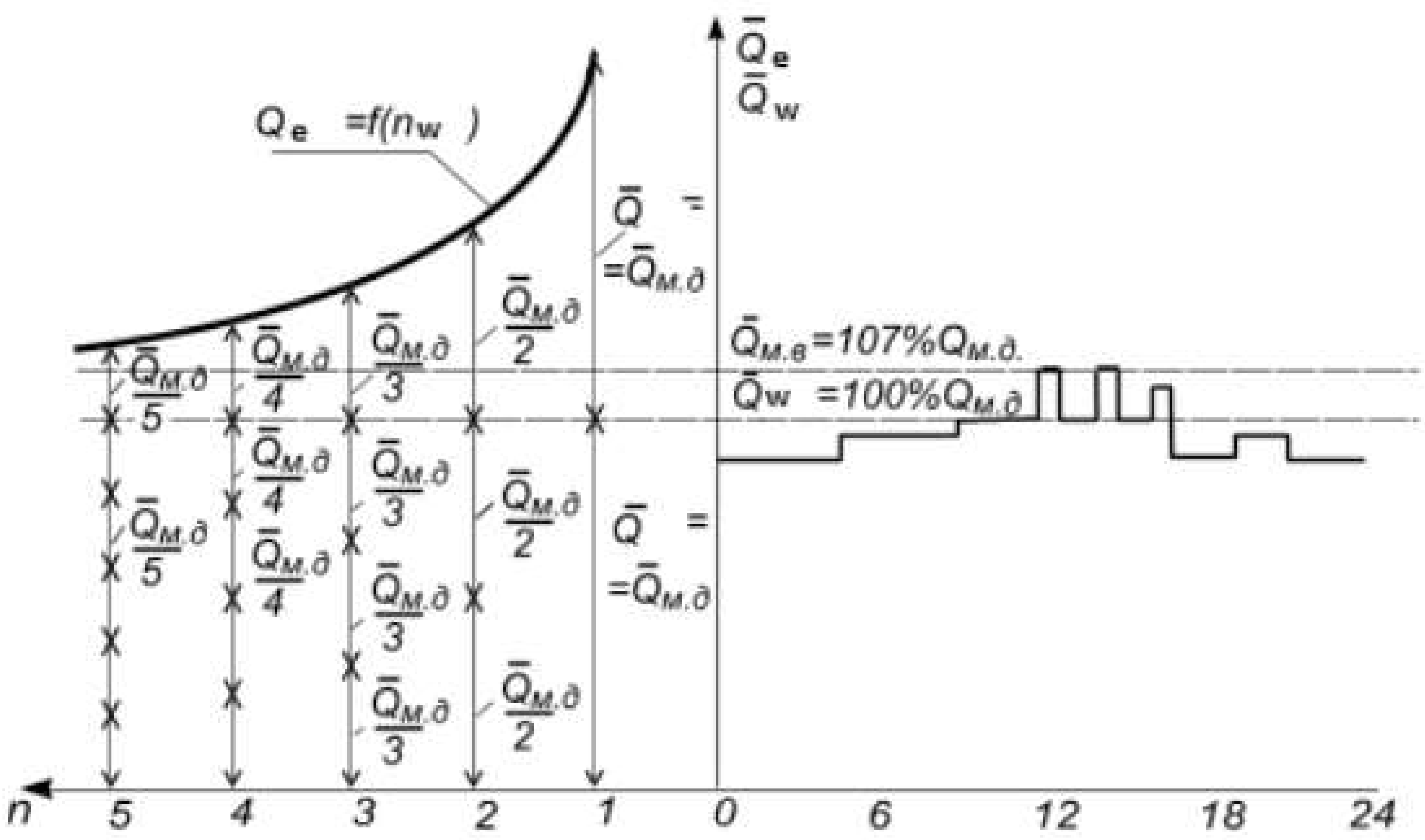


Figure 3.1. Dependence of the installed capacity of the compressor station on the number of working machines accepted for installation

The annual cost of producing compressed air is dominated by the cost of electricity for driving compressors and the salary of maintenance personnel. Wage costs increase with the growth of n_w.

At the same time, the cost of electricity is reduced, since with an increase in the number of units it will be more economical to regulate performance by turning off and on individual machines.

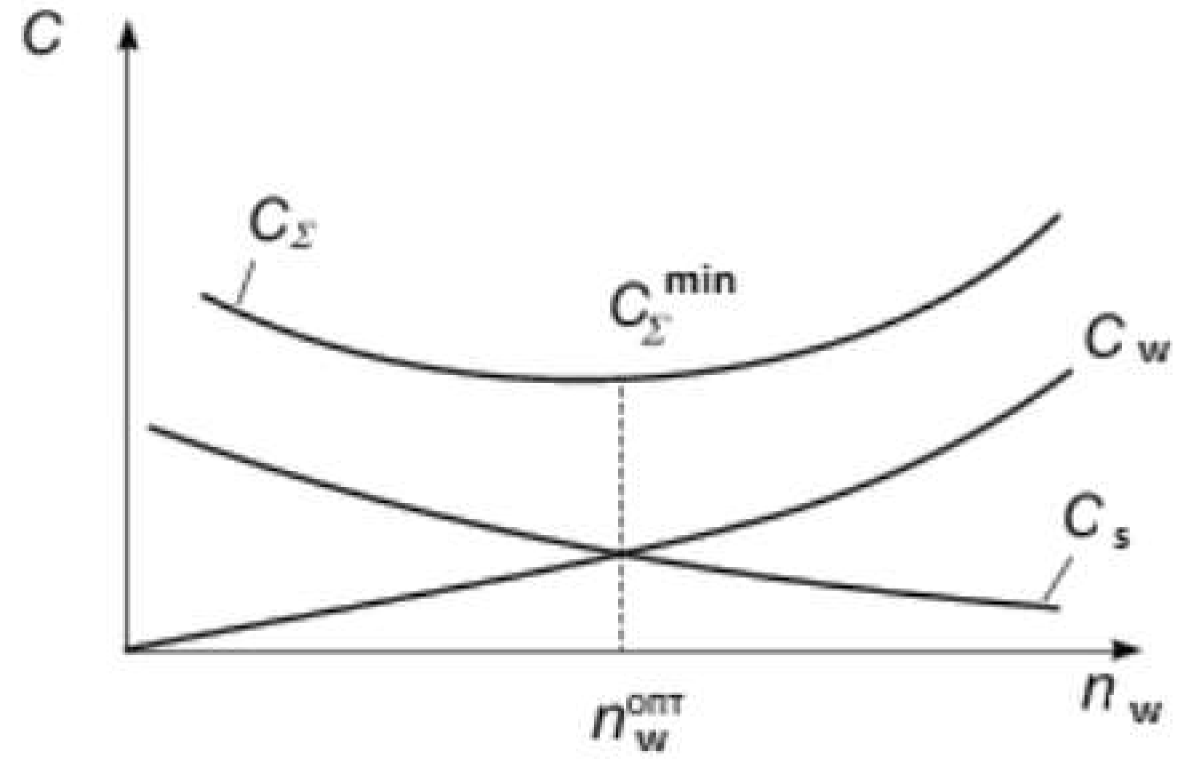


Figure 3.2. Dependences of the cost of working and standby compressors installed on the compressor station from the number of working machines accepted for installation

It can be seen from the diagram that with an increase in the number of working machines at the compressor station, the productivity of individual units Qk decreases. The values Q_w and Q_s are also decreasing. At the same time, however, the capital specific costs of Z_{cap} increase, since the cost of two cars is more than the cost of one car with the same capacity. Thus, with an increase in *n*, the cost of working compressors C_w, and the cost of the reserve C_s (see. Fig. 4.3). This indicates that there is an option when the total capital costs C_Σ have a minimum.
The simple adding of additional engines, will be efficient only in long-term perspective, and will highly increase the cost the maintenance.

3.1.2 Optimization of CS engines to increase the total efficiency, and pressure in the pipeline.

A typical optimization task for gas turbines would involve the increase of thermal efficiency (i.e. the fuel consumption per unit of power output), and the increase in power density (the amount of power per units of mass flow). Both parameters can be improved by increasing the firing temperature of the engine (Figure 3.3).

However, increasing the firing temperate on an existing engine will have a negative impact on engine life, that is the time between necessary overhauls, and related to that, the reliability of the engine. This effect can be countered by increasing the amount of cooling air for the turbine blades, but that will also reduce the engine efficiency. By being able to very accurately predict local blade temperature, as well as to verify these temperatures experimentally one can use cooling air as miserly as possible. Tools like conjugate heat transfer, irradiated crystal sensor measurement methods, as well as improved materials and coatings can help in this balancing act.

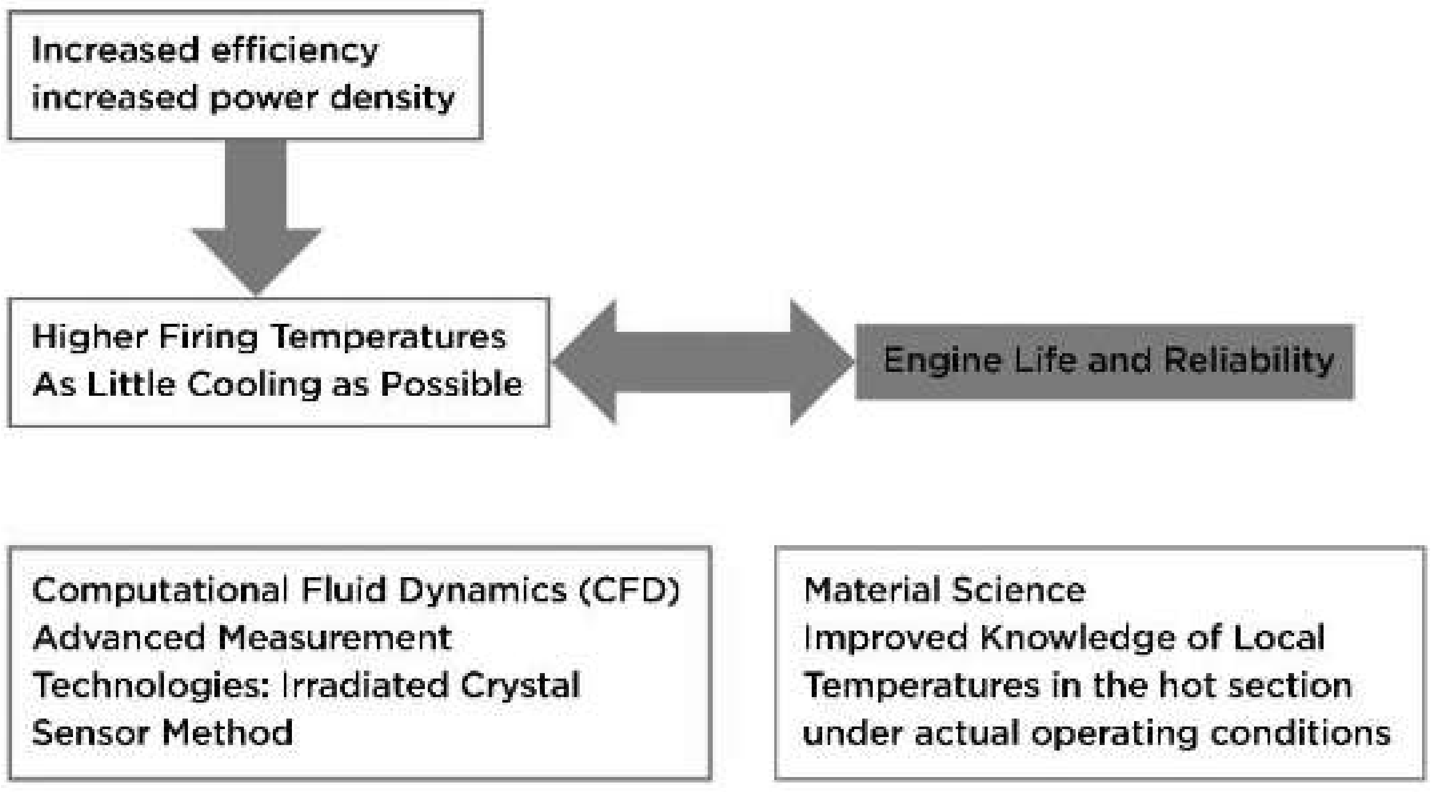


Figure 3.3 Optimizing for Contradicting Requirements.

Similarly, for the improvement of the centrifugal gas compressors (Figure 3.4), the change in operating conditions requires not just a higher efficiency (for the compressor, this is the ratio of work that an ideal, isentropic compressor would consume compared to the actual work consumed), but in particular, that a good efficiency is available over a wide flow range. Again, we have two contradictory requirements. CFD tools have allowed us to better understand flow structures in impellers and diffusers, and, together with experimental methods, led to improvements in both operating range and operating efficiency.

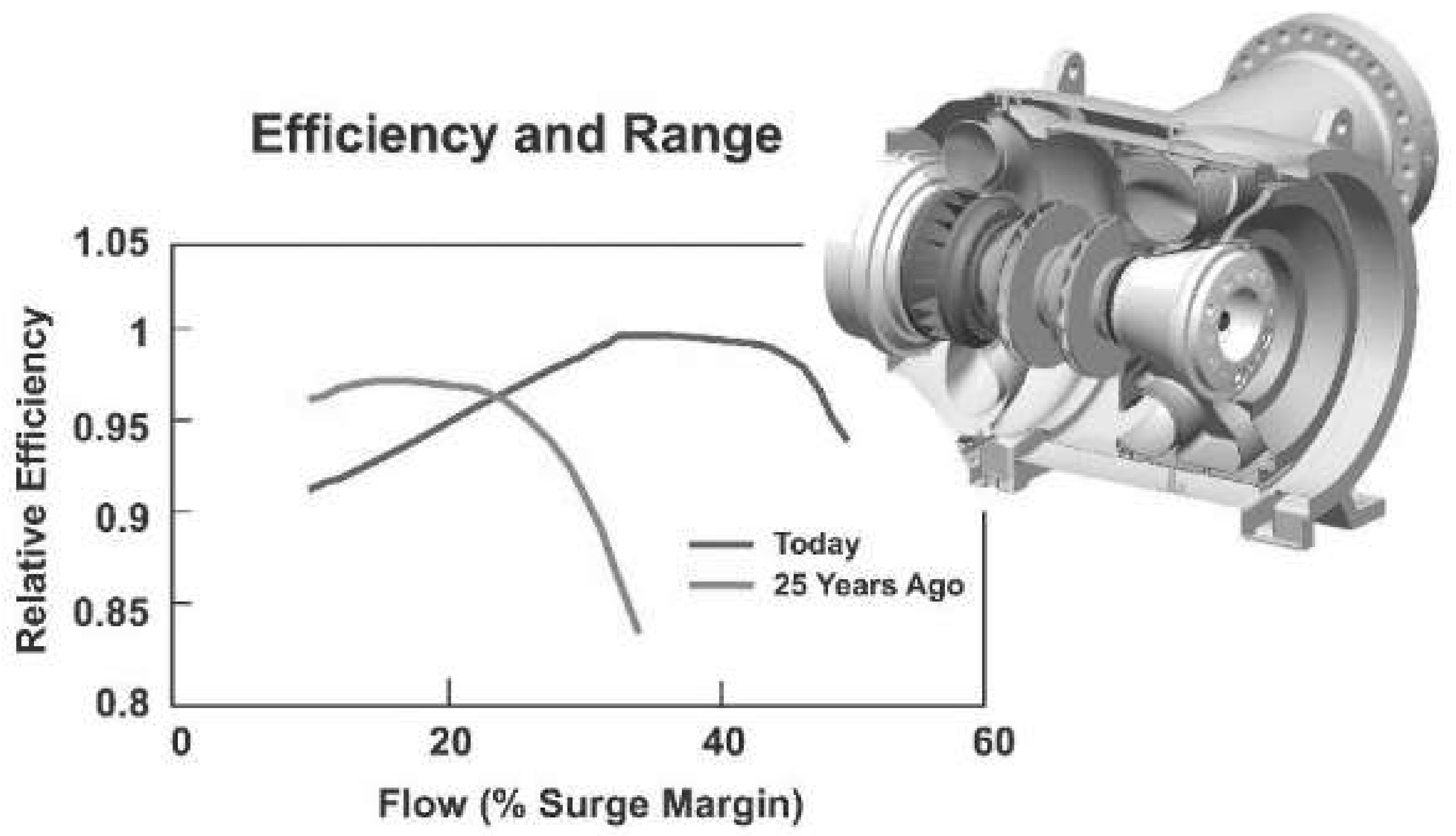


Figure 3.4 Gas Compressor Improvements

AS the conclusion of the CS engine optimization, we can achieve better efficiency, and due to better cooling processes, we can increase the output pressure. Still it is the short-term strategy, which will give a pressure boost, but will not give an opportunity to follow the market demands. (указать что оптимизация именно по давлению)

3.2 Modernization of UGTS with combined cycle power plans

The function of a combined cycle plant is to provide more capacity from a single stage of gas unit. The major components of a combined cycle plant are a gas turbine, a heat recovery steam generator, a steam turbine, and balance of plant systems.

In order to understand how a combined cycle plant differs from other types of power plants, the functioning principles of other plants should be understood. A simple cycle-type plant consists mainly of just gas turbines, and has no heat recovery capability. The hot exhaust from the turbines is simply released to the atmosphere, which makes the plant much less efficient than other types, but is cheaper to build and operate. In this type of plant, only a gas turbine is used to operate a generator for the capacity. Another type of plant is the co-generation plant. It uses the hot exhaust of a gas turbine to produce steam in a heat recovery steam generator. The steam is then sent offsite, where it is used at another facility or to provide steam for processes other than electrical generation. In this type of plant, similar to the simple cycle plant, only a gas turbine is used to operate a generator for the production of electricity. A plant is only considered a combined cycle plant when the hot exhaust from the gas turbine is used to produce steam in a heat recovery steam generator, which is then used to operate a steam turbine. In this type of plant, the gas turbine and the steam turbine are used to operate generators for the capacity.

Major Components is the overall purpose of a combined cycle plant is to provide capacity for compressors. The gas turbine produces capacity and hot exhaust; the heat recovery steam generator transfers the heat from this hot exhaust to water in order to make superheated steam. The superheated steam produced in the heat recovery steam generator is supplied to the steam turbine through steam piping. The steam is then used by the steam turbine to make capacity, and the exhaust steam is then cooled and condensed back into water in the condenser.

After provided research I have noticed that the highest efficiency type of combined cycle plants are S-STIG plants.

The thermal diagram of the gas and steam-water cycles of a gas turbine unit operating according to the A-STIG scheme (Fig. 3.4), in T-S coordinates (temperature-entropy) is shown in Fig. 3.4. In this diagram, lines J - C and 5-C 'correspond to the expansion process of the components of the gas-vapor mixture in the turbine flow path and the GTE outlet device, the JC line characterizing the gas expansion process, and the 5-C' line the overheated steam of a homogeneous mixture. The heat of the gas component of the gas-vapor mixture is given in the boiler through the C-YX line, and the steam component through the C'-6 'line. Due to the use of the heat of the gas-steam mixture leaving the gas turbine in the steam recovery boiler, water is sequentially heated in the economizer (line 2-3), water evaporates on the evaporator surfaces (line 3-4) and partial steam overheats in the steam superheater (line 4 - 4'). Further steam overheating is carried out in the combustion chamber of a gas turbine engine (line 4'-5). The process of cooling a gas-vapor mixture in a contact condenser to a temperature of cooling water occurs along the YX-H lines for gas and 6'-6 for steam. The process of steam condensation occurs along line 6-1, and a slight heating of the condensate in the condensate collector, condensate purification unit, feed water supply tank and pump before flowing into the economizer of the recovery boiler - through line 1-2. Compression of air in the compressor of a gas turbine engine occurs along the HK line. Gas heating in the combustion chamber of a gas turbine engine occurs along the line K-J

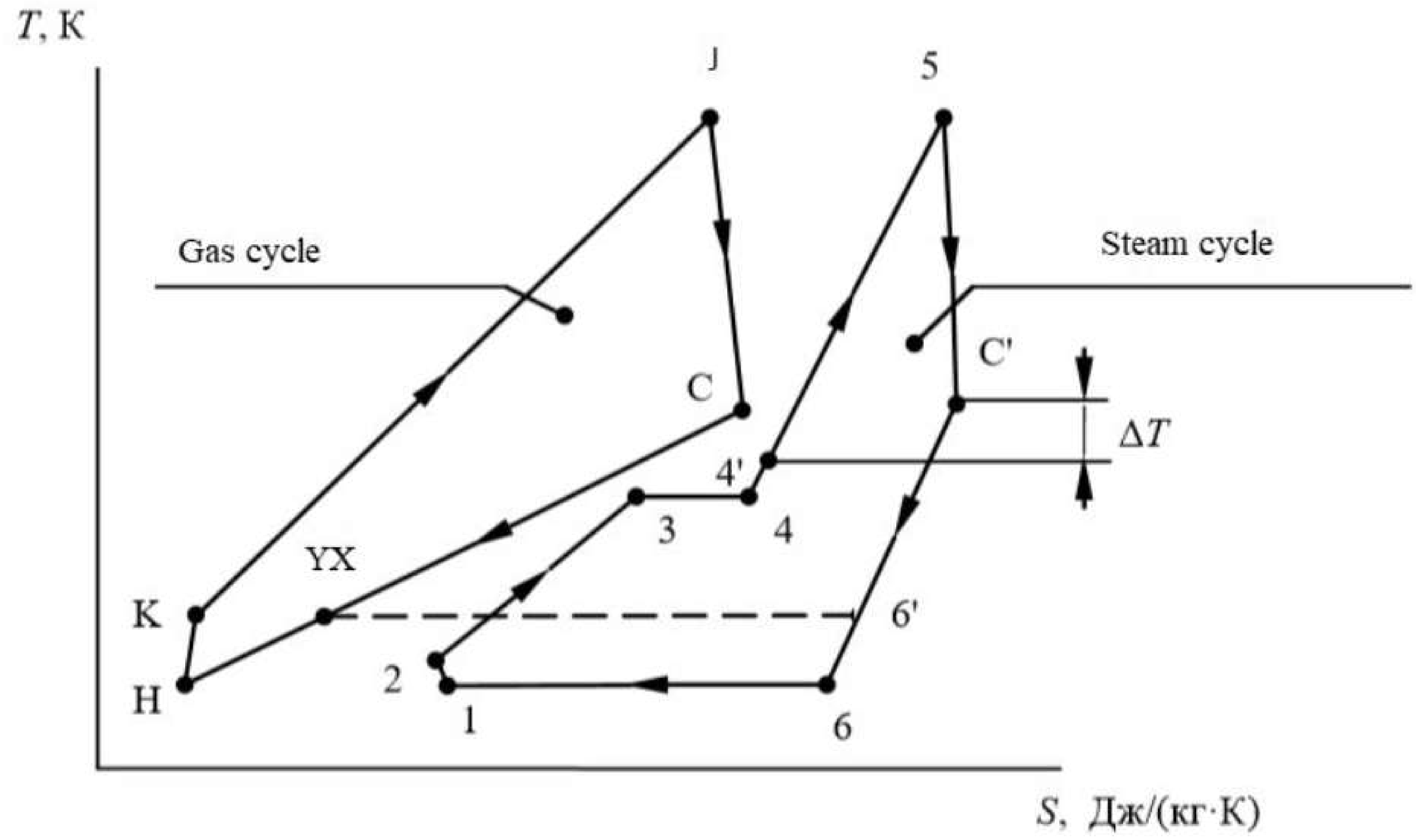


Figure 3.1 - Thermal diagram of the gas turbine A-STIG

An analysis of the main parameters and performance of gas turbine units with an improved scheme for injecting water vapor into the combustion chamber (GTU A-STIG) and steam-water cooling of a gas turbine proves the high energy and environmental efficiency of these combined plants. In the most modern schemes and constructions of GTU A-STIG, the efficiency is 45 ÷ 47%, and the heat utilization factor is 80 ÷ 90%. Research and development shows that in the future in promising A-STIG gas turbines with steam-water cooling and water injection into the compressor, a coefficient of performance equal to 55 ÷ 57% can be obtained. Industrial operation of these plants in different countries proves their advantage in comparison with various simple-cycle gas turbine plants, combined electric and cogeneration gas-steam turbine plants, STIG gas-turbine plants and

expands the possibilities of combined power plants in terms of their technical, economic and environmental efficiency application.
Using this type of machines in UGTS modernization, will give us a great capacity boost, and will allow us to use current machines for a long time, thus the combined cycle plant can be also used in various sphere of life, not only in gas transport.

3.3 Analysis of shut-off fittings at gas compressor stages and dynamic modeling in non-stationary modes in main gas pipeline

3.3.1 Analysis base

Multi-threat gas pipelines (MTP) are the most complex objects of the gas transportation system when modeling the modes of gas transport by MTP must take into account all the features of transporting gas under high pressure over long distances. At certain intervals, MTP installed compressor stations that maintain pressure in the gas pipeline and the taps necessary for localization in case of accident of a pipeline section or disconnection of pipeline sections during repair and maintenance work. In this technological MTP equipment, namely: shut-off and control devices that narrow the pipe cross-section - can affect the modes of gas transport, which, due to various causes, are non-stationary and non-isothermal. This effect is the hydraulic pressure loss on the so-called local hydraulic resistances: valves, taps, valves, diaphragms, all kinds of rounding, constrictions, extensions, etc., i.e., wherever the flow undergoes deformation which in individual cases can significantly affect the change in gas flow parameters (pressure, gas flow and temperature) when calculating real processes of natural gas flow. Therefore, the urgent task is to take these losses into account when modeling of non-stationary non-isothermal modes of gas transport (NSIMGT) by linear sections (LS) MTP through valves. In this regard, the goal of this study is to develop a solution method systems of differential equations of a mathematical model (MM) NSIMGT in LS MTP, including MM shutoff valves, in order to take into account the dynamics of its operation.

The following tasks are solved: selection of MM shutoff valves and development of a method calculation of NSIMGT in the LS MTP, the mathematical model of which includes MM sections of pipelines (PS) and shutoff valves.

3.3.2 Structure and mathematical model of LS MTP

The mathematical model of the structure of the LS MTP is the directed graph G (V, M), where V is the set of nodes of the graph, M is the set of arcs of the graph. The nodes of the graph represent the junction of technological elements with each other. The set of arcs M = M1∪M2, where M1 is the set of arcs of the graph corresponding to PS, M2 is the set of arcs of the graph corresponding to taps. The set of nodes V = V1 ∪ V2 ∪ V3 ∪ V4 ∪ V5, where V1, V2, V3, V4, V5 is the set of MTP inputs, the set of intermediate nodes, the set MTP outputs, the set of inputs and outputs to the f-th crane, f = 1, θ, f ∈ M2, respectively,

$$|V| = v,\ \ |V_1| = v_1,\ |V_2| = v_2,\ |V_3| = v_3,\ |V_4| = |V_5| = \theta.$$

As a mathematical model of NSIMGT along the pipeline section, which is a cylindrical pipe of constant diameter, we take a system of partial differential equations taking into account the throttle effect (the PS index is omitted for convenience)

$$\frac{\partial W}{\partial t}+(1-\alpha ST\frac{W^2}{P^2})\frac{\partial P}{\partial x}+2\alpha ST\frac{W}{P}\frac{\partial W}{\partial x}+\beta ST\frac{W|W|}{P}+\frac{g}{\alpha S}\frac{P}{T}\frac{dh}{dx}=0, \quad (1)$$

$$\frac{\partial P}{\partial t}+\alpha ST\frac{\partial W}{\partial x}=0, \quad (2)$$

$$\frac{\partial T}{\partial t}+\alpha S\gamma T\frac{W}{P}\frac{\partial T}{\partial x}+\alpha S(\gamma-1)\frac{T^2}{P}\frac{\partial W}{\partial x}+\frac{4K}{D}(\gamma-1)\frac{T}{P}\left(T-T_{rp}\right)+g(\gamma-1)\frac{WT}{P}\frac{dh}{dx}=0, \quad (3)$$

Where $\alpha=\frac{zgR}{S}$, $\beta=\frac{\lambda\alpha}{2D}$, $\gamma=\frac{C_p}{C_p-zgR}$; S is the cross-sectional area of the pipe; Cp is the specific heat of gas; z is the gas compressibility coefficient; W (x, t), P (x, t), T (x, t) - specific mass flow rate, pressure and temperature, t; x– temporal and spatial coordinates; λ is the coefficient of hydraulic resistance; D is the diameter of the pipe; K is the heat transfer coefficient from the pipe to the ground; Tgr - soil temperature; h is the depth of the pipe; g is the acceleration of gravity.

The system of equations (1) - (3) for the j-th PS (the PS index is omitted) is written in matrix form:

$$\frac{\partial \varphi}{\partial t}+B\frac{\partial \varphi}{\partial x}=\Phi, \quad (4)$$

Where

$$B=\begin{bmatrix} 2\alpha TS\frac{W}{P} & 1-\alpha TS\frac{W^2}{P^2} & 0 \\ \alpha TS & 0 & 0 \\ \alpha(\gamma-1)S\frac{T^2}{P} & 0 & \alpha\gamma TS\frac{W}{P} \end{bmatrix}, \quad \Phi=\begin{bmatrix} -\beta TS\frac{W|W|}{P}-\frac{g}{\alpha S}\frac{P}{T}\frac{dh}{dx} \\ 0 \\ -\frac{4K}{D}(\gamma-1)\frac{T}{P}\left(T-T_{rp}\right)-\frac{g}{S}(\gamma-1)\frac{TW}{P}\frac{dh}{dx} \end{bmatrix}, \varphi=(W,P,T).$$

Conventionally, all types of valves (valves, valves and cranes) will be called cranes. As an MM linear tap when translating all units of parameters into the system SI units, it is proposed to choose a model that represents the equations of energy conservation and local pressure loss, which describe the modes of gas transport (RTG) through the f-th crane (f = 1, θ). The model has the following form:

$$P_K^f = P_H^f - \zeta \frac{Rg}{2(F_{\text{вых}}^f)^2} \frac{T_K^f z_K^f}{P_K^f} (G_H^f)^2, \tag{5}$$

$$T_K^f = T_H^f - D_j (P_H^f - P_K^f), \tag{6}$$

where PHf, PKf - pressure at the inlet and outlet of the f-th valve, respectively; ζ is the coefficient of local hydraulic resistance; D j is the Joule-Thomson coefficient; zfK is the compressibility coefficient; GfH - mass gas flow rate at the inlet of the f-th crane; Foutf is the cross-sectional area of the pipe behind the tap; THf, TKf - temperature at the inlet and outlet of the f-th crane.

Equation (5) describes the local pressure loss. Since the tap can be considered as local resistance, the Joule-Thompson effect, i.e. change in gas temperature during adiabatic throttling - slow gas flow under the action of a constant pressure drop through the throttle, a local obstacle to the gas flow. Given the small length of the crane section, we use the calculation formula

For the m-th intermediate node, the conditions for matching the gas flow parameters (for mass flow, pressure and temperature, respectively) take the following form:

$$\sum_{j \in V_m^+} G_j(x^{++}, t) = \sum_{i \in V_m^-} G_i(x^+, t), m \in V_2, \tag{7}$$

$$P_j(x^{++}, t) = P_i(x^+, t), j \in V_m^+, i \in V_m^-, \tag{8}$$

$$\sum_{j \in V_m^+} ((G_j(x^{++}, t))^+ \cdot T_j(x^{++}, t)) + \sum_{i \in V_m^-} ((G_i(x^+, t))^- \cdot T_i(x^+, t)) = T_{cp}^m \cdot (\sum_{j \in V_m^+} ((G_j(x^{++}, t))^+ + \sum_{i \in V_m^-} (G_i(x^+, t))^-), \tag{9}$$

Besides that:

if G (x_j^{++}, t)<0, то $T_j^{m}(x^{++}, t) = T_{cp}^{m}(t)$, $\in j$ V_m^+,

if G (x , t)>0i $^+$, то $T_i^{m}(x^{+}, t) = T_{cp}^{m}(t)$, $\in i$ V_m^-,

where $(a)^+ = \begin{cases} a, a \geq 0 \\ 0, a < 0 \end{cases}$, $(a)^- = \begin{cases} -a, a < 0 \\ 0, a \geq 0 \end{cases}$, x^+, x^{++} - the initial and final coordinate of the corresponding section; V is the set of network nodes; Vm +, Vm− - the set of indices of arcs entering and leaving the mth node of the network; G (x, t), T (x, t), P (x, t) - mass flow rate (kg / s), pressure (Pa) and temperature (K) for the j-th section; Tcpm (t) is the average temperature of the gas (K) flowing from the mth node. The structure of the crane model is presented in the form of a two-terminal device having one input and one output. Matching conditions in the fth node (f ∈ V3), which is the input of the fth tap, (f = 1, θ) have the following form: P (x ++, t) = PHf (t), W (x ++, t) S = GfKP (t), T (x ++, t) = THf (t), where

x ++ - the final coordinate of the corresponding section adjacent to the input of the f-th crane; S is the cross-sectional area of the pipe of the corresponding section adjacent to the entrance of the f-th crane; W (x, t), P (x, t), T (x, t) - specific mass flow rate, pressure and gas temperature of the section adjacent to the inlet of the f-th crane; GfKP (t) - mass gas flow through the f-th valve.

Thus, the general mathematical model of NSIMGT in LS MTP is an interconnected system of partial differential equations corresponding to each PS, and a system of nonlinear algebraic equations corresponding to each tap, which are interconnected by a system of linear algebraic equations corresponding to the conditions for matching the gas flow parameters in nodes of the graph.

In order for the system of equations (4) to be solvable, it is necessary to set the boundary conditions for the nodes corresponding to the inputs and outputs of the LS MTP. The boundary nodes of the 1st type are nodes for which pressure is specified as a function of time, and of the 2nd type - flow rate is specified as a function of time.

The boundary conditions for the m-th output and input nodes are: Gmv (t) = Gm (t) (type 2) or Puzm (t) = Pm (t) (type 1), in addition, the inlet gas temperature Tvm (t) = Tm (t) is set at the inputs. The initial distribution of flow rates, pressures and temperatures for LS: $W_j(x_j, 0) = W_j^0(x_j)$, $P_j(x_j, 0) = P_j^0(x_j)$, $T_j(x_j, 0) = T_j^0(x_j)$ where $x_j \in [x_j^+, x_j^{++}], \forall j \in M$

3.3.3 The method of solving the system of equations of the mathematical model of LS MTP

To obtain a numerical solution, system (4) is approximated by difference equations using an implicit finite-difference scheme defined on a five-point template (with second-order difference operators of approximation in spatial and temporal variables) (Fig. 3.5).

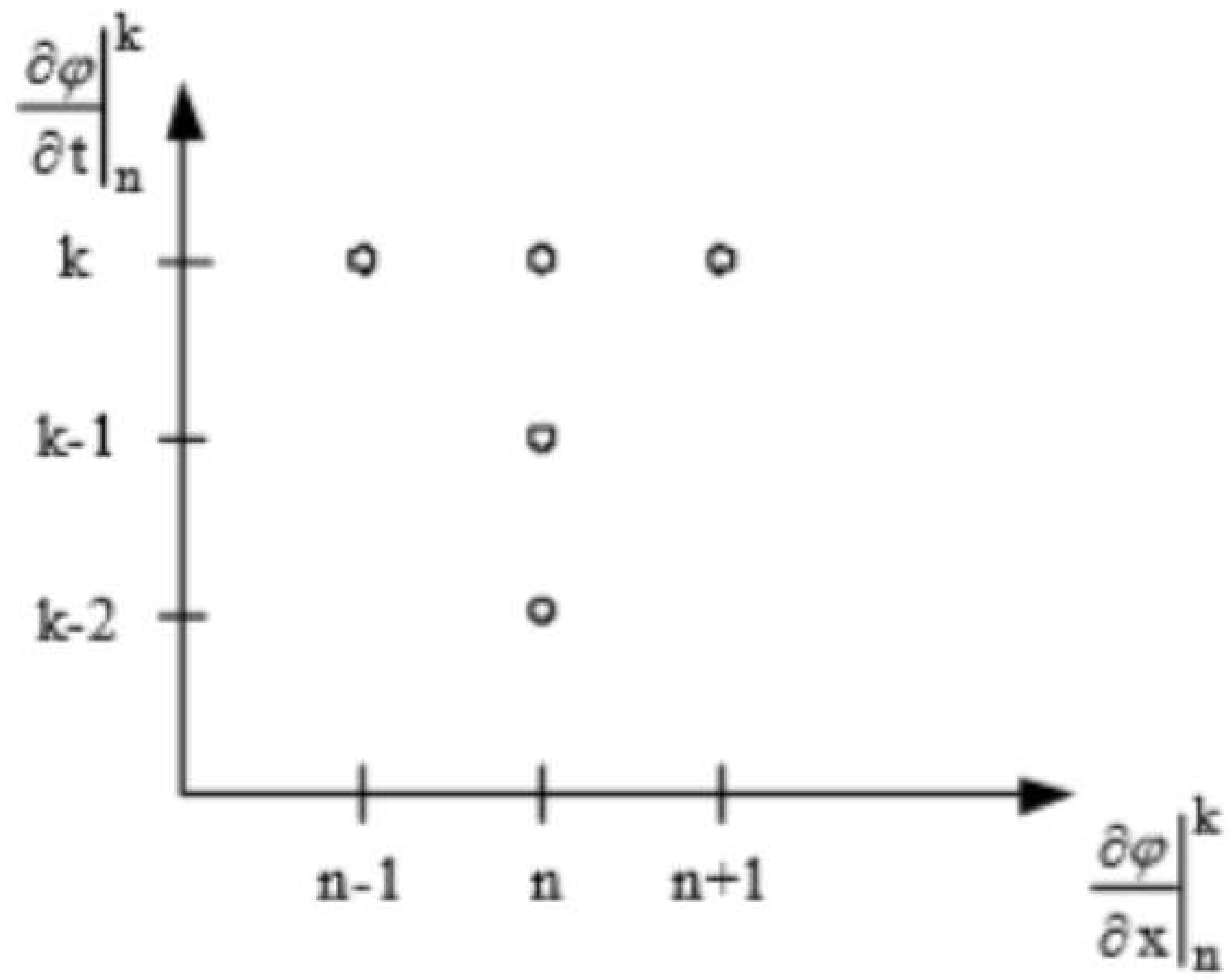


Fig. 3.5 Five-point pattern

We establish a uniform grid with constant steps in the spatial and temporal variables and approximate the system of differential equations by the following relations:

$$\left.\frac{\partial \varphi}{\partial t}\right|_{n}^{k}=\frac{3\varphi_{n}^{k}-4\varphi_{n}^{k-1}+\varphi_{n}^{k-2}}{2\Delta t}, n=\overline{0,N}, \tag{10}$$

$$\left.\frac{\partial \varphi}{\partial x}\right|_{n}^{k}=\begin{cases}\dfrac{\varphi_{1}^{k}-\varphi_{0}^{k}}{\Delta x}, & n=0,\\ \dfrac{\varphi_{n+1}^{k}-\varphi_{n-1}^{k}}{2\Delta x}, & n=\overline{1,N-1},\\ \dfrac{\varphi_{N}^{k}-\varphi_{N-1}^{k}}{\Delta x}, & n=N.\end{cases} \tag{11}$$

Taking into account (10) - (11), the system of equations for the j-th section of the pipeline takes the form:

$$\frac{3}{2\Delta t}\varphi_{0}^{k}-\frac{1}{\Delta x}B_{0}^{k}\varphi_{0}^{k}+\frac{1}{\Delta x}B_{0}^{k}\varphi_{1}^{k}=\Phi_{0}^{k}+\frac{2}{\Delta t}\varphi_{0}^{k-1}-\frac{1}{2\Delta t}\varphi_{0}^{k-2}, \tag{12}$$

$$-\frac{1}{2\Delta x}B_{n}^{k}\varphi_{n-1}^{k}+\frac{3}{2\Delta t}\varphi_{n}^{k}+\frac{1}{2\Delta x}B_{n}^{k}\varphi_{n+1}^{k}=\Phi_{n}^{k}+\frac{2}{\Delta t}\varphi_{n}^{k-1}-\frac{1}{2\Delta t}\varphi_{n}^{k-2}, n=\overline{1,N_{j}-1}, \tag{13}$$

$$\frac{3}{2\Delta t}\varphi_{N_j}^{k}+\frac{1}{\Delta x}B_{N_j}^{k}\varphi_{N_j}^{k}-\frac{1}{\Delta x}B_{N_j}^{k}\varphi_{N_j-1}^{k}=\Phi_{N_j}^{k}-\frac{2}{\Delta t}\varphi_{N_j}^{k-1}-\frac{1}{2\Delta t}\varphi_{N_j}^{k-2}. \tag{14}$$

Thus, we obtained a system of nonlinear algebraic equations that contains 3 (N + 1) equations and 3 (N + 1) variables. This system is linearized by the Newton method. The resulting linear system for the kth time layer, rth iteration, and jth UT is written in iterative form:

$$\left[\frac{\partial \psi}{\partial \varphi}\right]_0^{k,r,j-1} \delta\varphi_0^{k,r,j} + \frac{1}{\Delta} B_0^{k,r,j-1} \delta\varphi_1^{k,r,j} = \psi_0^{k,r,j-1},$$

$$\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots$$

$$-\frac{1}{2\Delta} B_n^{k,r,j-1} \delta\varphi_{n-1}^{k,r,j} + \left[\frac{\partial \psi}{\partial \varphi}\right]_n^{k,r,j-1} \delta\varphi_n^{k,r,j} + \frac{1}{2\Delta} B_n^{k,r,j-1} \delta\varphi_{n+1}^{k,r,j} = \psi_n^{k,r,j-1}, n = \overline{1,\ N_j-1} \quad (15)$$

$$\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots$$

$$\left[\frac{\partial \psi}{\partial \varphi}\right]_{N_j}^{k,r,j-1} \delta\varphi_{N_j}^{k,r,j} + \frac{1}{\Delta} B_{N_j}^{k,r,j-1} \delta\varphi_{N_j-1}^{k,r,j} = \psi_{N_j}^{k,r,j-1},$$

Where $\delta\varphi_0^{k,r,j}, \ldots, \delta\varphi_{N_j}^{k,r,j}$ correction vectors to unknowns; $\psi_0^{k,r,j-1}, \ldots, \psi_{N_j}^{k,r,j-1}$ - vectors residuals at the corresponding points in space;

$\left[\frac{\partial \psi}{\partial \varphi}\right]_0^{k,r,j-1}, \ldots, \left[\frac{\partial \psi}{\partial \varphi}\right]_{N_j}^{k,r,j-1}$ - Jacobi matrices at the corresponding points in space.

To solve the system of linear algebraic equations (15), it is necessary to calculate the residual vectors for each j-th pipeline section:

$$\psi_0^{k,r,j} = \frac{3}{2\Delta t}\varphi_0^{k,r,j} - \frac{1}{\Delta x} B_0^{k,r,j} \varphi_0^{k,r,j} + \frac{1}{\Delta x} B_0^{k,r,j} \varphi_1^{k,r,j} - \Phi_0^{k,r,j} - \frac{2}{\Delta t}\varphi_0^{k-1,j} + \frac{1}{2\Delta t}\varphi_0^{k-2,j},$$

$$\psi_n^{k,r,j} = -\frac{1}{2\Delta x} B_n^{k,r,j} \varphi_{n-1}^{k,r,j} + \frac{3}{2\Delta t}\varphi_n^{k,r,j} + \frac{1}{2\Delta x} B_n^{k,r,j} \varphi_{n+1}^{k,r,j} - \Phi_n^{k,r,j} - \frac{2}{\Delta t}\varphi_n^{k-1,j} + \frac{1}{2\Delta t}\varphi_n^{k-2,j}, n = \overline{1, N_j-1},$$

$$\psi_{N_j}^{k,r,j} = \frac{3}{2\Delta t}\varphi_{N_j}^{k,r,j} + \frac{1}{\Delta x} B_{N_j}^{k,r,j} \varphi_{N_j}^{k,r,j} - \frac{1}{\Delta x} B_{N_j}^{k,r,j} \varphi_{N_j-1}^{k,r,j} - \Phi_{N_j}^{k,r,j} - \frac{2}{\Delta t}\varphi_{N_j}^{k-1,j} + \frac{1}{2\Delta t}\varphi_{N_j}^{k-2,j}.$$

The residuals compute the elements of the Jacobi matrix $\left[\frac{\partial \psi}{\partial \varphi}\right]_0^{k,r,j}$ at starting point,

$\left[\frac{\partial \psi}{\partial \varphi}\right]_n^{k,r,j}$ at internal points $(n = \overline{1, N_j-1})$ and $\left[\frac{\partial \psi}{\partial \varphi}\right]_{N_j}^{k,r,j}$ at the end point N_j;

As a result, we simplify the obtained linear system of equations (15) using the matching conditions. The conditions for coordination (7) - (8) are linearized:

$$\sum_{j \in V_m^+} S_j \delta W_{N_j}^{k,r,j} = \sum_{i \in V_m^-} S_i \delta W_i^{k,r,i}, m \in V_2, \tag{16}$$

$$\delta P_{N_j}^{k,r,j} = \delta P_i^{k,r,i}, j \in V_m^+, i \in V_m^-. \tag{17}$$

In accordance with the proposed method, from condition (16), it is necessary to choose a variable for the specific mass flow rate $\delta W_{N_{j0}}^{k,r,j0}$, equal;

$$\delta W_{N_{j0}}^{k,r,j0} = \frac{\sum_{i \in V_m^-} S_i \delta W_i^{k,r,i}}{S_{j0} \sum_{\substack{j \in V_m^+ \\ j \neq j0}} S_j \delta W_{N_j}^{k,r,j}}, m \in V_2,$$

and exclude it from the system of equations. In this case, for intermediate nodes, the equation in which the excluded variable in specific mass flow rate was located is added to the equation with a similar variable in specific mass flow rate. From condition (17) for the intermediate nodes of the MTP graph, it is necessary to leave the variable $\delta P_{j0}^{k,r,j0}, j_0 \neq j_f$, and exclude all other variables from the system of equations, except for variables related to the pipeline sections on which the taps are located. Moreover, for intermediate nodes, the equations in which the excluded pressure variables were located are added to the equation with a similar pressure variable. The joint calculation of the modes of transport of UT gas through taps is carried out in the following way. For all nodes that are exits from the cranes, the corresponding equations for determining the residuals at the 0 point of the jth section in the UT are replaced by the residual equations of the crane model (the crane index is omitted for convenience):

$$\psi_{0,2\ кр}^{k,r,j} = P_H^{k,r,j} - P_K^{k,r,j} - \zeta \frac{Rg}{2(F_{вых}^{k,r,j})^2} \frac{T_K^{k,r,j} z_K^{k,r,j}}{P_K^{k,r,j}} (G_H^{k,r,j})^2,$$

$$\psi_{0,3\ кр}^{k,r,j} = T_H^{k,r,j} - T_K^{k,r,j} - D_j (P_H^{k,r,j} - P_K^{k,r,j}),$$

The matching conditions for the f-th valve are linearized:

$$\delta P_{N_j}^{k,r,j} = \delta P_H^f(t),\ \delta G_{KP}^f(t) = \delta W_{N_j}^{k,r,j} S_j,\ \delta T_{N_j}^{k,r,j} = \delta T_H^f(t),$$

$$\delta P_i^{k,r,i} = \delta P_K^f(t),\ \delta G_{KP}^f(t) = \delta W_i^{k,r,i} S_i,\ \delta T_i^{k,r,i} = \delta T_K^f(t).$$

Further, the variable in specific mass flow rate related to the input or output from the tap is also excluded from the system of equations in the following way. Replace the variable $\delta W_{N_j}^{k,r,j}$ in the system of equations in accordance with the matching conditions for the f-th valve according to the formula; $\delta W_{N_j}^{k,r,j} = \frac{S_i}{S_j} \cdot \delta W_i^{k,r,i}$. At the same time, for the nodes that are the input and output from the tap, the equation in which the excluded variable in specific mass flow was located is added to the equation with a similar variable in specific mass flow. After transformations, the system of equations is solved with respect to the vectors of corrections to unknowns using the Gauss method with the choice of the main element. At each step of the iterative process, after finding the gas flow parameters on each time layer, in accordance with formula (11), we calculate the average gas temperature in the MTP nodes.

3.3.4 Analysis example

Consider the example of calculating NSIMGT on MTP through stop valves taking into account the dynamics of closing the crane. The design diagram and graph of the MTP under consideration are presented in Fig. 2 and 3, respectively.

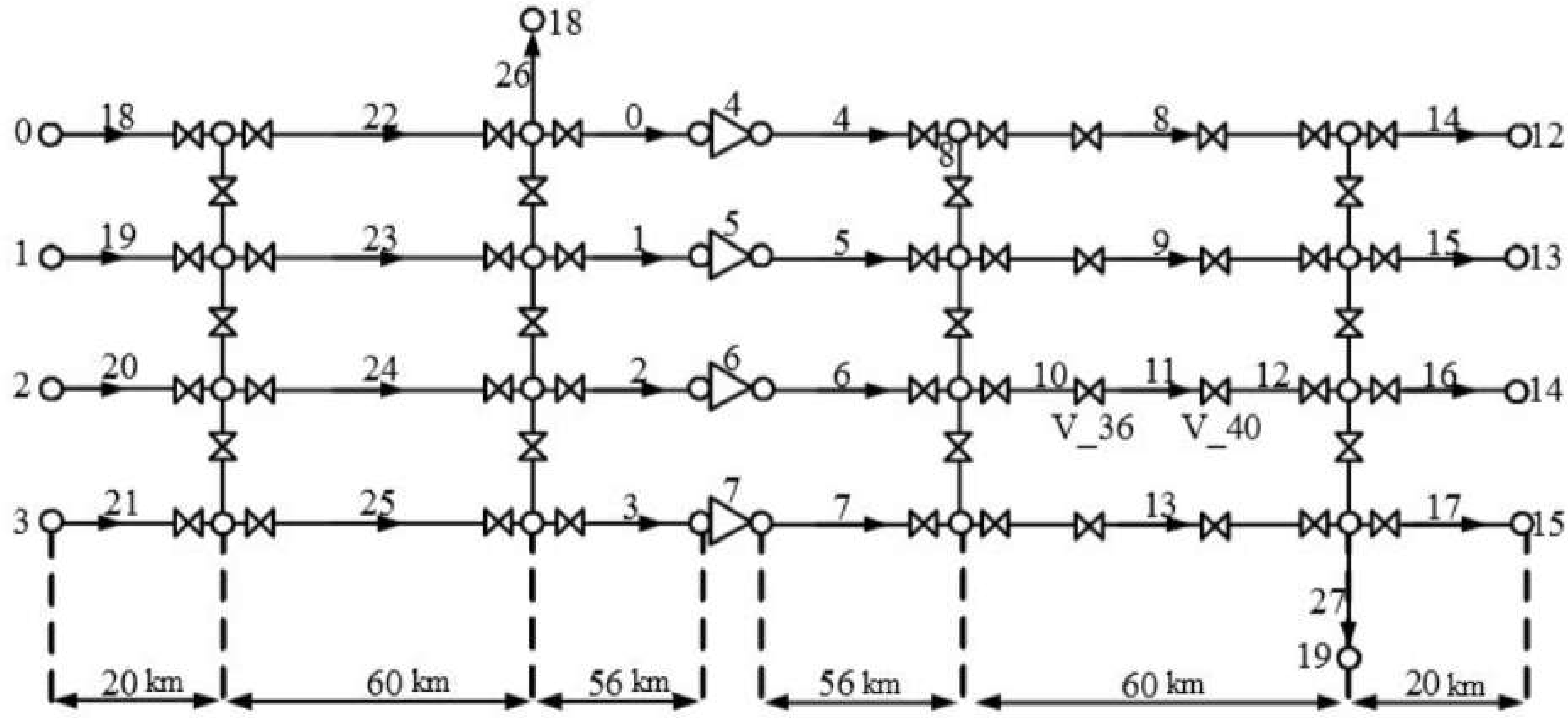


Fig. 3.6 The design scheme of a multi-line main gas pipeline

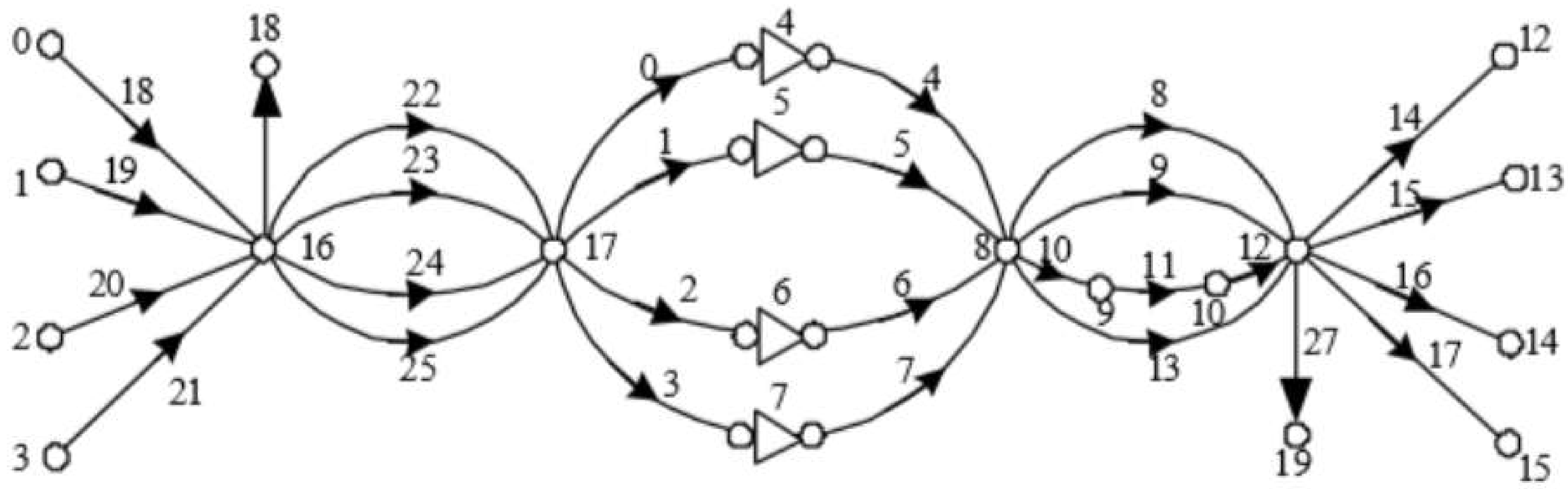


Fig. 3.7 Count of the design scheme of a multi-line main gas pipeline

As the initial condition, we take the stationary gas flow with a pressure of P = 84.636 atm., A temperature of T = 40 C 0, and a total commercial consumption of natural gas equal to $265*10^6$ m^3/day.

Table 1. Main parameters, their designations and values

Designation	Number value	Parameter name
Pipeline characteristics		
D_i	1400	Internal pipeline diameter

h	10	Wall thickness
Cp	0,655952	Specific heat, kcal / (kg·^{0}C)
K	1,4	The heat transfer coefficient from the pipe to the ground, kcal / (m^2·hour·^{0}C)
Δ	0,604707	Relative gas density in air
tгр	10	Soil temperature at the depth of the gas pipeline, C^0

In the test example, the 36 V_ and 40 V_ taps were linearly closed within 2 minutes after 30 and 45 minutes. after the start of the calculation, respectively, and the rest were supposed to be completely open. The graph of the cross-sectional area of the crane when it is closed on time is shown in Fig. 4. The boundary conditions are presented in table. 2. Compressor workshops were located in nodes 4–7 of the computational graph, the modeling of which was carried out according to the algorithm presented in. The calculation was carried out for 24 hours with a difference grid time step of 30 s and a space step of 10 km. As a result of the calculation, we obtain the distribution of gas flow parameters for the MTP under consideration. The simulated transient ends in flow and pressure in 1200 minutes, and in temperature in 24 hours. The dynamics of closing the crane was simulated by changing the cross-sectional area of the crane over time (Fig. 3.8).

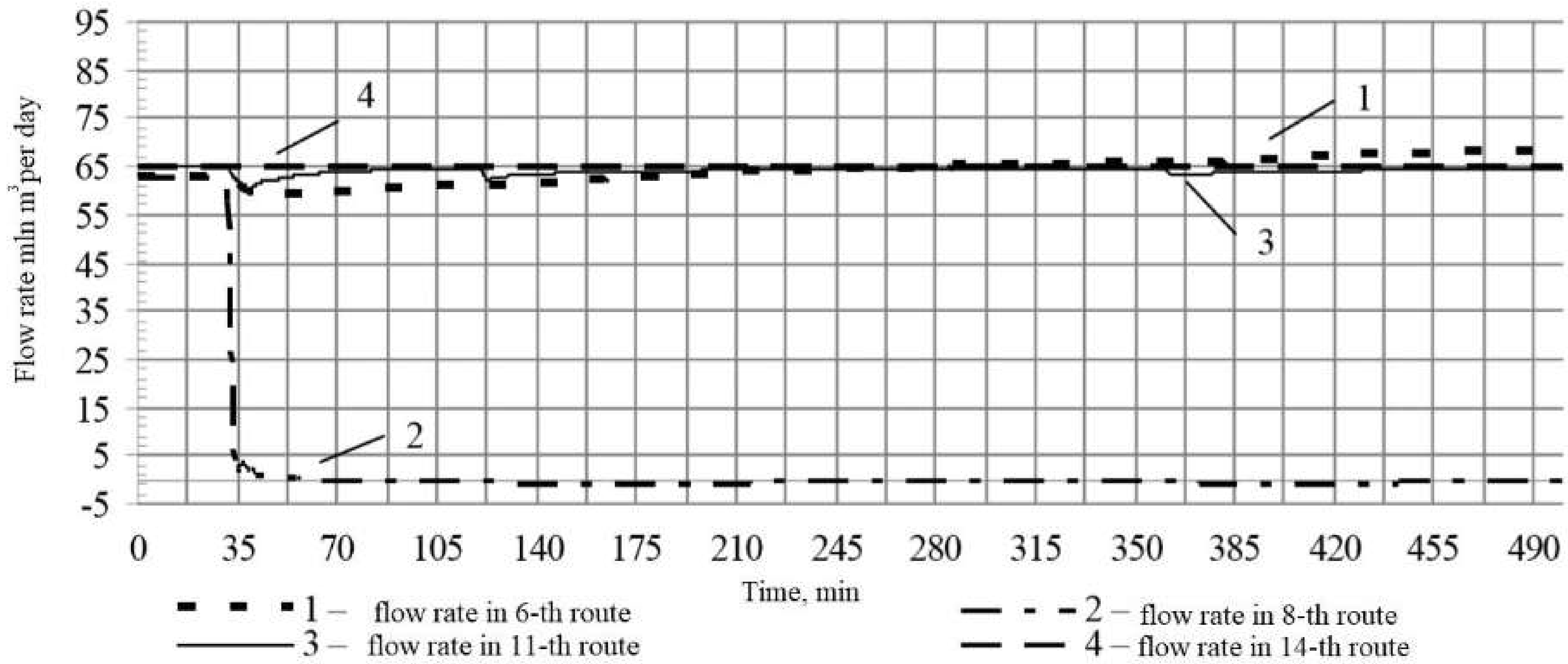


Fig. 3.8 The graph of the dependence of the cross-sectional area of the crane when it is closed on time

Table 2. Boundary conditions

Network routes	Values in the system routes	Network routes	Values in the system routes
0	P(t) = 8,4 MPa , T(t) = 40^{0}C	13	$q\ (t)^{13}$ = 60 mln.m^3/day, $t \geq 0$ min.
1	P(t) = 8,4 MPa, T(t) = 40^{0}C	14	$q\ (t)^{14}$ = 65 mln.m^3/day, $t \geq 0$ min.
2	P(t) = 8,4 MPa, T(t) = 40^{0}C	15	$q\ (t)^{15}$ = 60 mln.m^3/day, $t \geq 0$ min.
3	P(t) = 8,4 MPa, T(t) = 40^{0}C	18	$q\ (t)^{18}$ = 15 mln.m^3/day, $t \geq 0$ min.
12	$q^{12}(t)=\begin{cases}55\text{ mln. m3/day. } t \leq 360\text{ min.}\\70\text{ mln. m3/day. } t > 360\text{ min.}\end{cases}$	19	$q^{19}(t)=\begin{cases}10\text{ mln. m3/day. } t \leq 120\text{ min.}\\30\text{ mln. m3/day. } t > 120\text{ min.}\end{cases}$

In fig. 3.8 – 3.10 are graphs of the dependence of gas flow parameters on time for the 3rd string (on which cranes 36 V_ and 40 V_ are located) at the locations of the crane platforms.

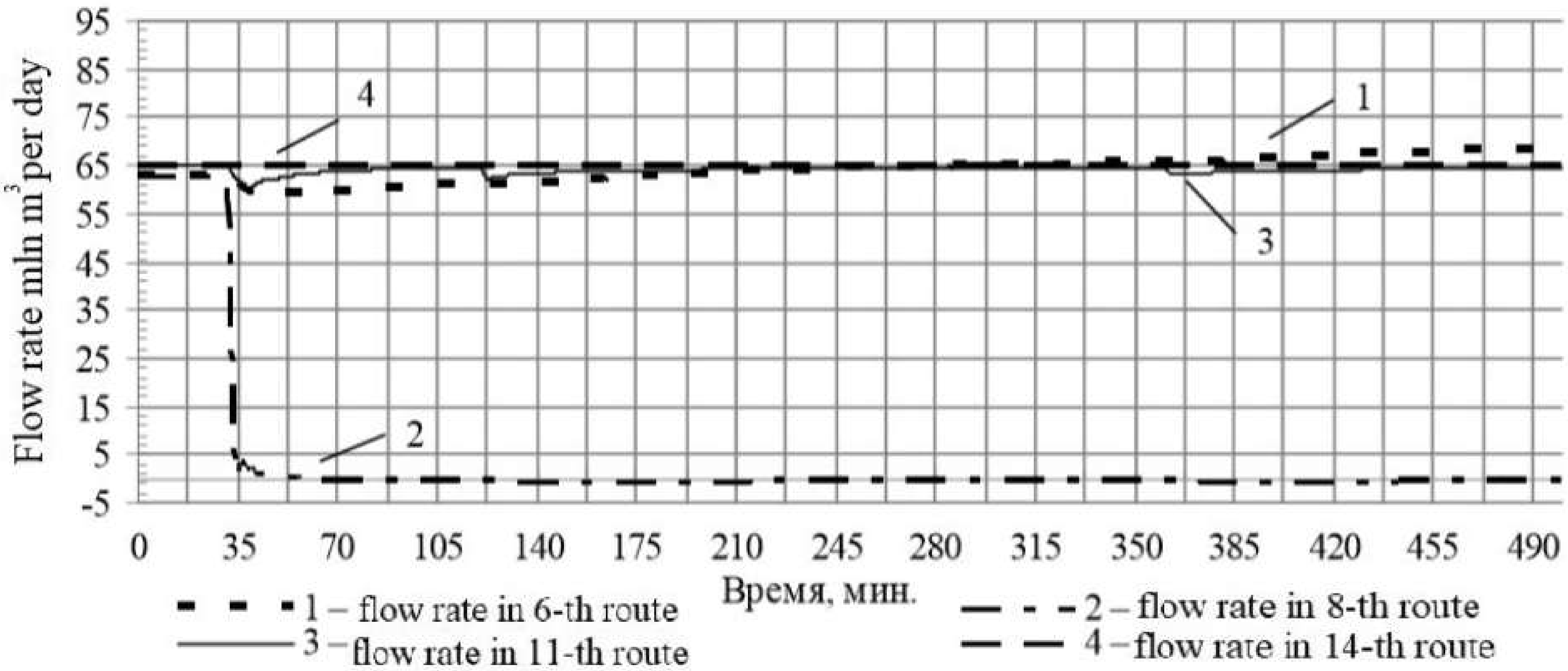


Fig. 3.8. The graph of the flow rate versus time in MTP nodes (3rd thread)

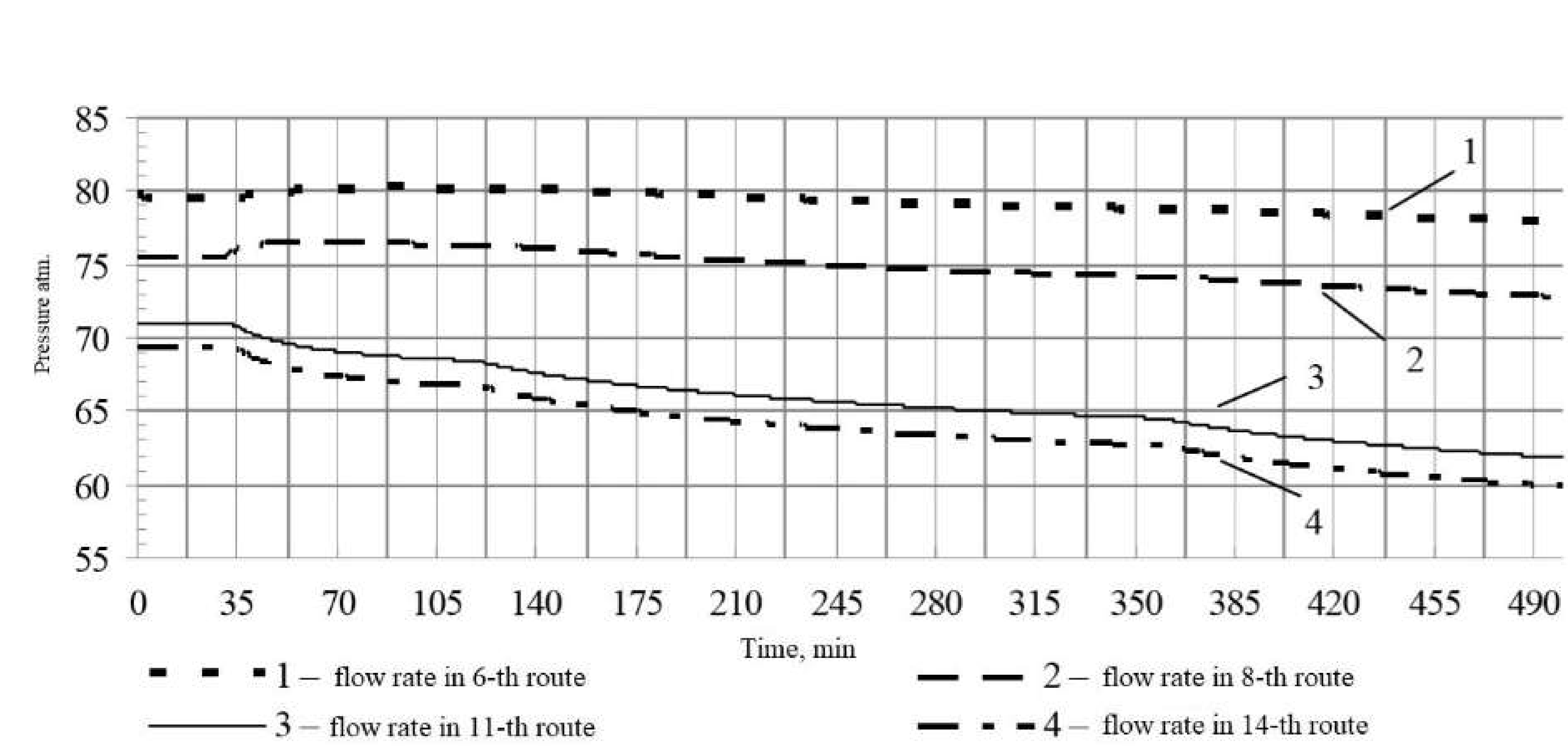


Fig. 3.9. The graph of the dependence of pressure on time in nodes MTP (3rd thread)

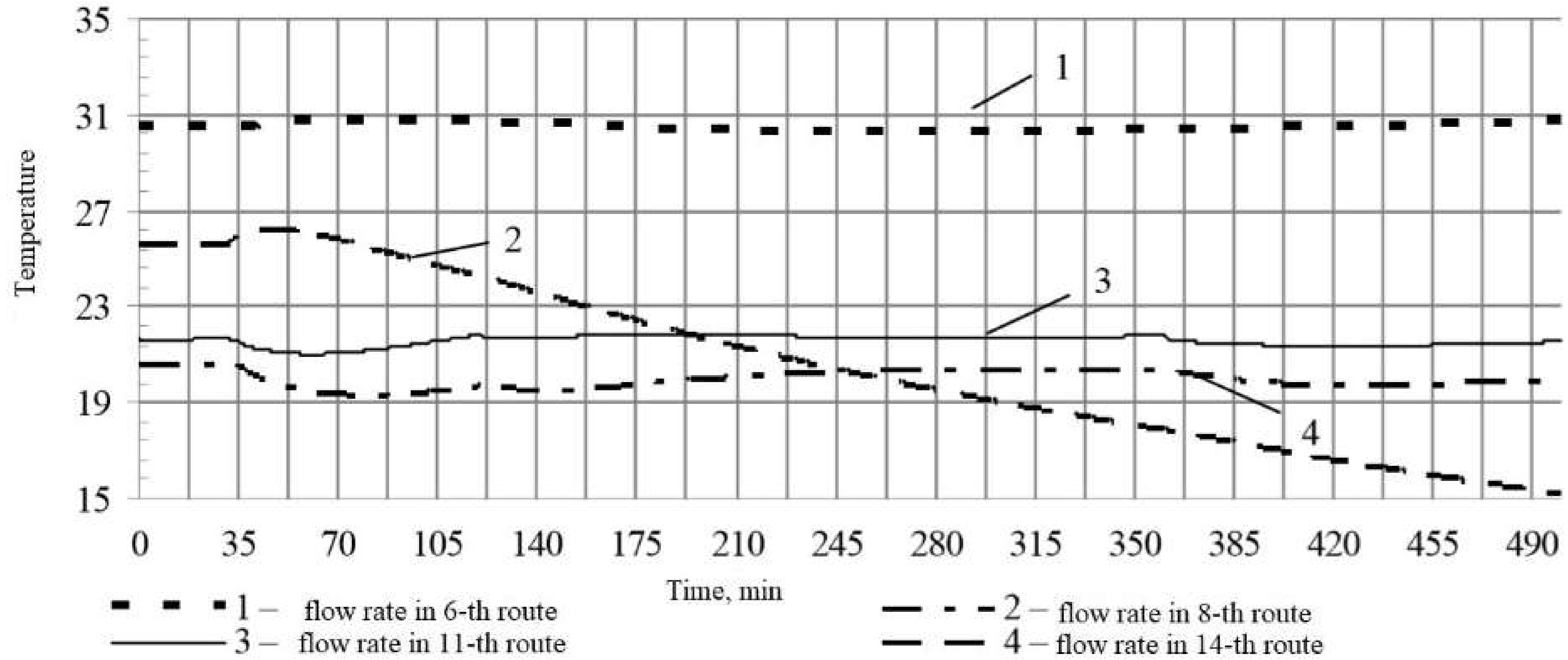


Fig. 3.10 The graph of temperature versus time in MTP nodes (3rd thread)

Conclusion to chapter 3

1. method for calculating NSIMGT in LS MTP was proposed, the mathematical model of which includes mathematical models of pipeline sections and stop valves, which differs from existing calculation methods by the method of including and excluding equations of models of all MTP elements in the general system of equations of the general mathematical

model and allowing high accurately calculate the real processes of gas flow in MTP in real time, as well as predict various situations in order to adopt the necessary control action. A system of differential equations in partial derivatives describing NSIMGT was taken as a mathematical model for UT.

2. Experimental studies conducted on various test examples showed that the proposed method for calculating NSIMGT in MTP through shut-off valves can be used to simulate transients in main gas pipelines taking into account the time of complete shutoff of the valves. Practical relevance - taking into account the action of stop valves on MTP allows the most adequate description of NSIMGT on LS in MTP. The importance of the developed method lies in the fact that the calculation of NSIMGT on MTP taking into account the dynamics of the valves will allow us to further model emergency situations in order to timely localize accidents on gas pipelines.

3. Using of modern technologies in CS constructing, will allow to incresethe pressure on the exit of CS with combined cycle power plants instead of single stage engines.

CHAPTER 4. ENVIRONMENTAL PROTECTION

4.1 Environmental protection of gas transmission systems at the industrial level.

Environmental safety of gas transmission facilities is directly related to reliability System. Many measures to improve the reliability of the gas transmission system must be implemented considered as a way to reduce the risk of human injury and death, as well as environmental damage, i.e. increase security.

The efficiency and safety of the gas transmission system are largely determined environmental compatibility of the main facilities of the system, one of which is a compressor station. This is exactly the place where the largest number of high-performance devices should be involved. technological process of transportation of natural gas is concentrated, branched systems of technological a large number of service personnel are involved in communication functions. The main sources of contaminants are gas compressors driven by gas turbines.

Centrifugal compressors, oil degassers and processes associated with their operation.

4.2 Main environmental factors of a gas compressor installation.

Consider the main factors the impact of the gas pumping unit on the environment: emissions of pollutants into the atmosphere;

- Toxic waste;
- Noise and vibration;
- heat exposure.
- Air pollutant emissions.

There are natural gas emissions and emissions of combustion products (gases from a gas turbine engine). Table 1. shows the parameters technical standards for the emission of pollutants into gas compressor units experimental. Natural gas is emitted into the compressor station when starting and stopping the gas compressor. The device, as well as during operation with leakage of its main parts. Exhaust gases result from the combustion of natural gas in a gas turbine of a gas pumping unit. They consist of: combustion products - nitrogen, water vapor, carbon dioxide; nitrogen oxides; carbon dioxide; Sulfur oxides; Hydrocarbons (also not completely burnt)(Methane); Soot. Not when burning gases containing hydrogen sulfide, sulfuric acid and sulfur-containing anhydrides burnt hydrogen sulfide is also released into the atmosphere. The maximum amount is set in Directive 2001/80 / EC of the European Parliament and of the Council. Emission values for gas turbine engines from previously certified gas pumping unitsStarted on November 27, 2002 and before November 27, 2003: with nitrogen oxides 50 mg / m3 use of natural gas with an inert gas content of 20% or less as fuel; nitrogen oxides 75 mg / m3 for gas turbine engines in cogeneration plants with a total efficiency of more 75% for gas turbines in thermal power plants with a total average annual efficiency more than 55%.

4.2.1 Toxic waste

The gas compressor unit has a gas turbine engine lubrication system and centrifugal compressor. Use hydraulic centrifugal loader shaft seal systems. Different types of lubricating oil with oil storage tanks for engine and centrifugal compressor respectively. During operation of the gas compressor unit, toxic substances are released from the oil tanks of the turbine engine and the centrifugal compressor from the oil degasser of the compressor seal
the system. Toxic substances that come from the gland system of the centrifugal fan due to the pressure difference in the charger and degasser, which is below atmospheric pressure, pressure is released in the degasser. Toxic substances are released into the air from oil tanks a gas turbine engine and a centrifugal supercharger in which oil is heated in the engine and loader pumps.

Irreversible oil losses should be avoided in gas pump units with an output of up to 10 MW exceed: for stationary gas turbine systems - 1.0 kg / h; with modified engines - 2.0 kg / h; for gas pump units with an output of more than 10 MW: for stationary gas turbine systems - 1.5 kg / h; with modified engines - 2.5 kg / h.

Table 4.1 - Emissions of polluting substances of gas-compressor units.

Parameter	Parameter values				
Power of gas-compressor unit [MW]	2 ÷ 4	6 ÷ 8	10 ÷ 13	16 ÷ 18	22 ÷ 31
Fuel gas consumption [m3/h]	1106 ÷ 1795	1824 ÷ 2872	3077 ÷ 5051	4720 ÷ 6593	6889 ÷ 10539
Temperature of combustion products [°K]	1113 ÷ 1270	947 ÷ 1295	1053 ÷ 1456	1130 ÷ 1456	1188 ÷ 1518
Concentration in dry combustion products:					
– nitrogen oxide [mg/m3]	35 ÷ 136	69 ÷ 202	69 ÷ 199	48 ÷ 179	50 ÷ 353
– carbonic oxide [mg/m3]	25 ÷ 82	33 ÷ 239	29 ÷ 275	48 ÷ 229	41 ÷ 441
Modified concentration:					
– nitrogen oxide [mg/m3]	50 ÷ 195	135 ÷ 490	100 ÷ 230	80 ÷ 250	50 ÷ 400
– carbonic oxide [mg/m3]	30 ÷ 130	90 ÷ 300	60 ÷ 300	80 ÷ 300	50 ÷ 500
Emission efficiency:					
– nitrogen oxide [g/s]	0.52 ÷ 1.67	2.58 ÷ 6.03	2.38 ÷ 7.81	3.29 ÷ 11.8	2.81 ÷ 26.65

4.2.2 Heat protection

The use of gas turbine engines in the drive of a gas pump unit also leads to thermal effects on the environment and the occurrence of geo-ecological factors. Thermal impact the compressor station for the environment is particularly relevant when operating in permafrost.

4.2.3 The problem of saving energy

Gas consumption for personal use, which is mainly covered with fuel gas up to 5% of the transport volume can be incurred in gas turbine engines of gas compressor units 1000 km of the route. Therefore in absolute values with a transport range of more than 5000km The gas consumption for the additional needs of the compressor station will be very high. solution

This problem has several main aspects: Reduction of energy consumption for gas compression due to use a gas pump unit with high efficiency of the gas turbine engine and centrifugal fan; Pressure reduction during gas transport in the gas line through: use of tubes with a smooth inner coating; regular cleaning of the inside of the pipeline; use of heavy duty equipment and gas line of the compressor station; Total energy savings consumption through the use of energy-saving technologies.

When developing gas transport technology, the energy saving of gas is the same factor as other important resource-saving factors related to investments, investments in metals and exploitation expenditure.

The choice of gas transport technology depends primarily on the ratio of the pipe costs and compressor station equipment on the one hand and gas used for own use the other hand. The final decision is made by optimizing the economic indicators. Characterize the effectiveness of the project and the environmental safety of the project.

4.3 Recommendations for reducing environmental harm

Methane is a greenhouse gas that also contributes to the formation of ozone, an air pollutant that affects human health and damages plants. Less methane in the atmosphere means better air quality and less greenhouse gases.

To reduce methane, gas sector can minimize ventilation. the waste sector can minimize solid waste in landfills; and the agricultural sector can reduce emissions from rice production - these are some options. Gas as industry can take measures to reduce methane emissions.

Then we have fluorocarbons (HFCs), a group of industrial chemicals that are mainly used for cooling and cooling. There are not many HFCs in the atmosphere at the moment, but they have one of the highest emissions among all greenhouse gases. If we had reduced their emissions (as the countries agreed in the Kigali amendment to the Montreal Protocol), we could have avoided warming to 0.5 degrees by 2100.

Emissions can be reduced by replacing HFCs with other refrigerants such as hydrocarbons or ammonia. or by improving insulation and building structure to reduce our dependence on air conditioning.

4.3.1 Noise reduction

Various methods are used to reduce noise in production facilities: reducing noise at the source; sound absorption and sound insulation; installation of silencers; rational placement of equipment; the use of personal protective equipment.

The most effective way of dealing with noise is at its source. The noise of the mechanisms is due to the elastic vibrations of both the whole mechanism and its individual parts. The causes of noise are mechanical, aerodynamic and electrical phenomena, which are determined by the structural and technological features of the equipment, as well as the operating conditions. In this regard, there are mechanical, aerodynamic and electrical noise. In order to reduce mechanical noise, it is necessary to repair equipment in a timely manner, replace the impact processes with the non-stressed ones, apply forced lubrication of friction surfaces more widely, and apply balancing of rotating parts.

Significant noise reduction is achieved when the rolling bearings are replaced by sliding bearings (noise is reduced by 15 dB), gears and chain transmissions with V-belt and gear-belt transmissions, metal parts - plastic parts, use of lubricants.

Reduced aerodynamic noise can be achieved by reducing the speed of the gas stream, improving the aerodynamics of the structure, sound insulation and installation of silencers. Electromagnetic noise is reduced by structural changes in electric machines.

The methods of noise reduction on the way of its spread are widely used by means of installation of sound insulation and sound absorbing obstacles in the form of screens, partitions, housings, cabins, facing of walls, ceilings, use of silencers, etc.

Soundproof enclosures cover the most noisy machines and mechanisms, thus localizing the noise source. For a heat-generating machine (electric motors, compressors, etc.), the housings provide ventilation devices with silencers. The casing should close the noise source tightly, but it should not be rigidly connected to the mechanism, since this gives a negative effect - the casing becomes an additional noise source.

Screens are installed between the noise source and the workplace. The acoustic effect of the screen is based on the formation of a shadow region where the sound waves only partially penetrate. The degree of penetration depends on the ratio between the screen size and the wavelength: the longer the wavelength, the smaller the screen shadow area behind the screen, and therefore the smaller the noise reduction. Therefore, the screens are used mainly for protection against medium and high frequency noise, and at low frequencies they are ineffective, since due to the diffraction effect, the sound easily bends them. Also important is the distance from the noise source to the screened workstation: the smaller it is, the greater the screen performance. The screen is effective when there are no reflected waves, ie either outdoors, or indoors (figure 5.1).

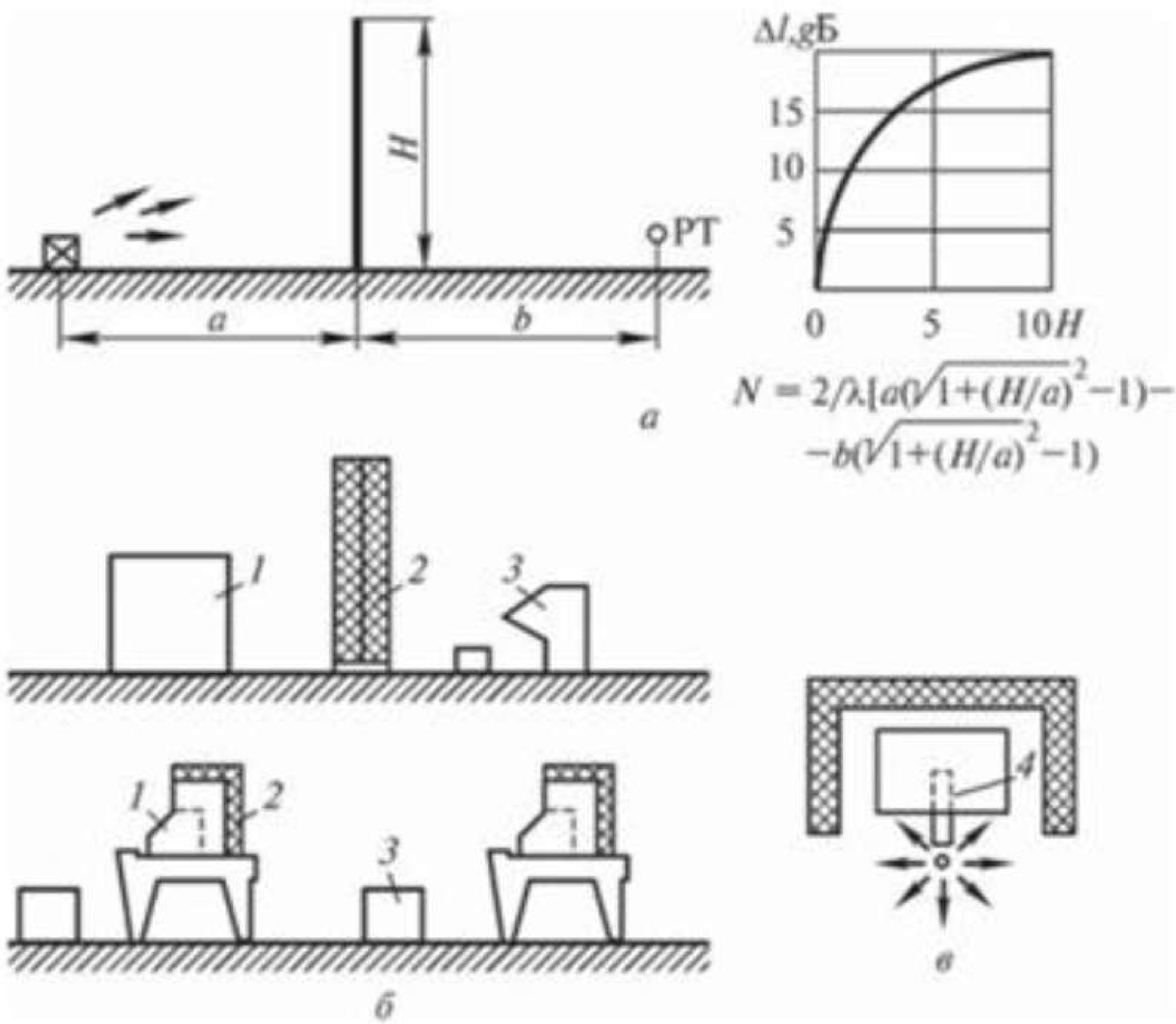


Figure 4.1 - shielding of noise sources.

4.3.2 Hazardous leakage prevention measures

Gas leak detection is a big problem. This is because some gases are colorless and odorless. If such gases escape, the detection takes a long time. Even if a leak is discovered, it is often difficult to determine the cause of the leak.

Methane emissions accounted for 2.8% of total greenhouse gas emissions in the energy sectors in 2013. Many companies have installed methane sensors to detect methane leaks. The leak source can be easily identified by installing several methane sensors. Methane sensors are not the only gas sensors used. Carbon monoxide and carbon dioxide sensors are also widely used. The industry is developing sensors for other toxic gases.

Establishment of industry-funded security institutions

Effective control of oil spills requires huge investments. Individual companies cannot adequately fund research into new spill prevention methods. To overcome this limitation, companies pool resources.

An example of a company sponsored by the industry is the Maritime Safety Center (COS). COS is funded by the US offshore oil and gas industry. The center prepares information and recommendations on safety and environmental management systems (SEMS). It also promotes cooperation between industry and external stakeholders, such as the government and local communities, in the fight against pollution.

4.3.3 Heat emission reduction

Elements of the organic Rankine cycle (ORC) consist of a heat exchanger in the exhaust system of a gas turbine, an expansion turbine that is driven by an electrical generator, a water- or air-cooled condenser, a working medium and a feed pump (Figure 1). Air cooling is usually preferable because water supply may not be available and air cooling avoids frost problems. The circuit is sealed and closed without the need for makeup fluid. A suitable working medium should be selected using parametric analysis in order to achieve the highest thermodynamic properties and minimal environmental impact during the cycle (Zeki Yilmazoglu et al., 2014).

The waste heat recovery system can be built from standard components and readily available components. For example, turbo expanders are widely used in the gas industry and are available in various sizes and performance features. Similarly, heat exchangers are available from several specialized organizations that are able to design and supply complete organic packaging for turbines of various sizes.

The performance of a single-stage Rankine cycle can be increased by using a cascade system with an additional stage of expansion, in series with the main

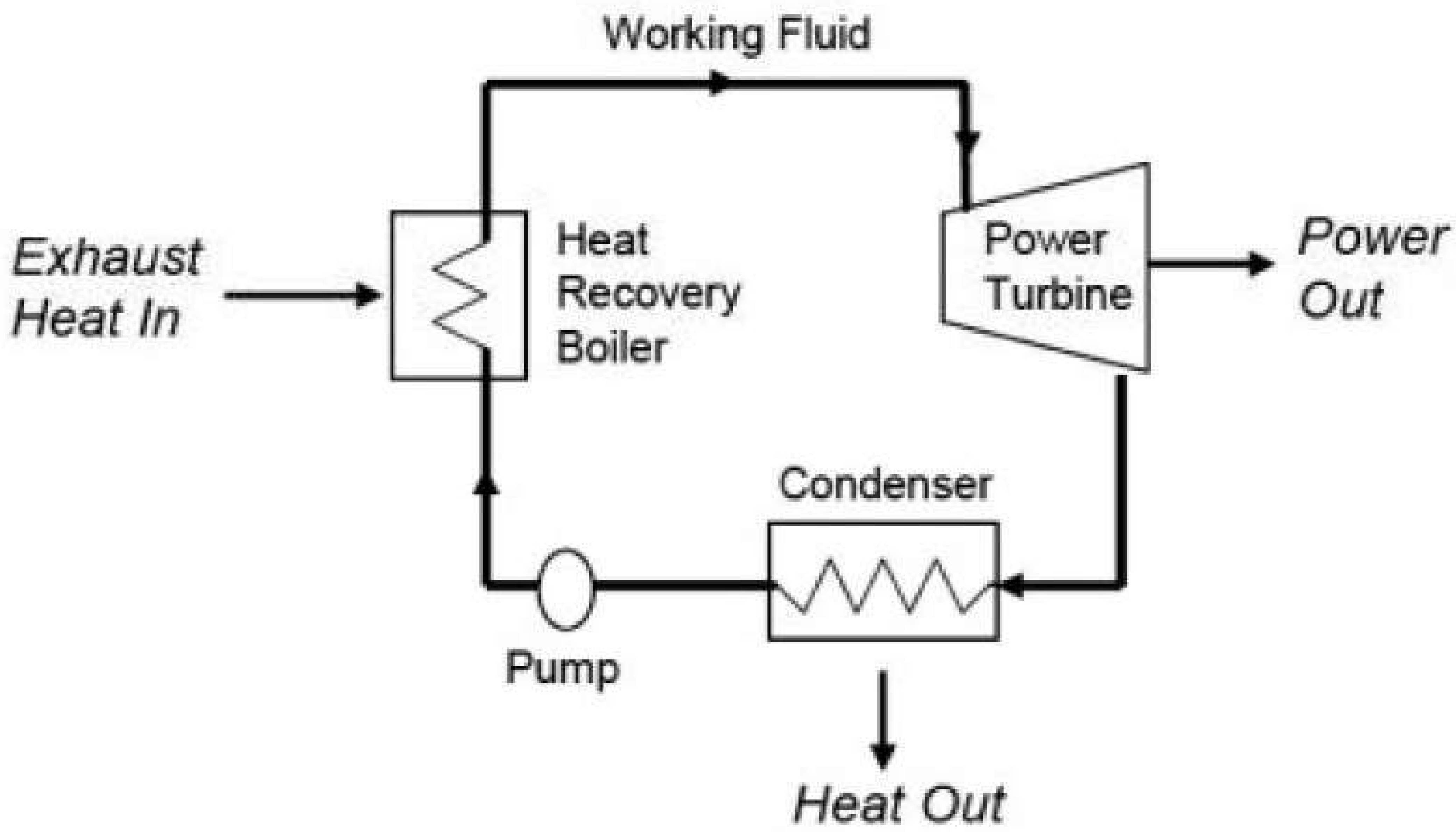


Figure 5.2 - Exhaust-heat recovery system.

expander.

The discharge from the primary expander is passed through a regenerator, in which the residual heat is exchanged with a stream of secondary liquid, which absorbs heat from the regenerator and flows through the second expander before returning to the condenser. This cascade system can contribute to the energy received in the single-stage expansion process.

The cascade system described above uses the same fluid for both stages of expansion, but another option is to have two different fluid circuits. The first absorbs heat from the exhaust pipe and passes through the regenerator before entering the condenser, and the second separate circuit, using a liquid with a lower boiling point, absorbs heat from the regenerator and expands through the second turbine expander. In both cases, the power of the expanders is used together to control one electric generator.

The current generation of large gas turbines with power from 25 to 35 megawatts (MW), which are used to drive gas compressors in pipelines, have an efficiency of almost or more than 40%. By adding a floor cycle, efficiency can be increased by 10% or more without increasing fuel consumption.

This means a significant increase in overall system efficiency. In addition, the generated electricity can displace a significant portion of the generation of electricity from fossil sources. This additional power has the additional advantage that it depends on the load, that is, the output power corresponds to the electricity needs in the gas line and, unlike other intermittent sources, such as wind or sun, is available for current peaks. Using this technique took some time to become practical and widespread. The main reasons for this are economic, not technical. When verifying the feasibility of each project, the following requirements must be taken into account: Additional capital costs per installation per kW worked out; This determines the payback period.

Access to the electricity grid to generate electricity, the distance needed to establish a connection, and the line capacity that corresponds to electricity.

Collaboration with an energy supplier who buys electricity, and fixed contractual agreements at the price payable for electricity.

Conclusion to chapter 4

1. There are a number of different harmful emissions from compressor stations ranging from toxic and harmful to the environment like gas emission to noise which is more harmful to human as a way of discomfort.

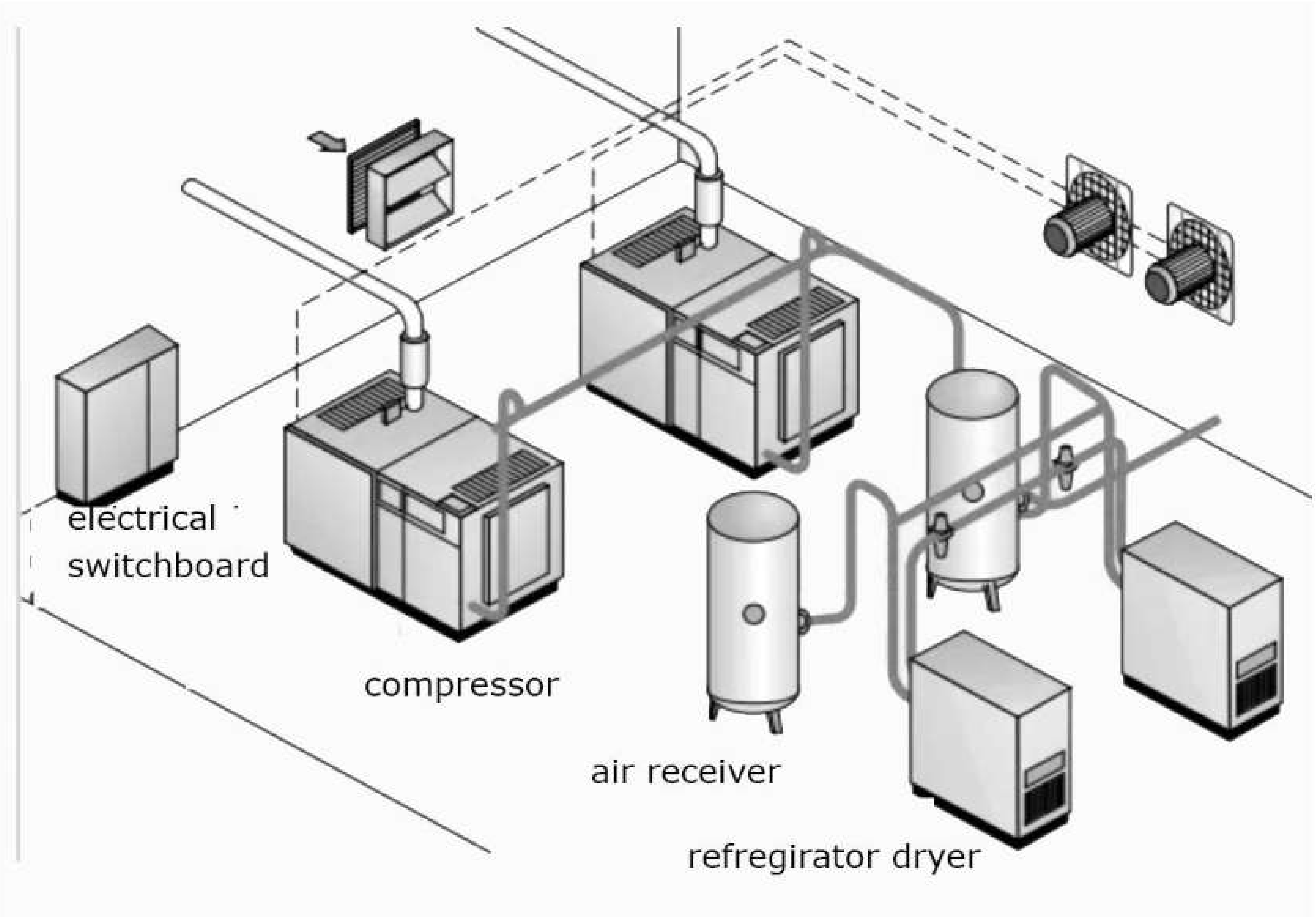
2. There are numerous ways to negate these emissions rather to be more environmentally friendly with use of filters for emoted gases or even those that can transform waste into useful energy, such as Rankine cycle.

CHAPTER 5. LABOR PRECAUTION.

5.1 Introduction

This compressor room was chosen due to direct correlation with the enterprise described in the diploma work. An employe will be present in the compressor room through the day for various task such as repairs, maintenance and diagnostics. The compressor room is a high risk area due to presence of the refrigerator unit which can malfunction and produce hazardous gas into the atmosphere. The electric distribution panel can be dangerous in events of faulty isolation when worked with or can catch fire because of power surges. Compressor emits high levels of noise and vibrations during operation which can be dangerous in case of prolonged exposure by human.

5.2 Analysis of the compressor stations engineers working place



The compressor room is the main work area of the engineer of compressor

Figure 5.1 - main compressor station room

station where they spend most of their working hours. It is a dangerous working place due to a number of factors. The employee can run into a number of dangerous cases with the appliances in the room e.g.

Electrical switch board - while usually secure it can cause lethal damage to humans in case of malfunction such as power spikes and sparking in case of faulty isolation of vires. In addition to deadly electric shock it can cause fires which is especially dangerous at the gas compressor station where everything is surrounded by highly flammable compressed natural gas.

Compressor - is a constant source of vibrations and noise which is harmful to humans with prolonged exposure. It can lead to health problems such as loss of hearing. Prolonged exposure to vibrations can lead to malfunction in correct operation of a number of human internal organs. And both of these can cause diminishing mental health of the working employee.

The guidelines of working procedures and technical operations at the workplace are described by ISO 13623:2017.
The required structure and workplace organizations should be done in Accordance with DSTU 7299:2013.

5.3 Industrial noise

Main gas pipelines are distinguished by an increased level of sound (noise), especially compressor shops, where the noise can reach up to 100 dBA.
The sound is characterized by height (frequency of oscillation) and intensity (sound pressure).
The frequency range of human acoustic perception of sound vibrations ranges from 16 to 20,000 Hz.
Sound pressure is a variable component of air and gas pressure resulting from sound vibrations, Pa.

Production noise distorts the reception of information, which affects errors and injuries. It causes fatigue. With prolonged exposure to noise decreases the acuity of hearing, changes blood pressure, weakens eyesight, changes in respiratory centers, which causes a change in coordination of movement, in addition, significantly increases the energy consumption under the same physical activity. Intense noise is the cause of cardiovascular disease, impaired gastric function and a number of other functional disorders of the human body. In noisy shops the most frequent cases of occupational injuries.

The impact of noise is reflected, first of all, on the hearing organs. There are three forms of exposure - hearing loss, noise trauma and professional hearing loss. The first is characterized by acute fatigue of ear cells and can cause the development of professional hearing loss. Noise injury can occur when exposed to high sound pressure, such as explosions, tests of powerful jet engines, and the like. At the same time, the victims experience dizziness, noise and pain in the ears, as well as lesions of the eardrum. Professional deafness leads to hearing loss until its complete loss.

Noise Limit (PDU) is a level of factors that should not cause illnesses or abnormalities in health which are found in daily (except weekends) work, but no more than 40 hours per week for the entire length of service research in the course of work or in the distant lifespan of present and future generations. Complying with the noise level of the noise does not exclude health disorders in hypersensitive persons.

The permissible noise level is a level that does not cause much concern to the person and significant changes in the performance of the systems and analyzers sensitive to noise.

5.3.1. Classification of noise affecting humans

The nature of the noise spectrum distinguish:

- Wideband noise with a continuous spectrum width of more than 1 octave;
- Tone noise, in the spectrum of which there are pronounced tones. The tonal character of the noise for practical purposes is set by measuring in 1/3 octave bands the frequency of exceeding the level in one band above the neighboring not less than 10 dB.

The temporal characteristics of noise distinguish:

- Constant noise, the sound level of which for 8-hour working day or during measurement in the premises of residential and public buildings, in the territory of a residential building changes in time no more than 5 dBA when measured on the temporal characteristic of the sound meter "slowly";
- Unstable noise, the level of which over an 8-hour working day, working shift or during measurements in residential and public buildings, in the territory of a residential building changes over time by more than 5 dBA when measured on the temporal characteristic of the sound meter "slowly".

Permanent noise is divided into:

- time-varying noise, the sound level of which changes continuously over time;

- intermittent noise, the sound level of which changes stepwise (by 5 dBA and more), and the duration of the intervals during which the level remains constant is 1 s, with the sound levels in dBA1 and dBA, measured according to the temporal characteristics of "impulse" and " slow ", not less than 7 dB.

5.3.2. Normalized parameters and maximum permissible noise levels in the workplace

The constant noise at work is characterized by sound pressure levels in dB in octave bands with geometric mean frequencies of 31.5; 63; 125; 250 Hours; 500 Hours; 1000; 2000; 4000; 8000 Hz, determined by the formula:

L = 20 lg P / P0, where P is the rms value of the sound pressure, PA; P0 is the initial value of the sound pressure in the air, which is 2 · 10-5 Pa.

It is permissible as a constant broadband noise in the workplace to receive the sound level in dBA, measured on the temporal characteristic of the "slow" sound level meter, which is determined by the formula:

LA = 20 lg PA / P0,

where PA is the rms value of the sound pressure, taking into account the correction "A" of the sound level meter, PA. The characteristic of non-continuous noise in the workplace is the equivalent (in energy) sound level in dBA. The maximum permissible sound levels and equivalent sound levels at workplaces, given the intensity and gravity of the work, are presented in Table. 5.1

Table 5.1 - Limit sound levels and equivalent sound levels at jobs for different categories of work

Category of labor process intensity	The severity of the work process				
	Easy exercise	Average exercise	Hard Work 1 degree	Hard work 2 degrees	Hard work 3 degrees
The tension is mild	80	80	75	75	75
Medium tensions	70	70	65	65	65
Hard work 1 degree	60	60	-	-	-
Hard work 2 degrees	50	50	-	-	-

5.4. Industrial vibration

Vibration - oscillatory movements of elastic bodies, structures, buildings near equilibrium.

The effect of vibration on humans is classified as:

• the method of transmission of vibration to humans;

• in the direction of vibration;

• by time of action.

According to the method of transmission to humans distinguish between general and local vibration.

General vibration is transmitted through the supporting surfaces to the body of a sitting or standing person;

The total vibration by its source is divided into categories:

1 (safety) - transport, affecting the operators of mobile self-propelled and towed cars - tractors, agricultural and industrial machines, cars, road construction machines;
2 (the limit of labor productivity decline) transport-technological, which affects the operators of machines with limited speed of movement - excavators, cranes, concrete pavers, floor production transport;
3 type "a" (boundary of decrease in productivity) - technological, affecting operators of stationary machines and equipment or transmitted to workplaces that have no sources of vibration - machine tools, forging and pressing equipment, pumping units, fans, boring machines, installations oil and gas, refining, etc. industries;
3 type "b" (comfort) - vibration in the workplaces of intellectual workers and non-physical laborers - dispatchers, factories, design bureaus, laboratories, computing centers, training rooms, office premises, health centers and others
Local vibration is transmitted through human hands. It can be attributed to the impact on the feet of the sitting person and the forearms in contact with vibrating surfaces.
In the direction of action vibration is divided according to the direction of the orthogonal coordinate system,
The time characteristic differs: constant vibration, for which the controlled parameter changes by no more than 2 times (by 6 dB); non-permanent vibration, for which these parameters change more than 2 times (by 6 dB) during the observation. At the action of vibration on a person the vibration speed (vibration acceleration), frequency range and time of influence of vibration are estimated.

Frequency response range of vibrations from 1 to 1000 Hz. Oscillations with a frequency below 20 Hz are perceived by the body only as vibration, and with a frequency above 20 Hz - simultaneously as vibration and sound.

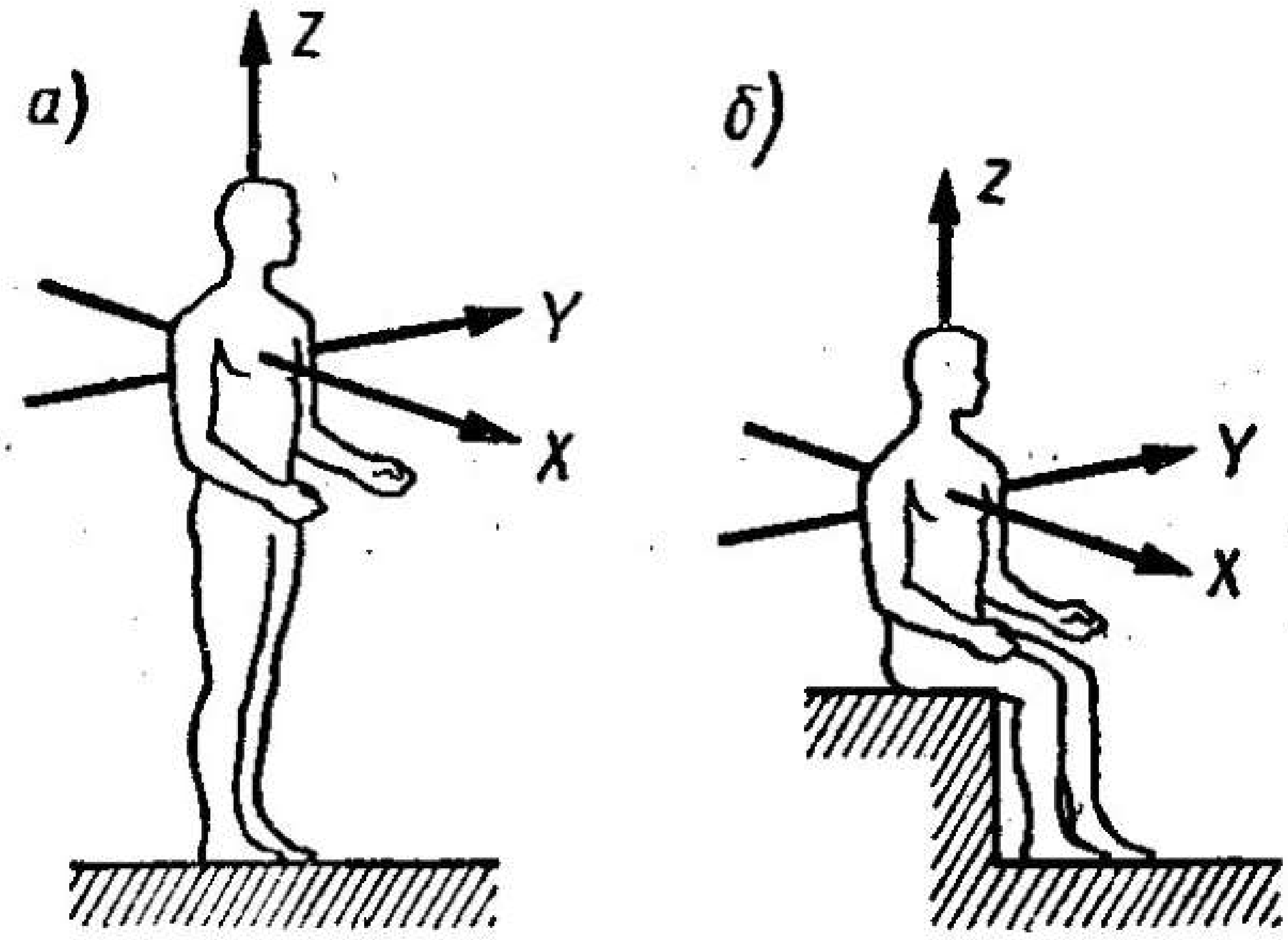


Figure 5.2 General vibration: a) standing, b) sitting

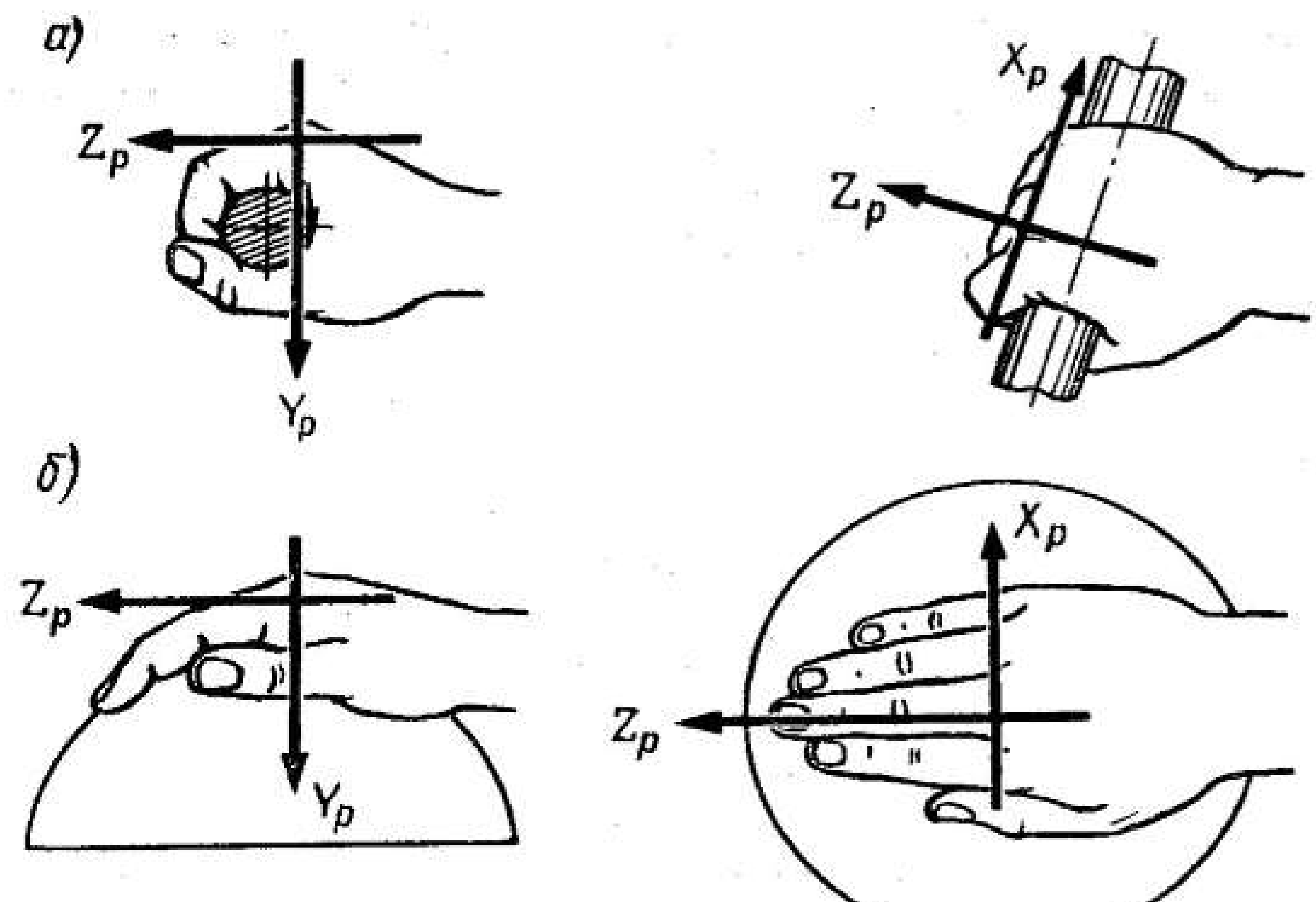


Figure 5.3- Direction of coordinate axes under the influence of general and local vibrations

Local vibration at the grip: a) end, b) spherical surfaces

Local vibration at the grip: a) end, b) spherical surfaces.

General vibration causes changes in the cardiovascular and central nervous system, the appearance of pain in individual organs. Local vibrations affect the central nervous system, increasing blood pressure, causing the capillaries to narrow in the tips of the fingers, losing their sensitivity. Under the influence of vibration, visual perception deteriorates, especially at frequencies of 25-40 and 60-90 Hz.

Vertical vibration is especially unfavorable for workers in a sitting position, horizontal - for workers standing.

The effect of vibration on a person becomes dangerous when the frequency of oscillation of the workplace approaches the frequency of its own oscillation of the organs of the human body: 4-6 Hz - oscillations of the head relative to the body, in a standing position; 20-30 Hz - in sitting position; 4-8 Hz - abdominal cavity; 6-9 Hz - most internal organs; 0.7 Hz - "duck", causing seasickness

5.5 Normalization of vibration

Normalized and controlled vibration parameters, according to DSTU 12.1.012: 2008, use the mean square values of vibration acceleration (a2, m / s2) or vibration velocity (V2, m / s), as well as their logarithmic levels in decibels (dB).
The logarithm of the level of vibration velocity (LV, dB) and vibration acceleration (La, dB) is determined by the formulas:
Where 5 · 10-8 10-6 are the reference values of vibration velocity and acceleration.
The normalized frequency range is set to:
for local vibration in octave bands with geometric mean frequencies (f2 / f1 = 2) - 1, 2, 4, 8, 16, 31.5, 63, 125, 250, 500, 1000 Hz;

for general vibration, vibration in octave and 1/3 octave bands with geometric frequencies (f2 / f1 =) - 0.8, 1, 1.25, 1.6, 2.0, 2.5, 3.1, 4.0, 5.0, 6.3, 8.0, 10.0, 12.5, 16.0 , 20, 25, 31.5, 40, 50, 63, 80 Hz.

Vibration can also be normalized by the one-digit value of the controlled parameter (frequency-corrected)(Table 5.2).

$$\tilde{V} = \sqrt{\sum_{i=1}^{n} (V_i, K_i)^2},$$

$$L_{\tilde{V}} = 10\lg\Sigma 10^{0.1\cdot\left(L_{V_i} + L_{K_i}\right)},$$

where Vi and LVi are the rms values of the controlled vibration parameter or its logarithmic level in the i-th frequency band; n is the number of octave bands; Ki and LKi are the axial coefficients for the i-th frequency band (Table 5.3).

Table 5.3 - Single-digit indicators of vibration load on the operator for an eight-hour day

Kind of vibration	Vibration category	Directed action	Normative, frequency-corrected values			
			vibration acceleration		vibration speed	
			$m/s \cdot 10^{-2}$	dB	$m/s \cdot 10^{-2}$	dB
Locally	-	$X_Л$, $Y_Л$, $Z_Л$	2,0	126	2,0	112
Overall	1	Z_0	0,56	115	1,1	107
		Y_0, X_0	0,4	112	3,2	116
	2	Z_0, Y_0, X_0	0,28	109	0,56	101
	3 type «a»	Z_0, Y_0, X_0	0,1	100	0,2	92
	3 type «в»	Z_0, Y_0, X_0	0,014	83	0,028	75

Table 5.4 - Vibration correction weights for different types and directions of vibration in an octave

Average geometric frequencies of the bands, Hz		1	2	4	8	16	31,5	63
Z	K_i	0,045	0,016	0,45	0,9	1,0	1,0	1,0
	L_{Ki}	-25	-16	-7	-1	0	0	0
X, Y	K_i	0,5	0,9	1	1	1	1	1
	L_{Ki}	-6	-1	0	0	0	0	0

The vibration dose (D) is determined by the formula:

$$D = \int_0^T \widetilde{V}^m(t)dt,$$

where (t) is the frequency-adjusted value of the controlled parameter at time t, ms-2 or ms-1; T - vibration exposure time, s; m is an indicator of the equivalence of the physiological effect of vibration established by sanitary standards.

The equivalent corrected value (Vekv) is determined by the formula:

$$V_{экв} = \sqrt[m]{\frac{D}{T}}.$$

5.6 Fire and gas work. Conducting them in the conditions of the compressor station

The existing compressor station refers to objects of high explosion and fire hazard. For this reason, the production of works involving the use of open flames, as well as works related to the inspection, cleaning and repair of equipment, during which there is or may not be excluded the possibility of explosion - and flammable vapors and gases into the work area, careful attention must be paid.

To fire include work with the use of open fire and sparking, heating equipment, tools to the ignition temperature of the gas-air environment. Fireworks in existing compressor shops are considered welding works, gas cutting and related operations carried out in explosive premises of the shop or directly on existing gas communications, as well as on communications within the shut-off valves of the CS (Nos. 7 and 8).

Fireworks are performed according to the plan of their carrying out and with registration of the outfit-admission to their performance. The admission permit specifies the personnel involved in the work, responsible for the production of works, assigned to the order, measures for the preparation and safe conduct of fire work, fire prevention measures.

The plan of carrying out fire works with the addition of the necessary circuits reflects the issues of arrangement of used mechanisms and machines, schemes of technological pipelines and provisions of shut-off valves for the period of these works, the direct order of carrying out of works and the procedure of discharge and supply of gas, safety measures. Particular attention in the production of fire works is given to measures that prevent the spontaneous rearrangement of cranes, and work on technological pipelines using gas cutting and electric welding. To do this, switch off the impulse gas coming to the crane, remove the handles and controls of the manual control of the cranes, post the prohibiting and warning posters. On especially responsible sites exhibit observation posts from service personnel.

Fire (welding) work in the premises of the gallery of the superchargers should be carried out with a complete stop of the shop with gas poisoning from all communications, for which a special act must be drawn up. In all cases, the execution of fireworks in the gallery of blowers should be carried out according to a special plan, with obligatory observance of all safety measures and rules of fire safety. During the opening of the superchargers it is forbidden to carry out fireworks in the gallery of the superchargers and on the repaired unit in the engine room. Fireworks in the gallery of the superchargers are made under the personal guidance of the Chief Engineer of the enterprise (UMG).

Gas hazardous is considered to be work performed in a contaminated environment or in which gas can escape from gas pipelines, their communications and apparatus, gas equipment, shut-off valves or units. Gas works include:

a) connection of newly installed gas pipeline communications, apparatus (dust collectors, filters, gas heaters, etc.) to existing communications located indoors and out;

b) commissioning of gas communications;

c) revision, repair and replacement of gas communications, underground and above-ground gas pipelines under pressure;

d) opening of centrifugal superchargers (performed by special instruction);

e) inspection and audit of protective grilles on suction nozzles of centrifugal superchargers;

e) replacement of the centrifugal pump bearing seal;

g) pouring into the technological communications of the reagents to eliminate the hydrates of the formations;

h) start-up of the gas-pumping unit (performed by special instruction);

i) inspection and ventilation of wells with shut-off valves;

j) drainage of condensate from dust collectors and possible places of accumulation of it in technological strapping of equipment;

l) preventive maintenance of gas-operated appliances and equipment.

Each CU develops a list of hazardous works that are performed with the outfit-clearance and without the outfit-approval, but with the mandatory registration of such work before they begin. The procedure for designing gas works is similar to the design of fireworks.

Before the start of fire and in the course of work periodically measured air pollution, the presence and serviceability of personal protective equipment.

Special precautions are taken when carrying out work inside pressure vessels. In these cases, the vessels to be opened for inspection, cleaning and preparation for repair and maintenance must be disconnected from the pipelines and free from gas. Vessels may only be opened under the supervision of the person in charge of the work. If there are several hatches, they should be opened consecutively, starting from the top. Immediately before the descent (lifting) of the worker into the vessel, the person responsible for carrying out the work must check the health of the workers (by means of a survey), the presence of appropriate overalls, PPE, rescue equipment and other equipment listed in the work permit. To go down into vessels and apparatus and to begin work in them is allowed only in the presence of the person responsible for carrying out these works.

To protect the respiratory system of persons working inside the vessels, only hose gas masks should be used. Hose gas mask with carefully adjusted helmet-mask and adjustable fresh air supply the worker puts on just before descending into the vessel. The tightness of the gas mask fitting and the air blower is checked by the person responsible for carrying out these works. The intake hose fitting of the gas mask hose must be removed into the zone of clean air from the leeward side and secured. The hose should be placed in such a way as to preclude cessation of air through the kinks, distortion and the like. When working in a hose gas mask, the period of one-time stay of people in the vessel should not exceed 20-30 minutes. Rest in the air must be at least 15 minutes.

The worker is obliged to wear a safety belt on top of the overalls with a cross-strap and a durable signal-rescue rope attached to them. The free end of it (at least 10 m in length) should be brought out and handed over to the observer. All tools and materials required for the work should be placed in a bag or other container after the worker has lowered himself into the receptacle.

Work inside the vessel should be carried out during the day. It allows only one person to work. If the working conditions require two (or more) people to be in the vessel at the same time, additional safety measures should be developed and listed in the permit.

Work inside the tank should be carried out by a team of three or more people: one - the contractor, two - watching. When carrying out the work, they are obliged to stay close to the tank, to constantly monitor their work and to provide it with uninterrupted clean air. Observers should be dressed in the same way as the one working. If necessary, they should assist him. If any malfunctions (hose puncture , blower stop, breakage of the rescue rope, etc.) are found, the work should be stopped immediately and the worker should be removed from the tank.

When working inside the tank, it is allowed to use only lamps with a voltage of no more than 12 V in explosion-proof version. They should be switched on and off without capacity

5.7 Requirements for work in the gallery of blowers with disassembly of the blower

Execution of the supercharger opening is gas-hazardous and is made only in the presence of the issued outfit-tolerance. Before starting work it is necessary to:

- check the closing of cranes № 1, 2, 4, 4 bis, fill them with sealing oil;

- check the opening of cranes № 3, 3 bis and 5;

- posters "Do not open" and open "Do not close" posters;

- to prevent involuntary rearrangement of cranes it is necessary to remove hoses (RVD) or tubes (depending on the design of the crane) of the supply of impulse gas to the hydraulic pneumatic cylinders , to close the valves of gas selection on the impulse manifold, to disconnect the supply lines of the impulse gas to the cranes; on the tap # 4 to install a bracket (block-lock) which excludes opening it manually; The control panels of the unit are de-energized and posters "Do not switch on, people work" are posted on them.

At the inlet and outlet of gas between the supercharger and the sunroof, install rubber inflatable balls, leaving the sunroof open. Filling of rubber balls is carried out by air to a pressure not higher than 500 mm of water . with its control by a U-shaped manometer.

All personnel involved in the job of opening the supercharger must be instructed before work begins on the procedure for carrying out the work and the rules of safety with the instruction in a special journal.

After organizational and technical measures have been taken to ensure the safe operation of the supercharger opening work, the person responsible for carrying out this work and the alternate engineer-dispatcher shall allow the personnel of the repair team to get started .

During the period of opening and repair of the supercharger, the supply and exhaust ventilation must be in constant operation. In addition to the operation of automatic gas analyzers in the pump gallery systematically, but not less than 30 minutes, it is necessary to perform an air analysis at the place of work with the entry in a special journal. If the gas content is more than 1%, the work is stopped and measures are taken to prevent gas penetration. During the repair work on the supercharger systematic control over the condition of inflatable rubber balls is established.

The opening and assembly of the supercharger must be carried out by the same personnel. During the opening of the supercharger, only persons who perform the work and shift personnel servicing the working units may be in the room.

During short breaks (up to one hour) in the work of repairing the supercharger, the monitoring of the exposed supercharger is performed by shift personnel. If it is impossible to finish the work on the open supercharger in one change of interchangeable break in work are not allowed.

If, under the conditions of repair, the supercharger remains without the rotor for a long time, it is necessary to seal its gas cavity by installing a power plug instead of the sealing bearing and end cap. The plug must be rated for operating pressure.

At the time of opening and repair of the supercharger, it is forbidden to perform any other work in the gallery of superchargers, which are not related to this opening and repair.

After repair, the supercharger can be closed only after a thorough check for its absence, as well as in the suction and discharge pipelines of foreign objects.

It is prohibited in the compressor shop:

- laying temporary electrical networks;

- use machine housings, pipelines and metal structures of buildings as grounded welding machines and welding products;

- to dry overalls on central heating devices, hot surfaces of units and gas communications;

- clutter passageways and exits from the premises, as well as approaches to fire extinguishers and external fixed steps;

- work in explosive rooms in shoes with steel horseshoes and on nails;

- use open fire for heating of pipelines, shut-off devices and other equipment;

- carry out welding work in violation of current rules and instructions;

- carry out any work related to the replacement and repair of valves in oil pipelines and disassembly of control parts (except for replacement of pressure gauges) with the unit running.

In case of fire production personnel are obliged to:

- Immediately block the access of gas or oil to the place of fire;

- call a fire brigade or a volunteer fire brigade;

- to take measures to extinguish the fire with available fire extinguishing means;

- to inform the management of the compressor shop and the UMG;

- disable inflow and exhaust ventilation.

For rapid elimination of emergency and clear interaction, it is necessary that all personnel know their specific duties and actions in case of fire. For this purpose it is necessary to conduct regular training sessions on fire elimination, an approximate list of fires which can be discussed in the instructions for the elimination of fires in shops, buildings and other premises of the station.

5.8 GPU fire extinguishing system and its operation

A number of structures of the compressor station belong to the highest category A. According to the degree of fire hazard, among these structures, first of all - the compressor shop, which is the main source of fire safety at the COP. This is due to the fact that in the event of an oil-lubrication system and sealing, oil may get into the hot parts of the GTU, which will inevitably lead to an outbreak. In addition, explosive mixtures of combustible gases can occur during an accident, which can also be a source of explosion or explosion.

To prevent fires and extinguish fires, compressor shops have fire extinguishing systems performed in accordance with DSTU ISO 23932:2017.

The system includes:

- automatic all -fire extinguishing system;
- fire water supply system with fire pumps, ring, collector with hydrants and sleeves;
- portable individual fire extinguishers.

The automatic general fire extinguishing system is designed to signal the appearance of flames, smoke in the engine room of the compressor shop or fire at the GPU, emergency stop of the GPU and eliminate the ignition by automatically supplying the fire extinguishing agent to the combustion zone.

The following automatic fire extinguishing systems are used at the CU:

- foam fire extinguishing system used on stationary gas-pumping units of types: GTK-5, GT-750-6, GTK-10, GTN-6, GT-6-750 and GTN-25-1;

- gas fire extinguishing system used on units of block execution, types GPU-C-6,3, GPU-C-16, GTK-25I, " Solar ", EGPU-25;

- powder fire extinguishing system, additionally installed on GPU-Ts-6,3, " Solar ".

Schematic diagram of the foam fire extinguishing system is presented in (Fig. 4.1.) This system includes:

water tanks E1 (primary and backup), each with a capacity of 50 m, pumping stations H1, consisting of two centrifugal pumps type 4K-6 (one reserve pump) electric two Foam - mixer, providing a wipe and 1.44 liters of foam / s, pi but -generator sets a type of PRT-600 and PRT-200 that provide, if necessary, to supply foam GPU in the amount of, respectively, 600 and 200 1 / s, fire alarm sensors such STS-038, providing a supply signal fire for 7 s (at 20 ° s / s temperature rise and you are still), pipeline systems with valves and valves designed for water and foam to foam - generator.

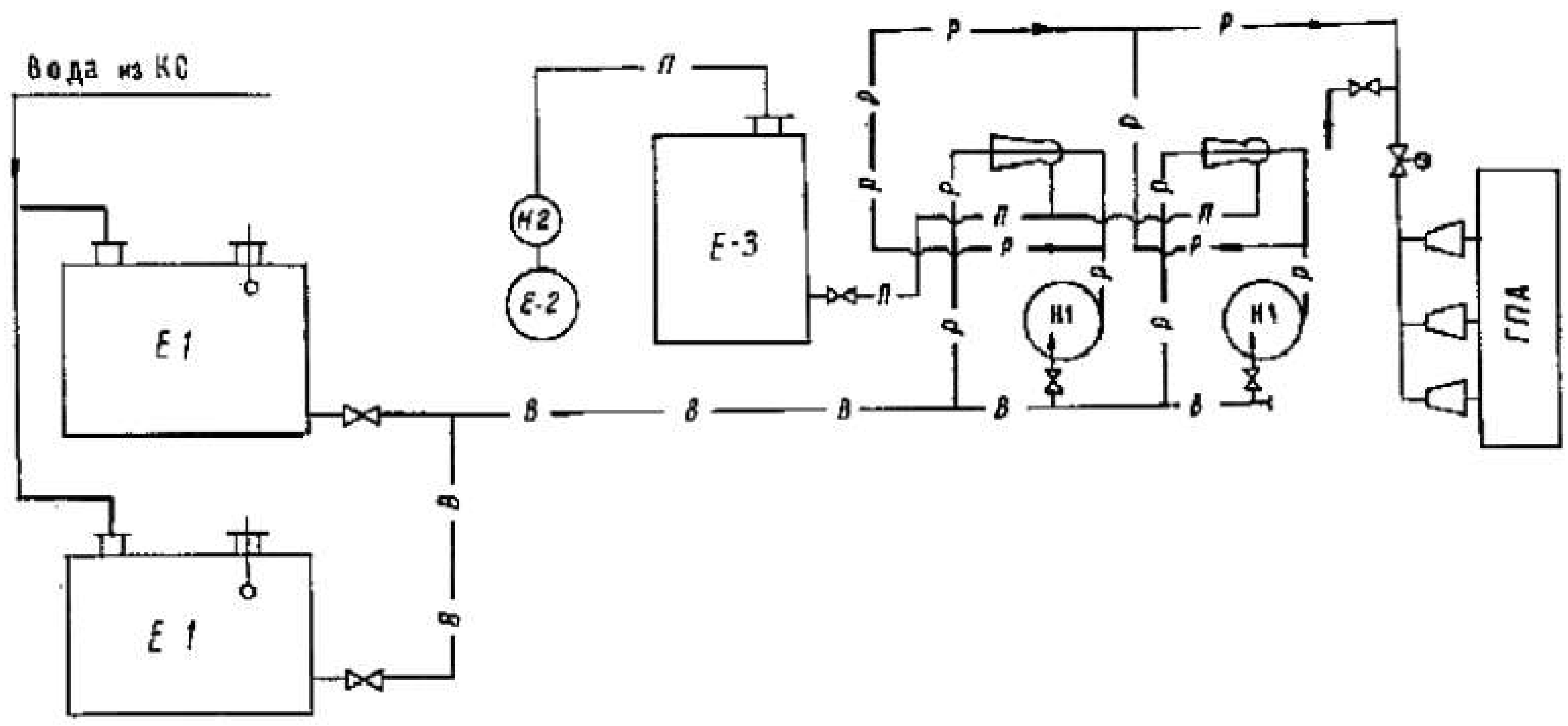


Figure 5.4 - Schematic diagram of the foam fire extinguishing system of the compressor shop

----- p ---- - 4% solution of foaming agent; ----- n ------ - foaming agent; ----- in ---- - water; - - latch; foam generator; - latch with electric drive

Foam - generators supply PRTs provide a stable, compact spray foam especially in these most likely focus fire on the turbine, the front, middle and rear bearings, the oil tank and combustion chamber GPU.

When the temperature above the GPU is higher than 150 ° C with the rate of its growth not lower than 20 ° C / s, the sensor is triggered, the fire alarm system circuit and the fire extinguishing system are closed; sound and light signals are fed to the main shield of the compressor shop. At the same time the water fire pump is put into operation, the shutters for supplying the solution to the foam generator and into the combustion zone are opened. If within 10 minutes fire will not be eliminated, remote start again manually by pressing a button.

The scheme of fire automatics provides for the possibility of both automatic and remote activation of the fire extinguishing system from the automatic fire extinguisher board installed in the control room.

At each compressor station, a periodic inspection of the operation of the entire fire extinguishing system and its individual components must be carried out within a specified time frame.

The principle of operation of gas and powder fire extinguishing systems is similar to the operation of the foam fire extinguishing system except that the extinguishing agent is in pressurized cylinders.

In addition to aggregate automatic fire extinguishing systems, a conventional fire-fighting water supply system is envisaged at the COP: primary fire extinguishers are widely used, in particular, portable and mobile powder and carbon dioxide extinguishers.

The choice of type and calculation of the required number of portable fire extinguishers is made depending on their extinguishing ability, limit area and the class of fire of combustible substances at the COP. In accordance with the standards of equipment for premises with manual fire extinguishers, the compressor shops are equipped with the following types of fire extinguishers:

- portable
- powder OP-5, OP-10;
- carbon dioxide OU-5, OU-8;
- air-foam OVP-10;

move

- powder OP-50, OP-100;
- Carbon dioxide OU-80, OU-400;

Of the portable powder extinguishers, the OP-10 type of fire extinguishers were most widely used in compressor shops. This type of fire extinguisher is designed for extinguishing fire spilled flammable and combustible liquids, petroleum products and electrical installations under voltage up to 1000 V. Fire used at ambient temperatures from -50 to +50 ° C. Construction of OP-10 is illustrated below.

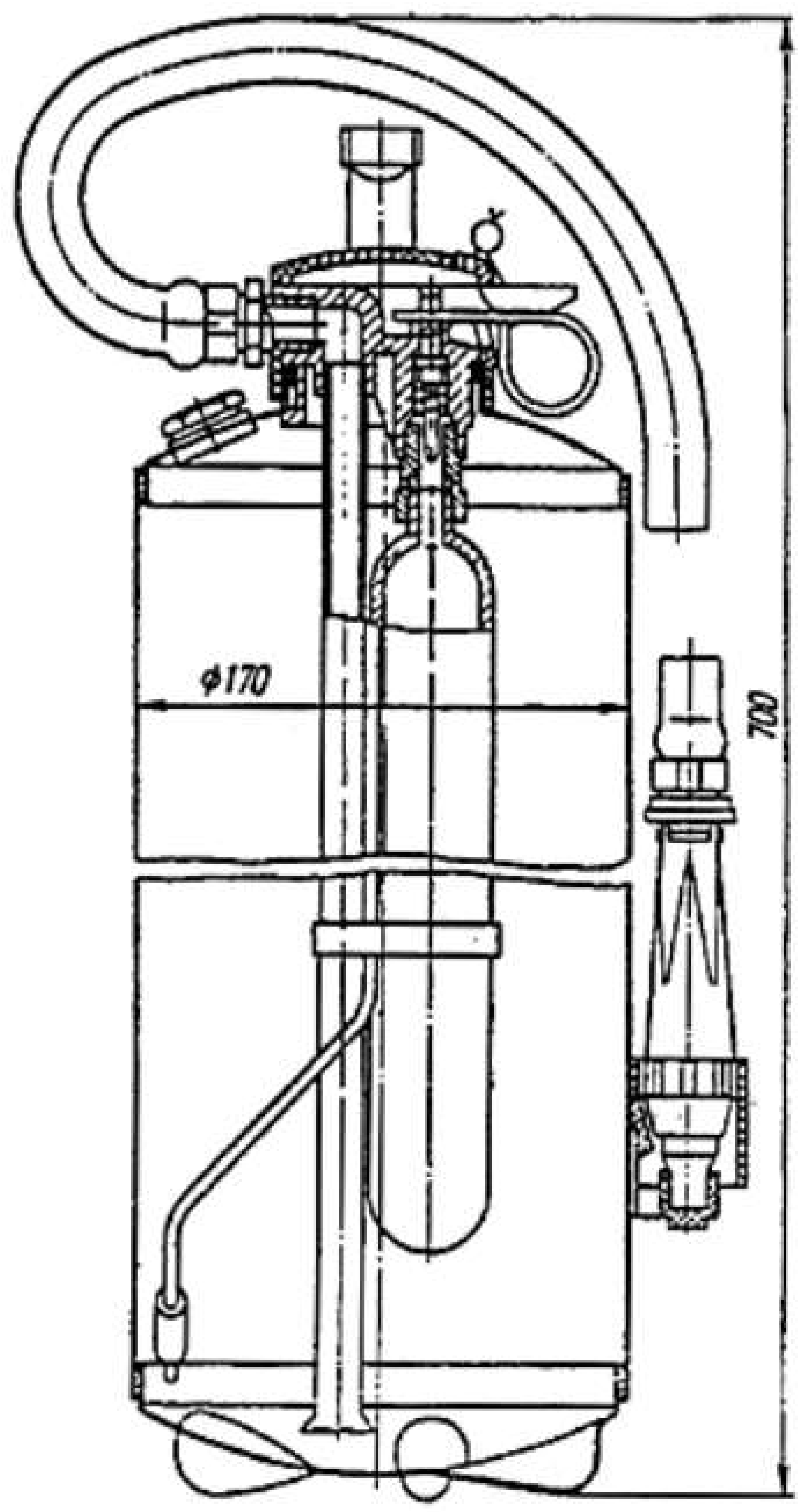


Figure. 5.5 - Fire extinguisher powder OP-10, model 01

The fire extinguisher consists of a steel housing, a cylinder for storage of working gas, through which the supply of powder from the housing of the fire extinguisher, the cover with the shut-off device, siphon tubes and tubes for supplying the working gas to the lower part of the housing of the extinguisher, a hose that pops up the barrel-nozzle formation and direction of the jet of powder to the combustion cell.

The most popular of the fire extinguishers are OP-100 powder extinguishers. The schematic diagram of the work of the fire extinguisher is similar to the scheme OP-10, design differences are only in the geometric dimensions of the body of the fire extinguisher. OP-100 mobile fire extinguishers have chassis with pneumatic tires. As re extinguishing agents, powders applied Pi welt A P-1A PF.

During operation of compressor shop staff responsible for fire safety, regularly monitors the impermeability of the junction of pipelines good condition of Oil economy, serviceability accidental discharges of oil tank, serviceability of ventilation systems, tightness fire wall between room and engine room pumps and galleries etc.

The outer surface of the gas turbines and gas ducts must have good thermal insulation and be enclosed with a decorative casing. The outside insulation temperature should not exceed 45 ° C.

Heating of apparatus, communications and shut-off devices during the autumn-winter period of operation is allowed to be carried out only by steam, hot water or hot air.

In the event of a fire in the compressor shop, the CU staff is obliged to call the fire brigade and at the same time start its extinguishing on its own using fire extinguishers, carbon dioxide installations, foam fire extinguishing systems, etc. At the same time it is necessary to block the access of gas or oil to the place of fire, stop the working unit, disconnect the power supply to the unit, switch on the fire extinguishing system. At the same time it is necessary to pay attention that the fire did not spread to the room of the superchargers and to the roof of the building.

When repairing water supply systems at the substation, it is necessary for the purpose of fire safety to have the necessary supply of water in accordance with current regulations.

In case of gas ignition at station communications and inability of rapid elimination of ignition, the entire compressor shop with gas poisoning must be stopped from all technological connection of the compressor station.

Conclusion to chapter 5

The data on the effect of production noise on the people working on the objects of the gas transmission system, the data on the effect of production vibration on the people working on the compressor station are presented and described. Ways to protect against noise and vibration.

General Conclusions

1. The Ukrainian Gas Transport System is able to transport gas volumes of 120 bcm per annum. On average, 80% of Russian gas exports to Europe pass via the Ukrainian system. This equates to around 20% of the EU's total gas demand. Ukraine uses the same pipeline system to transport gas for domestic use. These facts demonstrate that the impeccable functionality of the Ukrainian system is in the common interest of Naftogaz Ukraine, Gazprom, as well as Ukrainian and European consumers.
2. Using these methods, major of these problems would be resolved in the recent decade, which allow us to dictate market conditions, and use it for bigger profit, which can be spent of further modernization.
3. method for calculating NSIMGT in LS MTP was proposed, the mathematical model of which includes mathematical models of pipeline sections and stop valves, which differs from existing calculation methods by the method of including and excluding equations of models of all MTP elements in the general system of equations of the general mathematical model and allowing high accurately calculate the real processes of gas flow in MTP in real time, as well as predict various situations in order to adopt the necessary control action. A system of differential equations in partial derivatives describing NSIMGT was taken as a mathematical model for PT.

LIST OF ABBREVIATIONS

GTU	-	Gas turbine unit
HPC	-	High pressure compessor
HPT	-	High pressure turbine
LPC	-	Low pressure compressor
LPT	-	Low pressure turbine
SAM	-	Sound absorbing materials
MTP	-	Multithread pipeline
OP	-	Operational parameters
GTP	-	Gas Turbine plant,
MM	-	Mathematical model
OP	-	Operational parameters
MTP	-	Multithread pipeline
LS	-	linear sections

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