

MINISTRY OF EDUCATION AND SCIENCE OF UKRAINE
National Aviation University

**NATURAL PRESSURED GAS
VEHICLES FILLING STATIONS**

Course Project Method Guide for students of the specialty
code 142 «Power Machinery» of the specialization
«Gas Turbine Plants and Compressor Stations»

Kyiv - 2017

УДК 656.065.3:621.45.037(076.5)

ББК 0330-082.057-4p

N 36

Compilers:

K.I. Kapitanchuk - PhD, prof. - chapter 1, 3;

M.Y. Bogdanov - PhD, associate prof. - chapter 2;

N.G. Denisenko - associate prof. - chapter 3, 4.

The English Language Adviser *N.G. Denisenko*

Reviewer:

Approved by Methodical and Editorial Board of National Aviation University (Minutes № 1/17 of 16.03/2017).

Представлено загальні методичні рекомендації, зміст і порядок виконання курсового проекту, обсяг його графічної частини, а також список літератури.

Для студентів спеціальності 142 «Енергетичне машинобудування» спеціалізації «Газотурбінні установки і компресорні станції».

Natural pressured gas vehicles filling stations: guidelines for N 36 performing a course project / compilers: K.I. Kapitanchuk, M.Y. Bogdanov, N.G. Denisenko. – K: NAU, 2017. -30 p.

The guide includes general guidelines, content and procedure for performing a course project, amount of its graphical part and list of literature.

For students of the specialty code 142 «Power Machinery» of the specialization «Gas Turbine Plants and Compressor Stations».

CONTENT

INTRODUCTION	4
1. GENERAL GUIDELINES	5
1.1. Purpose of course project	5
1.2. Scope of course project	6
1.3. Contents of calculating and explanatory notes	6
1.4. Contents and scope of graphical part	7
2. PROCEDURE OF PERFORMING THE COURSE	
PPOJECT	7
2.1. Definition of physical and chemical properties of natural gas	8
2.2. Thermodynamic calculation of piston compressor	13
2.2.1. Selection of compressor stage numbers	13
2.2.2. Determination of suction and pumping pressures for separate stages	14
2.2.3. Determination of gas parameters by compressor stages	17
2.2.4. Determination of volumetric coefficients by stages	19
2.2.5. Determination of the gas suction amount by compressor stages	20
2.2.6. Choosing of a type and a schemes of a compressor	21
2.2.7. Choosing of piston stroke and rotation frequency	22
2.2.8. Determination of stages of cylinders diameters	23
2.2.9. Determination of compressor power consumption and selection of an engine	24
3. CALCULATION OF COMPRESSOR AND BATTERY	
PACK OPERATION	25
4. SCIENTIFIC AND RESEARCH PART OF COURSE	
PROJECT	27
LIST OF LITERATURE	28
Appendix 1	29
Appendix 2	30

INTRODUCTION

Nowadays, many countries worldwide focus on searching and developing alternatives for vehicles operating on petroleum motor fuels. At the same time, preference is given to natural gas in countries that have lack of petroleum motor fuels and both in countries with sufficient oil resources.

Natural gas by its properties (heat of combustion, utilization efficiency, environmental friendly of combustion products) exceeds traditional petroleum motor fuel. A network of natural pressured gas vicles filling stations (NPGVFS) created in Ukraine, gives the opportunity of using vehicles that are fueled by compressed natural gas, not only for urban transportations but also for intercity connections.

Solving the problem of using natural gas in Ukraine as motor fuel has a particular importance, because this gives an opportunity to substantially reduce dependence of economics on oil import and petrochemicals and also improves the environmental situation in industrial centers of our country.

A network of NPGVFS created in Ukraine allows using vehicles that are fueled by compressed natural gas, not only for urban transportations but also for intercity connections. Consequently - the problem of NPGVFS creation on the base of high pressure piston compressor that has a capacity from 50 and more fillings per day and could be installed at the plants with a large amount of vicles is very urgent.

Ukraine has all necessary conditions for rapid solution of this problem, that means:

- own resources of natural gas;
- more than 10 thousand kilometers of main gas pipelines of great diameter for providing natural gas safety transportation and more than 30 thousand kilometers of distributions pipelines, that cover all regions of Ukraine;
- network of NPGVFS provides transportations of automobiles operating on pressured natural gas to any parts of a country;
- nonstop construction of metal high pressured tanks and design of NPGVFS at the plants of Ukraine;
- scientific and technical researches that refer to the problem of gas consumption.

1. GENERAL GUIDELINES

1.1. Purpose of course project

Literate operation of modern compressor units for natural gas compression, its dewatering and filling of the automobile high pressured gas ballons requires highly skilled specialists.

Performing of the course project of «Natural pressured gas vehicles filling stations» is one of the methods of consolidating knowledge and skills, that student learns during the education process of discipline like «Technical thermodynamics», «Piston engines and compressors» and «Diagnostic of GTU and compressors».

Performing of course project is an important step in preparing for carrying out the graduation design of future specialists in energetics.

The specific purpose of course project is implied in the development of main technical and physical principles of operation of natural gas vicles filling stations and their main technological blocks according to guidelines of the course projects.

For successfully performing the course project a student should know the basic principles of using the compressed natural gas as the petroleum motor fuel for the internal combustion engines, the principle technological schemes of NPGV filling stations, the main construction principles and the physical basement of main and auxiliary equipment operation of modern NPGV filling stations, the technological schedules of NPGV filling stations.

A student should know also the process of high pressured vehicle tanks filled by compressed natural gas, main principle technological schemes of NPGV filling stations, basic direction of development of technological process and equipment of NPGV filling stations with the purpose of increasing their economic and operation efficiency, main rules of safe exploitation and technical maintenance of NPGV filling stations.

1.2. Scope of course project

Course project consists of the following parts:

- definition of physicochemical properties of natural gas;
- thermogasdynamic calculation of piston compressor;
- calculation of compressor operation;
- scientific and research part of course project;
- graphical part of course project.

1.3. Contents of calculating and explanatory notes

Calculating and explanatory note that contains 30...35 pages is carried out at one page side of A4 (210×297 mm) format. Explanatory note should include the order of calculation, calculation equations, necessary explanation and substantiation of adopted values and coefficients.

Explanatory note consists of the following parts:

- calculation of physical-chemical and thermodynamical properties of natural gas predetermined composition;
- themogasdynamical calculation of piston compressor;
- calculations of compressor structural parameters;
- calculations of compressor operation modes and battery tanks;
- scientific and research part.

NPGV filling stations design begins from the definition of the required number of refills per day. This value is defined by the task variant of the course project. Therefore, further calculation depending on the efficiency necessary filling the specified number of cars, thermodynamic calculation of piston compressor is carried out, which results in defining the basic parameters and geometric characteristics of the piston group. If it is necessary, the value of the design variables further recalculation is changed.

The next step of NPGV filling stations design is a calculation of compressor and battery tanks operation, as a consequence, we can define the sizes of batteries and the numbers of compressors that are necessary for safety station operation.

1.4. Contents and scope of graphical part

The scope of the graphical part of the project should be three sheets of A1 format.

Sheets should include:

- on the first half of the sheet A1 format there should be a technological scheme of NPGV filling stations with specification according to the number of refills per day;
- on the second half of the first sheet A1 format there should be a kinematic scheme of compressor according to calculation of the number of steps of injection with the specification of major components and structural elements;

– on the second sheet of A1 format there should be drawings of piston compressor with a crosshead at 1:1; 1:2 or 1:2,5 scale calculated by the design variables and specification of the main construct elements;

– on the third sheet of A1 format there should be the graphic scientific and research part of the project as the sort of dependencies and conceptual drawings, namely:

– charts, diagrams, structural designs, structural designs solutions that are represented in a separate part of calculating and explanatory notes the scope of 4...6 pages;

– design solutions from the resulting elaboration of scientific and research themes of the project they can be used in the technological scheme of NPGVFS.

When ordering an explanatory note from the scientific and research part one should bring specific and complete references on the used sources (author, source name, publisher, year of publication, magazine number, etc.).

During a course project defense, a student must show good knowledge of a subject in this field.

2. PROCEDURE OF PERFORMING THE COURSE PPROJECT

For the previous analysis of the initial data and for choosing a prototype of NPGV filling station it is recommended to view a technical characteristics of a typical NPGV filling stations that are given in the table 1.

Initial data are taken according to the task and the appendix 1 and 2:

1. Composition of natural gas;
2. The number of refills per day;
3. Pressure of natural gas at the inlet of NPGV filling station.

2.1. Definition of physical and chemical properties of natural gas

Natural gas that is used as a motor fuel is a mixture of hydrocarbon gases. The main component of natural gas is methane. In addition to methane gas composition includes ethane, propane, butane, pentane, carbon dioxide and nitrogen. The composition of natural gas (in percentage by volume) is taken in accordance with Appendix 1 according to the number of individual tasks for performing a course project.

Table 1

Main technological indexes of a typical NPGVFS

Indexes	NPGVFS-50	NPGVFS-75	NPGVFS-125	NPGVFS-250	NPGVFS-500
Productivity of station (average)	2.3 ... 4.9	–	11.8 ... 13.4	21.2 ... 23.2	40.1 ... 44.5
– daily, th. m ³	0.7 ... 1.5	0.7 ... 1.5	–	6.1 ... 6.7	11.6 ... 12.8
– yearly, mln. m ³					
Defined power, kW	86	100	–	450 ... 640	1020 ... 1180
Numbers of compressors	1	1	2	3	2 ... 5
Power of one compressor, kW	80	75	100	130 120	320 160
Productivity of one compressor (average), m ³ /hour	153 ... 330	270	400	720 800	1380 780
Gas pressure at the side of suction, MPa	0.05 ... 0.23	0.15 ... 0.25	0.3 ... 0.6	0.6 ... 1.2	0.4 ... 0.6 0.8 ... 1.2
Injection pressure, MPa	23.0	25.0	32.0	25.0	25.0
Numbers of refills	10	10	4	6. 8	8
Average numbers of automobiles, that are refilled with natural gas (P=20 MPa), unit/day	60 ... 100	75	200	400	800
Geometrical capacity of a tank, m ³	1.0	1.0	5.0	5.0 10.0	2×9.0

Hydrocarbon gases don't correspond to the ideal gas equation $PV=RT$. That is why, parameters of real gases are defined according to the equation $PV=ZRT$, where the compression coefficient Z is a variable value, that depends on composition, pressure and the temperature of gas.

Thermodynamic parameters of natural gas can be determined either by the accurate equation of state, which contains a large number of variables and solution of it requires complex software and powerful PC or by using empirical relationships obtained by summarizing experimental results based on the law of corresponding states of carbohydrate gases.

The course project is recommended to perform by the technique that involves the use of generalized empirical relationships to determine the parameters of natural gas with a given composition.

1. The density of natural gas (in kg/m^3) according to normal conditions ($t_0=0\text{ }^\circ\text{C}$; $P_0 = 101.3\text{ kPa}$) is defined like:

$$\rho = \sum \frac{\mu_i r_i}{22.4}, \quad (1)$$

where r_i is a volume part of the i^{th} component of natural gas (in parts of a whole); μ_i is a molecular mass of the i^{th} component; $22.4\text{ m}^3/\text{k-mol}$ is a volume of one kilomole according to normal conditions.

2. Relative gas density by air:

$$\Delta_{air} = \frac{\rho}{1.293}, \quad (2)$$

where 1.293 kg/m^3 is density of air according to normal conditions.

3. Natural gas constant is determined with the help of Δ_{air} :

$$R = \frac{0.287}{\Delta_{air}}, \quad (3)$$

where $R_{air} = 287\text{ kJ/kg}\cdot\text{K}$ is a gas constant.

4. Pseudocritical values of temperature (in K) and pressure (in MPa) of natural gas is defined by formulas:

$$T_{cr} = 163.8 \cdot (0.613 + \Delta_{air}), \quad (4)$$

$$P_{cr} = 0.1 \cdot (47.9 - \Delta_{air}). \quad (5)$$

5. Given critical parameters of natural gas are determined with the help of specified parameters of gas pressure P and temperature T :

$$\pi = \frac{P}{P_{cr}}, \quad \mathcal{G} = \frac{T}{T_{cr}}. \quad (6)$$

Parameters of natural gas can be calculated for any tested cross-section of a compressor unit: for a compressor inlet (P_1 and T_1), for a stage outlet (P_2 , T_2), for average parameters of compression process (P_{av} , T_{av}), for gas cooling heat exchanger, etc.

All other parameters of natural gas are calculated by reduced parameters π and \mathcal{G} .

6. Compressibility coefficient of natural gas is determined by a formula:

$$Z = 1 - \left(\frac{0.41}{\mathcal{G}^3} - \frac{0.061}{\mathcal{G}} \right) \pi - \frac{0.04}{\mathcal{G}^3} \pi^2. \quad (7)$$

7. Function of isobaric compressibility of natural gas is calculated like this:

$$\chi = \frac{\pi}{\mathcal{G} \cdot Z} \left(\frac{1.23}{\mathcal{G}^2} - 0.061 + \frac{0.12}{\mathcal{G}^2} \pi \right). \quad (8)$$

8. Mole isobaric heat capacity of natural gas at ideal gas condition is calculated as:

$$\mu c_{p0} = 21.563 + (23.656 + 0.071 \cdot t) \cdot \Delta_{air}, \quad (9)$$

where t , °C is a gas temperature.

Note, that ideal gas condition, of real gas is a condition when absolute gas pressure approaches to zero $P \rightarrow 0$.

Mass isobaric gas heat capacity at the ideal gas condition is determined as the ratio:

$$c_{p0} = \frac{\mu c_{p0}}{\mu}, \quad (10)$$

where μ is a molar mass of natural gas, that is

$$\mu = \sum \mu_i r_i. \quad (11)$$

9. Mass heat capacity of hydrocarbon gases at specified parameters P i T is determined as

$$c_p = c_{p0} + \Delta_{air}c_p, \quad (12)$$

where $\Delta_{air}c_p$ is a natural gas heat capacity deviation from an heat capacity at the ideal gas condition, which is determined by formula:

$$\frac{\Delta_{air}c_p}{R} = \frac{6\pi}{g^3}(0.41 + 0.02\pi), \quad (13)$$

where R is a gas constant of natural gas.

10. Adiabatic factor of the natural gas compression process at the ideal gas condition is as follows:

$$\frac{k_0}{k_0 - 1} = \frac{\mu c_{p0}}{R_0} = \frac{\mu c_{p0}}{8.3144}, \quad (14)$$

where $R_0 = 8.3144$ kJ/k-mol·K is a universal gas constant.

11. A pseudoisentropic factor of natural gas compression process at the compressor stage is determined with the help of a polytrophic coefficient of the compressor stage efficiency η_{pol} by the formula

$$\frac{k_p}{k_p - 1} = \frac{k_0}{k_0 - 1} \frac{1 + \frac{\Delta_{air}c_p}{R} \frac{k_0 - 1}{k_0}}{Z(1 + \eta_{pol}\chi)}, \quad (15)$$

where $\frac{\Delta_{air}c_p}{R}$ and χ are determined by formulas (13) and (8) accordingly.

A polytrophic efficiency of a piston compressor stage can be accepted in the range $\eta_{pol} = 0.73$ K 0.76 .

12. Polytrophic thermal factor of the process is determined by the ratio:

$$\frac{n_T}{n_T - 1} = \eta_{pol} \frac{k_p}{k_p - 1}, \quad (16)$$

and the temperature after the compression process is calculated according to the equation:

$$T_2 = T_1 \left[\frac{P_2}{P_1} \right]^{\frac{k_p - 1}{k_p \eta_{pol}}} = T_1 \left[\frac{P_2}{P_1} \right]^{\frac{n_T - 1}{n_T}}. \quad (17)$$

13. Internal head in (kJ/kg) of a compressor stage (operation, which is applied to 1 kg of natural gas taking into account all the energy losses except the mechanical energy losses in the bearings of a compressor) is determined by the formula:

$$H_i = \frac{k_p}{k_p - 1} ZR(T_2 - T_1). \quad (18)$$

14. Polytrophic head in (kJ/kg) of compressor stages is determined by the formula:

$$H_{pol} = \frac{n_T}{n_T - 1} = ZRT_1 \left[\left(\frac{P_2}{P_1} \right)^{\frac{n_T - 1}{n_T}} - 1 \right]. \quad (19)$$

Calculation results of natural gas physical properties are summarized in the table 2.

Таблиця 2

Результати розрахунків фізичних властивостей газу

Parameter	Method of definition
Molecular gas weight μ , kg/k-mol	by the formula (11)
Density ρ , at $t=0$, °C $P=101.325$, kPa	by the formula (1)
Relative gas density to that of air Δ_{air}	by the formula (2)
Mole isobaric heat capacity of natural gas at ideal gas condition at $t_1=15$ °C μc_{p0} , kJ/k-mol·K	by the formula (9)
Gas constant R , kJ/kg·K	by the formula (3)
Pseudocritical parameters: pressure P_{cr} , MPa; temperature T_{cr} , K	by the formula (4) and formula (5)
Adiabatic factor of natural gas at ideal gas condition $k_0/(k_0-1)$	by the formula (14)

2.2. Thermodynamic calculation of a piston compressor

The purpose of thermodynamic calculation of a piston compressor is to determine gas parameters at different stages of its compression, to select a compressor type and its main sizes and choose the engine type.

Initial data required for the calculation:

- the number of refills per day;
- gas pressure (as a rule, 22.6...24.7 MPa) at the outlet from the compressor unit;
- physical parameters of a suction gas (pressure P_1 , MPa according to variant), temperature $t_1=15$ °C, density ρ_1 , kg/m³ (according to previous calculation);

– relative gas density Δ_{air} ;

– temperature of a cooler (water from $t_w=20$ °C).

The following order of piston compressor calculation is stated:

- selection of compressor stage numbers;
- determination of suction and pumping pressures for separate stages;
- determination of gas parameters by compressor stages;
- determination of volumetric coefficients by stages;
- determination of the gas suction amount by compressor stages;
- choosing of a type and a schemes of a compressor;
- choosing of piston stroke and rotation frequency;
- determination of stages of cylinders diameters;
- determination of compressor power consumption and selection of an engine.

2.2.1. Selection of compressor stage numbers

Number of compressor stages is determined by the formula

$$Z_{st} = \frac{\lg \frac{P_{out}}{P_1}}{\lg \frac{\varepsilon_{st}}{\psi}}, \quad (20)$$

where ε_{st} is a degree of the pressure increase in one compressor stage (the value of ε_{st} for compressors of NPGV filling stations is accepted in the range $\varepsilon_{st}=2...4$; $\psi=1.1...1.15$ is a pressure losses coefficient at the intermediate coolant between the stages).

The value of Z_{st} is selected as an integer number, after that the average degree of the pressure increase at one stage is calculated by means of a formula:

$$\varepsilon_{st} = \psi \left(\frac{P_{out}}{P_1} \right)^{1/Z_{st}}. \quad (21)$$

NOTE! The most of compressors at NPGVFS have 4...5 stages.

2.2.2. Determination of suction and pumping pressures for separate stages

Minimal indicator operation of gas compression in a multi-stage compressor can be achieved only under a condition of optimal values of interstage gas pressures.

From the compression machines theory, it is known, that the distribution of optimal pressures between the stages does not correspond to the equality of the pressure degree increase in different stages. In order to determine optimal interstage pressures it is recommended to use nomogram, shown on (fig. 1).

The nomogram is designed for the initial gas pressure before the first stage $P_{suc_i}=0.1$ MPa (excessive pressure). To determine the optimal interstage pressures by means of the nomogram it is necessary to find a point of intersection of a horizontal line which corresponds to the final pressure of pumping ($P_{out}=22.6$ or 24.7 MPa), with an inclined line that corresponds to the accepted number of the compressor stages Z_{st} .

A vertical line, drawn from this point, determines the value of the optimal pressure after each stage, a number of which is marked on other inclined lines (at points of intersection of a vertical line with inclined ones).

As an example, data of interstage pressures for one of modern compressors which is used at a garage type of NPGV filling station-50 are given in the table 3.

A compressor is built on the "W-type" base, the number of stages is $Z_{st}=5$, the final pressure $P_{out}=23.4$ MPa (the pressure $P_{out}=23.4$ MPa means that a given construction of NPGV filling station with accumulating capacity has a block of standard automobile gas balloons of high pressure, which are designate for the same operation pressure 23.5 MPa).

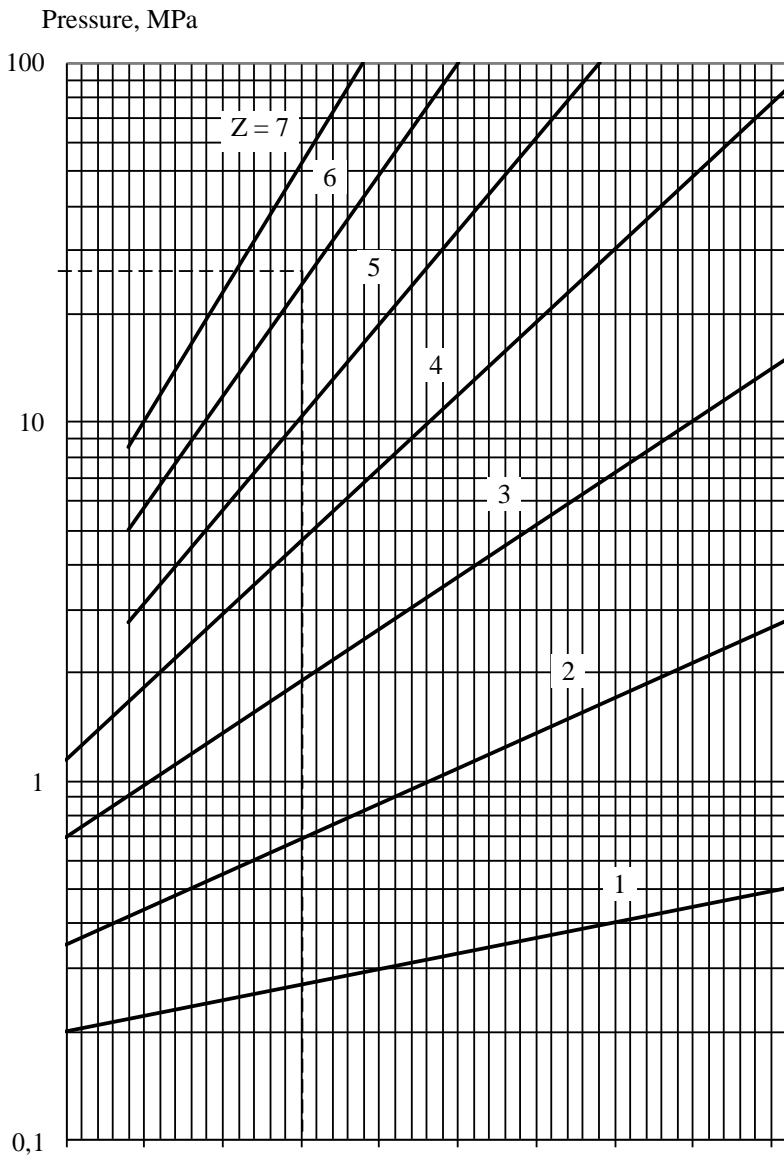


Fig. 1. Nomogram of determination of an optimal value of nominal interstage pressures at specified final pressure in an outlet of a compressor according to the numbers of compression stages

According to the data in (fig. 1) and (tab. 3), let's determine the gas pressure value from the side of suction and the pumping pressure for each compressor stage.

Table 3

Gas parameters for NPGVFS-50 according to compressor stages at different values of pressure before a compressor

Excessive pressure at an inlet to the first stage of a comp., MPa	Pressure at the stages outlet, MPa					Maximum allowable gas temperature at an outlet from a stage, °C				
	I	II	III	IV	V	I	II	III	IV	V
0.05	0.24÷0.30	0.9÷1.10	2.0÷2.5	6.0÷6.8	23.4	123	148	130	143	173
0.10	0.43÷0.53	1.25÷1.55	2.7÷3.3	7.2÷8.2	23.4	123	148	130	143	173
0.15	0.52÷0.64	1.55÷1.95	3.3÷4.1	8.5÷9.5	23.4	123	148	130	143	173
0.20	0.64÷0.79	1.95÷2.35	3.9÷4.7	9.7÷10.7	23.4	123	148	130	143	173
0.25	0.85÷0.95	2.35÷2.75	4.6÷5.5	11.2÷12.2	23.4	123	148	130	143	173

It should be taken into account, that excessive pressure P_1 before a compressor unit, is determined by the task for a course project and should decrease due to the hydraulic losses of a suction line of a compressor, and these losses are equal to 3...6%.

That is why the suction pressure of the first stage would be $P_{suc} = \delta_{suc} P_1$, where $\delta_{suc} = 0.97 \dots 0.94$.

The pumping pressure for each stage ($P_{inj\ i}$, $P_{inj\ II}$, $P_{inj\ III}$, ..., $P_{inj\ Z_{st}}$) should be equal to the value which is determined as optimal.

The suction pressure of each next stage is less than the pumping pressure of the previous stage by the value of the energy losses at the interstage gas coolant, and this value is calculated by means of a coefficient ψ :

$$P_{sucII} = \frac{P_{inj\ i}}{\psi}; P_{sucIII} = \frac{P_{inj\ II}}{\psi}; P_{sucIV} = \frac{P_{inj\ III}}{\psi}; P_{suc(i)} = \frac{P_{inj(i-1)}}{\psi}.$$

Calculation results are summarized into the table 4.

Table 4

Pressure distribution between stages of a compressor

Stage number	Suction pressure, MPa	Pumping pressure, MPa	Degree of pressure increase
I	$P_{suc\ I} = \delta_{suc} P_1$	$P_{inj\ I}$	$\varepsilon_I = P_{inj\ I} / P_{suc\ I}$
II	$P_{suc\ II} = P_{inj\ I} / \psi$	$P_{inj\ II}$	$\varepsilon_{II} = P_{inj\ II} / P_{suc\ II}$
III	$P_{suc\ III} = P_{inj\ II} / \psi$	$P_{inj\ III}$	$\varepsilon_{III} = P_{inj\ III} / P_{suc\ III}$
...
Z_{st}	$P_{suc\ Z_{st}} = P_{suc} (Z_{st} - 1) / \psi$	$P_{suc\ i\ Z_{st}} = P_{out}$	$\varepsilon_{Z_{st}} = P_{inj\ i\ Z_{st}} / P_{suc\ i\ Z_{st}}$

2.2.3. Determination of gas parameters by compressor stages

Absolute gas temperature from the suction side to the first stage of a compressor is accepted as $T_1 = 288 \text{ °K}$ ($t_1 = 15 \text{ °C}$).

To all other stages of a compressor, gas is sucked after cooling in the interstage cooling heat-exchangers.

Gas temperature at the outlet from the interstage heat-exchanger is accepted $5 \dots 8 \text{ °C}$ higher than temperature of cooling water, which is equal to $t_{cool} = 20 \text{ °C}$ at summer period.

This value of gas temperature could be taken for all stages, except the first stage.

Gas temperature for the first stage should be accepted according to the simple ratio:

$$t_1 = t_{cool} + (5 \dots 8) \text{ °C}.$$

Gas temperature at the end of a stage compression process is calculated by the formulas (15...17). Calculation order is listed in the table 5.

According to the results of calculations, the maximum allowing injection temperature value is determined.

Otherwise, it is necessary to take measures to reduce the injection temperature (by more intensive cooling in the interim refrigerator or by increasing the number of stages).

Table 5

Determination of parameters of gas by stages

Parameter	Method of definition	Dimension	Parameter values by stages				
			I	II	III	IV	V
Suction pressure, P_{suc}	from the table 4	MPa					
Injection pressure, P_{inj}	from the table 3 and 4	MPa					
Average pressure of a stage, P_{av}	$P_{av}=0.5(P_{in}+P_{inj})$	MPa					
Temperature at the inlet of a stage, T_{in}	accepted as	K	288	296	296	296	296
Critical temperature, T_{cr}	by the formula 4	K					
Critical pressure, P_{cr}	by the formula 5	MPa					
Reduced pressure, π	$\pi=P_{av}/P_{cr}$	-					
Reduce temperature, \mathcal{G}	$\mathcal{G}=T_{in}/T_{cr}$	-					
Compressibility factor, Z_1	by the formula 7	-					
Function of the isobaric heat capacity, χ	by the formula 8	-					
Mole isobar heat capacity of an ideal gas state, μc_{p0}	by the formula 9	kJ/k-mol·K					
Deviations of the isobaric heat capacity, $\Delta_{air}c_p/R$	by the formula 12	kJ/k-mol·K					
Adiabatic index of gas in the ideal gas state	$\frac{k_0}{k_0-1} = \frac{\mu c_{p0}}{R_0}$						
Polytropic efficiency level, η_{pol}	the same as the comp-ressor efficiency η_{pol}						
Index of pseudoisentrope, $k_p/(k_p-1)$	by the formula 15						
Index of a polytropic process of gas compression degree, $n_T/(n_T-1)$	by the formula 16						

The end of the table 5

Parameter	Method of definition	Dimension	Parameter values by stages				
			I	II	III	IV	V
Gas temperature at the outlet from the stage, T_2	by the formula 17	K					
Reduced pressure at the outlet of the stage	$\pi_2 = P_{inj}/P_{cr}$	-					
Reduced temperature at the outlet of the stage	$\vartheta_2 = T_2/T_{cr}$	-					
Compressibility factor at the outlet stage parameters, Z_2	by the formula 7	-					

2.2.4. Determination of volumetric coefficients by stages

For the stages of high pressure a volume ratio is calculated by this formula:

$$\lambda_0 = 1 - \alpha \left(\varepsilon^{n_r} \cdot \frac{Z_1}{Z_2} - 1 \right). \quad (22)$$

The relative value of the dead space for the first stage can be taken as $\alpha_1 = 0.05$.

For the next stages the relative dead space is increased by reducing the volume of cylinders and is defined by the formula:

$$\alpha_i = \alpha_1 + 0.025(i - 1), \quad (23)$$

where i is a stage number.

The values of compressibility coefficients Z_1 (by the parameters of a gas at the inlet of a stage) and Z_2 (by the parameters at the outlet of stage) are taken according to the table 5.

The polytropic coefficient of gas expansion in the dead space is considered to be the same as for the process of gas compression in the stage cylinder (calculated according to the table 5).

The output data and the results of the volumetric coefficients calculation are presented in the table 6.

Table 6

Determination of volumetric coefficients

Number of stage	α	n_T	$\frac{1}{\varepsilon^{n_T}}$	λ_0
I	α_1	n_{T1}	$\frac{1}{\varepsilon^{n_{T1}}}$	λ_{0I}
II	α_2	n_{TII}	$\frac{1}{\varepsilon^{n_{TII}}}$	λ_{0II}
...
Z_{st}	α_{Zst}	n_{TZst}	$\frac{1}{\varepsilon^{n_{TZst}}}$	λ_{0Zst}

2.2.5. Determination of the gas suction amount by compressor stages

To ensure the specified productivity conditions of a compressor under suction conditions it is necessary that the working volume of the cylinder of the first stage equals to

$$Q_{PI} = \frac{Q_I}{\lambda_{0I} \lambda_{ef}}. \quad (24)$$

The absorption efficiency coefficient λ_{ef} can be determined by the formula:

$$\lambda_{ef} = 1.01 - 0.022 \varepsilon_{st}. \quad (25)$$

The operation volume of cylinders (m^3/min) these stages can be determined by the formula:

$$Q_{Pi} = \frac{Q_{PI}}{\lambda_{0I} \lambda_{ef}} \frac{Z_{1i}}{Z_{2i}} \frac{P_{suc1}}{P_{suc(i)}} \frac{T_{in1}}{T_1}, \quad (26)$$

where Z_{1i} and Z_{2i} are characteristics coefficients at suction pressure $P_{suc i}$ and $P_{inj i}$.

The calculation results are shown in the table 7.

Table 7

Operation volume of stages cylinders

Stage number	$Q_p, m^3/min$
I	
...	
Z_{st}	

2.2.6. Choosing of a type and a schemes of a compressor

A type and a scheme of a compressor are determined mainly by the location of a cylinder in space by the number of rows and by the position of stages in a row. During selection of cylinders in space, attention should be paid to the fact that the compressor features that are derived from the received cylinders position should meet the conditions of its operation.

For the compressors of low productivity, first of all, the demand for compactness and simplicity of maintenance is required. For compressors of high productivity reliability of operation is more important, although the problem of compactness and maintainability shouldn't be excluded from consideration.

The requirements of reliability and service life are especially important when a compressor is designed for continuous operation without interruption. Vertical location of the cylinders has advantages in comparison to horizontal location:

1. Less intense wearout of the cylinder and the piston;
2. More favorable impact of forces inertia of the compressor moving parts on the foundation;
3. Possibility to reduce bedplate reduction weight due to a lack of bending stresses;
4. Possibility to increase compressor rotation frequency;
5. Less area occupied;
6. Provision of cylinder thermal and elastic deformations without special supports.

In its turn, the horizontal compressors have got the following advantages over the vertical ones:

1. Low height, convenient exploitation and maintenance;
2. Possibility to install in the building of low height;
3. Possibility to install in one row of larger amount of the stages, allows to construct a multi stage compressor with the small number of rows.

Each compressor row has its own motion mechanism. Hence, the reducing of row numbers leads to the simplicity of a compressor. On the other hand, the bigger number of rows leads to reducing of mass of the movement parts and the action of inertia forces on a bedplate and base.

Compressors execution layout is given on the fig. 2.

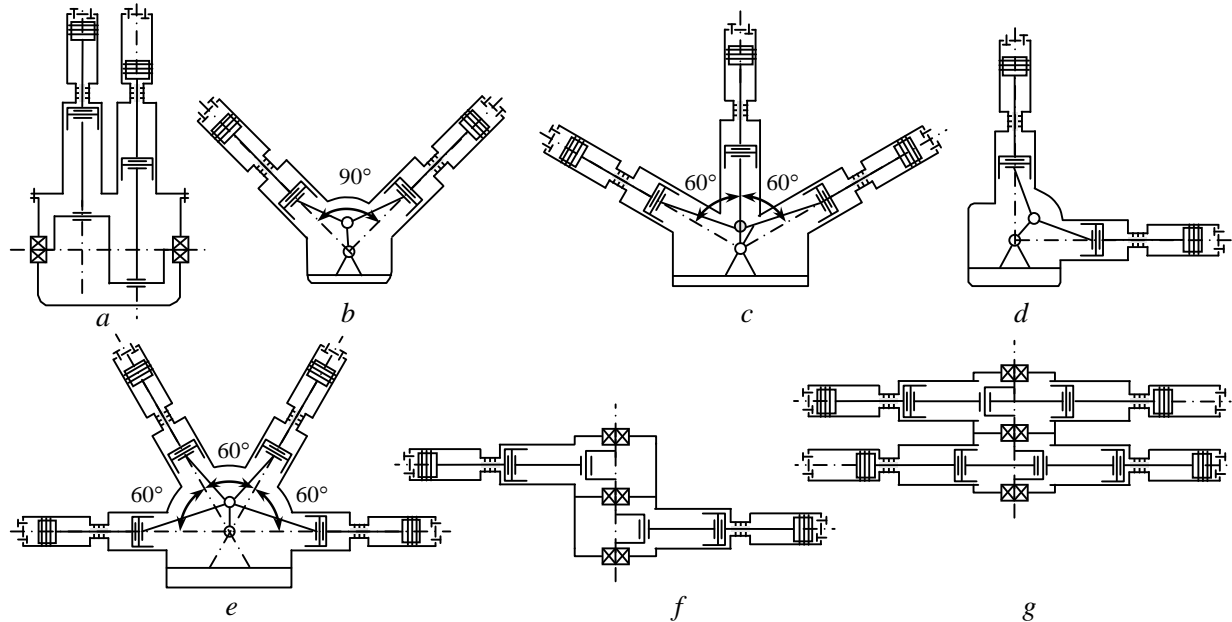


Fig.2. Cross head compressors scheme of constructive operation:

a – a vertical one with the number of cylinders - 2; *b* – a V-type compressor with the number of cylinders - 2, 4, 6; *c* – a W-type compressor with the number of cylinders - 3, 6; *d* – a trunk compressor with the number of cylinders - 2, 4, 6; *e* – an X-type compressor with the number of cylinders - 4 and 8; *f* – a flat double-row compressor with the number of cylinders - 2, 4, 6, 8; *g* – a flat double-row compressor with the number of cylinders in the 1st row – 2, 4, in the 2nd row – 2, 4 with - 4, 8 compression stages respectively

2.2.7. Choosing of piston stroke and rotation frequency

The piston velocity m/s depends on the piston stroke S and the rotational speed n :

$$C_{av} = \frac{2Sn}{60}. \quad (27)$$

The average piston velocity in its turn, influences cylinder and piston wearing, moving parts inertia forces magnitude, valve plate velocity, and gas velocity in the valve slots.

Hence, it is one of the factors which determine compressor durability.

The average piston velocity is generally accepted from 3 to 5 (m/s), sometimes up to 7 (m/s). In order to properly arrange the compressor, it is reasonable to connect the average piston velocity and the 1st stage cylinder diameter. In order to reduce the height of cross-head type vertical compressors it is usually considered that:

$$\frac{S}{D_1} = 0.4 \dots 0.6,$$

where D_1 is the 1st stage cylinder diameter.

2.2.8. Determination of stages of cylinders diameters

In a cause, if a piston of the first stage is not differential, its theoretical volume (m³/min) is defined by formula:

$$Q_{Pi} = \frac{\pi D_1^3}{4} Sn. \quad (28)$$

As it follows from the expression (27), $Sn=30C_{av}$, therefore operating volume of the 1st stage cylinder (m³/s) is calculated as follows:

$$Q_{Pi} = \frac{\pi D_1^3}{4} 30C_{av}. \quad (29)$$

The last relation allows to determine the 1st stage cylinder diameter:

$$D_1 = \sqrt[3]{\frac{4Q_{Pi}}{30\pi C_{av}}}. \quad (30)$$

In a similar manner, for the i^{th} stage with a circle shaped operation area it can be calculated as:

$$D_i = \sqrt{\frac{4Q_{pi}}{30\pi C_{av}}}. \quad (31)$$

In a cause, when a piston is differential and has a ring-shaped operation area the previous stage diameter is calculated as follows:

$$D_i = \sqrt{\frac{4(Q_{PI} + Q_{pc})}{30\pi C_{av}}}, \quad (32)$$

where Q_{pc} is capacity of a near by (next) stage.

After D_1 determination and using the ratio $S_1D_1=0.4\dots0.6$, the piston stroke S could be calculated.

Calculation results are listed in the table 8.

Table 8

Cylinders diameters by stage

Stage number	D_i , m
I	
II	
III	
Z_{st}	

2.2.9. Determination of compressor power consumption and selection of an engine

The compressor power is determined as the sum of power of all stages. Compressor mass efficiency (kg/s) in a suction condition is calculated as follows:

$$M_{m.ef} = \frac{\rho_g Q_{PI}}{60}, \quad (33)$$

where ρ_g is gas density (kg/m³) at initial conditions before entering the 1st stage.

Therefore,

$$\rho_g = \frac{\rho_0 P_1 T_0}{P_0 T_1 Z_1}. \quad (34)$$

Internal head pressure of a compressor stage (kJ/kg) is calculated as follows:

$$H_{in} = \frac{k_p}{k_p - 1} Z_1 R (T_2 - T_1). \quad (35)$$

Internal head pressure is the total energy supplied to 1kg of gas in the stage taking into account all energy losses except of mechanical energy losses in bearings.

The power (kW) that is consumed by any compressor stage is the product of internal head pressure and a mass loss of gas per second:

$$N_{st} = M_{m.ef} H_{in}. \quad (36)$$

The multistage compressor power can be determined by the following formula:

$$N_c = M_{m.ef} \sum H_{in}, \quad (37)$$

where $M_{m.ef}$ is a mass compressor efficiency, kg/s; $\sum H_{in}$ is a sum of internal head pressures of all stages.

The shaft power of a compressor is determined by the following formula:

$$N_{SP} = \frac{N_c}{\eta_{m.ef}} K_{ep}, \quad (38)$$

where $\eta_{m.ef}$ is a compressor mechanical efficiency, which includes losses in bearings and electric motor transmissions.

Excessive power coefficient is usually in the following range $K_{ep}=1.10\dots1.15$.

3. CALCULATION OF COMPRESSOR AND BATTERY PACK OPERATION

Rapid filling of a compressed gas vehicle at a NPGV filling station is provided from the tanks, that have the initial pressure in the limits $P_{1tank}=23.0\dots25.0$ MPa at the constant temperature $T_{tank}=288\dots303$ K.

Automobile filling is carried out at shut-down NPGV station compressors, at the same time initial pressure at the tank decrease to the value of $P_{2tank}=21.0\dots22.0$ MPa at the constant temperature T_{tank} .

After achieving of the pressure P_{2tank} a performed automatic launching of compressors is done that supply compressed gas to tanks and increase the pressure up to the initial value P_{1tank} .

In a case when geometrical volume of the tank is equal to V_{tank} (m^3) then gas capacity Q_{tank} (m^3), that is supplied for filling of compressed gas vehicles from tanks and reduced to the normal conditions ($P_0=101.3$ MPa and $T_0=273$ K), is calculated by natural gas condition equation:

$$Q_{tank} = V_{tank} \frac{P_{1tank} - P_{2tank}}{P_0} \frac{Z_0}{Z_1} \frac{T_0}{T_{tank}}, \quad (39)$$

where Z_1 is a natural gas compression coefficient in accordance with the parameters P_{1tank} and T_{tank} ; Z_0 is a compression coefficient at P_0 and T_0 .

The value of coefficients Z_1 and Z_0 is calculated by formulas (6) and (7).

The volume of one industrial truck filling is $55...60 m^3$, taking into account that most of the cars, that arrive to NPGV station for filling have pressure at their tanks at about $2.5...3.5$ MPa.

In a case when the amount of cars that can be filled at the same time on NPGV station is equal to n_{auto} per one hour (this value depends on the number of filling points), then the time of pressure decreasing in (hours) at the tank from P_{1tank} to P_{2tank} is equal:

$$\tau_1 = \frac{Q_{tank}}{n_{auto} q_f}, \quad (40)$$

where $q_f=55...60 m^3$ /vehicle.

For providing the filling of n_{auto} vehicles per one hour the total efficiency of compressors (m^3 /hour) should be:

$$Q_c \geq n_{auto} q_f. \quad (41)$$

In this cause, the value that was calculated by formula (41) is less than the initial one (given at the beginning of the course project), then the number of n_{auto} should be decreased.

In cause of a compressor installation turning on during vehicle filling from a tank, time of pressure increasing in (hours) from P_{2tank} to P_{1tank} in a tank should be:

$$\tau_2 = \frac{Q_{tank}}{Q_c}. \quad (42)$$

The maximum number of launching and shutting-down of a compressor per day should be:

$$m = 24 / \tau_1. \quad (43)$$

Combined calculation of the equations (39), (40) and (43) gives an opportunity to define the volume of a NPGV station tank (m^3) for providing the necessary amount m of shutting-down and launching of compressor units per day:

$$V_{tank} = \frac{24n_{auto}q_f}{m(P_{1tank} - P_{2tank})/P_0} \frac{Z_1 T_{tank}}{Z_0 T_0}. \quad (44)$$

4. SCIENTIFIC AND RESEARCH PART OF COURSE PROJECT

A scientific and research part of a course project is a necessary element of it, the purpose of it is:

- developing of students skills of a scientific and reasonable analysis of piston compressors quality that are supplied to the enterprises also skills for searching faults that cause emergency situations of piston compressors and provide activities for accident prevention;

- promotion of accomplishing skills in research activities, that consist of literature references approbation of a subject an in international practice;

- promotion of creative ideas of students, the ability to implement the results of theoretical research into the constructive solutions;

- promotion of development of work skills with periodical and special literature.

The title of the scientific and research part of the project is chosen by the student and approved by lecturer during the course project preparation.

The scientific and research part of a project is carried out as a themes reference view of works, that are published in periodical and special books. At the same time the chief of the project defines only the minimal amount of academic books. The further search of the necessary materials is done by a student.

The scientific and research part of the course project should consist of one page of A1 format.

LIST OF LITERATURE

1. *Биков Г.О.* Автомобільні газонаповнювальні компресорні станції. / [Г.О. Биков, К.І. Капітанчук, М.С. Кулик та ін.]. – К.: НАУ, 2006. – 292 с.
2. *Биков Г.О., Бикова О.Г.* Термодинаміка реальних макросистем і вуглеводневих газів: Навч. посіб. – К.: НАУ, 2003. – 134 с.
3. *Гайнулин Ф.Г.* Природный газ как моторное топливо на транспорте. / [Ф.Г. Гайнулин, А.И. Гриценко, Ю.Н. Васильев и др.]. – М.: Недра, 1986. – 255 с.3.
4. *Елин В.И.* Насосы и компрессоры. / [В.И. Елин, К.Н. Солдатов, С.М. Соколовский и др.]. – М.: Гостоптехиздат, 1958. – 371 с.
5. *Фотин Б.С.* Поршневые компрессоры. / [Б.С. Фотин, И.Б. Пирумов, И.К. Прилуцкий и др.]. - Л.: Машиностроение, 1987. – 372 с.
6. *Храпач Г.К.* Машинист компрессорной станции. – М.: Недра . 1966. – 345 с.

Natural gas compound variants

Variant number	Compound of natural gas, volume percentage, %						
	CH ₄	C ₂ H ₆	C ₃ H ₈	C ₄ H ₁₀	C ₆ H ₁₂	N ₂	CO ₂
1	92.6	4.3	1.0	0.36	0.14	1.5	0.1
2	98.7	0.11	0.01	0.04	0.01	1.0	0.13
3	85.4	5.4	3.5	1.65	0.65	3.0	0.4
4	93.5	2.2	0.5	0.54	0.16	2.6	0.5
5	95.3	0.04	0.05	0.01	-	4.2	0.4
6	85.2	5.5	2.1	0.9	0.3	5.0	1.0
7	93.0	4.1	0.72	0.26	0.08	1.54	0.3
8	86.5	0.26	0.11	0.05	0.02	13.0	0.06
9	93.0	0.34	0.12	0.11	0.03	6.0	0.4
10	95.8	0.66	0.36	0.28	0.13	2.2	0.57
11	97.0	1.14	0.57	0.54	0.1	0.47	0.18
12	91.5	4.4	1.24	0.47	0.19	1.8	0.4
13	92.0	3.6	1.0	0.5	0.3	2.0	0.6
14	89.3	5.8	1.0	0.8	0.2	2.6	0.3
15	93.0	3.4	1.4	0.25	0.05	1.3	0.6
16	86.5	6.5	2.6	1.7	0.1	2.0	0.6
17	91.9	3.4	0.85	0.36	0.17	2.8	0.52
18	95.9	0.7	0.2	0.09	0.01	3.0	0.1
19	78.3	11.9	4.5	1.7	0.5	2.1	1.0
20	86.4	6.8	2.3	1.0	0.5	2.5	0.5
21	84.2	3.8	1.2	0.46	0.24	9.1	1.0
22	82.7	4.7	2.9	1.5	0.2	7.5	0.5
23	96.4	0.6	0.4	0.14	0.06	2.1	0.3
24	94.2	2.7	0.6	0.55	0.15	1.6	0.2

**Tasks by NPGVFS variants according to the number of fillings,
efficiency, physical parameters of gas suction**

Variant number	NPGVFS parameters		
	The number of fillings	Efficiency, m ³ /day	Inlet pressure, MPa
1	50	75	0.2
2	75	110	0.2
3	125	180	0.5
4	150	220	0.5
5	60	85	0.2
6	500	750	1
7	250	360	1
8	250	360	1
9	75	110	0.2
10	75	110	0.2
11	50	75	0.2
12	50	75	0.2
13	30	45	0.2
14	125	180	0.8
15	100	150	0.8
16	75	110	0.5
17	500	750	1
18	50	75	0.2
19	500	750	1
20	150	220	0.5
21	500	750	1
22	250	360	1
23	75	110	0.5
24	50	75	0.2

Навчальне видання

**АВТОМОБІЛЬНІ
ГАЗОНАПОВНЮВАЛЬНІ
КОМПРЕСОРНІ СТАНЦІЇ**

Методичні рекомендації
до виконання курсового проекту
для студентів спеціальності 142 «Енергетичне машинобудування»
спеціалізації «Газотурбінні установки і компресорні станції»

Укладачі:
КАПІТАНЧУК Костянтин Іванович
БОГДАНОВ Микола Юрійович
ДЕНИСЕНКО Наталія Григорівна

(Англійською мовою)

Редактор
Технічний редактор

Підп. до друку . Формат 60×84/16. Папір офс.
Офс. друк. Ум. друк. арк. Обл.-вид. арк. .
Тираж 100 пр. Замовлення №

Видавець і виготівник
Національний авіаційний університет
03068, Київ-58, проспект Космонавта Комарова, 1

Свідоцтво про внесення до Державного реєстру ДК, № 977 від 05.07.2002