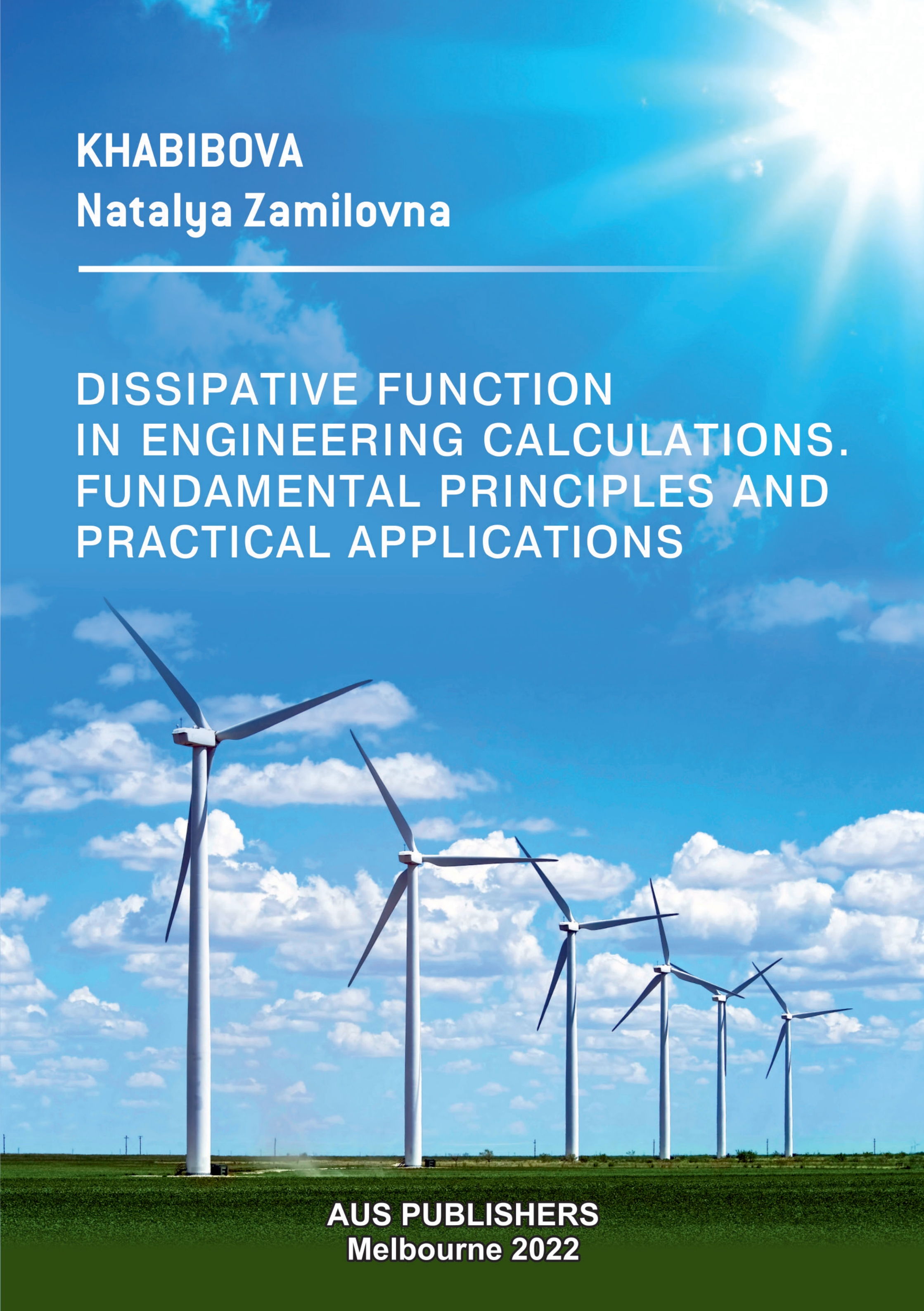

KHABIBOVA
Natalya Zamilovna

DISSIPATIVE FUNCTION IN ENGINEERING CALCULATIONS. FUNDAMENTAL PRINCIPLES AND PRACTICAL APPLICATIONS

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Khabibova Natalya Zamilovna

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Monograph

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The book brought to the attention of the audience is devoted to the analytical apparatus and quantitative assessment of the effectiveness of processes occurring in real thermodynamic systems. It is obvious that a reliable qualitative characterization and a quantitative description of phenomena occurring at finite speeds in apparatuses of specific dimensions require consideration of thermodynamic limitations, the essence of which is expressed by the second law of thermodynamics of irreversible processes of energy and matter conversion. The main idea of the presented material is to determine the produced entropy flow generated within the process as a quantitative measure of energy dissipation. The calculation method based on the concept of a local dissipative function is considered in sufficient detail. For clarity and the ability to reveal the internal logic of the interpretation of the material, examples of the flow of viscous media, processes of heat conduction and diffusion membrane transfer of matter are demonstrated.

The monograph will serve as a source of useful information for students and graduate students of technological universities and colleges, as well as attract the attention of young scientists in solving global problems of energy and resource conservation in the modern world, the search for alternative energy sources.

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List of symbols

$\rho_{\text{ид}}$ – ideal gas density, kg/m^3;
ρ – density of nonideal gas, kg/m^3;
$R_{\text{м}}$ – specific gas constant, J/(kg·K);
M – molar mass, kg/kmol;
V – specific volume, m^3/kg;
T – temperature, K;
P – pressure, Pa;
T_{c} – critical temperature, K;
P_{c} – critical pressure, Pa;
$T_{\text{г}}$ – reduced temperature, dimensionless value;
B – second virial coefficient, m^3/kg;
Z – gas compressibility factor, dimensionless value;
$\Delta h_{\text{T}}^{\partial}$ – isothermal deviation of enthalpy from the ideal gas state, J/kg;
$h(T; P)$ – specific enthalpy, J/kg;
$H(T; P)$ – enthalpy, J;
$S(T; P)$ – entropy, J/(kg·K);
C_{V} – isochoric heat capacity, J/(kg·K);
C_{P} – isobaric heat capacity, J/(kg·K);
C_{n} – heat capacity of the polytropic process, J/(kg·K);
$G(T; P)$ – Gibbs energy, J;
$\dot{\Psi}_{12}^{\text{тр}}$ – dissipation of kinetic energy, W;
$\dot{W}^{\text{tex}}$ – technical power, W;
$\dot{Q}$ – convective heat flux, W;
$\dot{W}_{\text{пол}}$ – power of the polytropic process, W;
n – polytropic index;
k – adiabatic index;
E_{x} – exergy, J;
$\dot{E}\text{x}$ – convective exergy flux, W;
η_{S} – adiabatic efficiency;
η_{ex} – exergy efficiency;
$\eta_{\text{пол}}$ – polytropic efficiency;
ε_{x} – coefficient of performance;
α_{S} – differential effect of cooling in isentropic gas expansion, K/Pa;
α_{h} – differential effect of throttling, K/Pa;
$\dot{W}_{\text{x}}$ – power of the refrigeration device, kW;
$\dot{Q}_{\text{x}}$ – refrigeration capacity of RU, kW;
$\dot{E}_{\text{x тр}}$ – transit flow of exergy, kW;
$in\dot{S}$ – entropy production rate, kW/K;
$in\dot{D}$ – internal exergy losses, kW;
$ex\dot{D}$ – external exergy losses, kW;
I_S^V – local flow of entropy generated inside the process, $\frac{\text{kW}}{\text{K}\cdot\text{m}^3}$;
$\dot{\Psi}^V$ – local dissipative function, $\frac{\text{kW}}{\text{m}^3}$;
μ – dynamic viscosity of the medium, Pa·s.

Introduction

The book "Dissipative function in engineering calculations" brought to the attention of the audience is devoted to the analytical apparatus and quantitative assessment of the effectiveness of processes occurring in real thermodynamic systems. It is obvious that a reliable qualitative characterization and a quantitative description of phenomena occurring at finite speeds in apparatuses of specific dimensions require consideration of thermodynamic limitations, the essence of which is expressed by the second law of thermodynamics of irreversible processes of energy and matter conversion. The main idea of the presented material is to determine the produced entropy flow generated within the process as a quantitative measure of energy dissipation. The calculation method based on the concept of a local dissipative function is considered in sufficient detail. For clarity and the ability to reveal the internal logic of the interpretation of the material, examples of the flow of viscous media, processes of heat conduction and diffusion membrane transfer of matter are demonstrated. When considering the integral method of analyzing the perfection of irreversible processes and the phenomena that accompany them, the structure of the text adheres to an engineering orientation. When quantifying the degree of perfection of a thermodynamic system and determining the magnitude of the final losses of convertible energy, the author seeks to outline the main patterns by which an engineering apparatus for calculating and determining specific values of physical characteristics and process parameters is created. In the final chapters, the reader is offered an analysis of individual tasks, the logical sequence of presentation of which allows deepening the understanding of the problem raised. Since engineering analysis becomes useful and gives practical skills in the event that not only the course of the solution is considered and the numerical answer of the desired value is determined, but also ways are revealed to search for a more perfect organization of the technology and energy of the system under study. The author expresses the hope that the publication will serve as a source of useful information for students and graduate students of technological universities and colleges, as well as attract the attention of young scientists in solving global problems of energy and resource conservation in the modern world, the search for alternative energy sources.

1. Analytical apparatus of non-equilibrium processes of transformation of energy and matter

1.1 Method of dissipative functions

Any natural process that manifests itself in nature or proceeds in technical devices at a finite rate is non-equilibrium and irreversible. A non-equilibrium change in the state of a macrobody cannot be described using classical thermodynamics. The application of the equation of state is possible only for the equilibrium flow of the process, when the system can be characterized by certain intensive parameters. The imbalance leads to inhomogeneous distribution of temperature, pressure, concentration of ingredients of a multiphase and multicomponent system, as well as the existence of relaxation material and energy flows. Relaxation processes tend to equalize the uneven distribution of parameters, compensating for the external influence that causes the system's inhomogeneity. The non-equilibrium flow of phenomena associated with the transformation of energy and matter leads to states when the system no longer has a single temperature or density, the same on its mass. To describe the state of a thermodynamic system participating in a nonequilibrium process, more detailed characteristics of the state change are required. Therefore, non-equilibrium processes cannot be represented on a thermodynamic diagram in the coordinates *PV, NS, TS*, as it is traditionally accepted for equilibrium changes in the object under study. Quantitative judgments about a real process are possible only if the system at the beginning and at the end of the process is in certain equilibrium states.

It has been established that all real processes are irreversible and can spontaneously proceed in only one direction. This principle of irreversibility applies to all real processes without exception, both natural and technological. The irreversibility of a real process is manifested in the fact that its course is always accompanied by residual effects. The consequences of a real process cannot be completely eliminated, which is confirmed by numerous experiments. Not a single fact has yet been verified that refutes this distinctive feature of the real process, which, apparently, is associated with the peculiarities of energy transformations. The course of the real process consists in a special transformation of energy in the macrobody. The total energy of an isolated thermodynamic system is conserved only quantitatively, while in a qualitative sense, energy is constantly depreciating, i.e. dissipates. Dissipation does not mean the complete dissipation of energy in space, but the loss of its valuable, convertible part, called exergy. All types of energy, including internal, are limited by convertibility and consist of two parts: a convertible, called exergy, and a ballast part, which is associated with entropy:

$$H = G + TS \tag{1.1}$$

The total energy of the system *H* (enthalpy) is the sum of its two terms: the

Gibbs energy *G* (convertible part) and the ballast part *TS* associated with entropy.

Any real process is accompanied by the loss of some part of the convertible energy, i.e. exergy. Only an ideal process allows you to fully preserve the entire supply of energy in a quantitative and qualitative sense, therefore it is characterized by energy perfection, i.e. no losses during energy conversions. Indicators of an ideal process can serve as initial information for assessing the quality and efficiency of technical devices, which is one of the main tasks of energy technology.

The quantitative assessment of the principle of irreversibility is expressed by the second law of thermodynamics, by introducing an increase in entropy as a measure of dissipation. The entropy of an isolated or closed thermodynamic system, when any real process takes place in it, will certainly increase, thus any real process generates an increase in entropy.

The change in entropy in a thermodynamic system during a real process in it is the sum of two terms:

$$\frac{dS}{d\tau} = \frac{e(dS)}{d\tau} + \frac{in(dS)}{d\tau}, \tag{1.2}$$

where $\frac{e(dS)}{d\tau} = e\dot{S}$ – the first term that characterizes the change in entropy due to the equilibrium transfer of entropy along with the fluxes of heat and matter through the boundary of the thermodynamic system.

$$\frac{e(dS)}{d\tau} = \frac{e\left(dS^{\text{диф}}\right)}{d\tau} + \frac{e\left(dS^{\text{т/о}}\right)}{d\tau}, \tag{1.3}$$

where $\frac{e\left(dS^{\text{диф}}\right)}{d\tau} = e\dot{S}^{\text{диф}}$ – the flow of entropy carried with the flow of matter;

$\frac{e\left(dS^{\text{т/о}}\right)}{d\tau} = e\dot{S}^{\text{т/о}}$ – the flow of entropy carried with the flow of heat.

The second term in equation (1.2) is the rate of entropy production within the process itself due to its irreversibility. This flow of entropy cannot be transferred through the boundaries of the thermodynamic system. The entropy generated within a real process can act as a quantitative measure of its irreversibility. For this, the concept of a local dissipative function is introduced, which is related to the rate of entropy production in a unit volume by the following relation:

$$\dot{\Psi}^V = TI_S^V, \tag{1.4}$$

where I_S^V – local flow of entropy generated within the process.

$$I_S^V = \lim_{V \to 0} \frac{in\dot{S}}{V}, \tag{1.5}$$

where $in\dot{S}$ – entropy production rate within the process;

$\dot{\psi}^V$ – local dissipative function, i.e. the value of the dissipative function in a unit volume contracted to a point:

$$\dot{\Psi}^V = \lim_{V \to 0} \frac{\dot{\Psi}}{V}. \tag{1.6}$$

According to the second law of thermodynamics of irreversible processes, the rate of entropy production within the process itself is strictly positive:

$$in\dot{S} \geq 0. \tag{1.7}$$

The entropy generated within a real process is a quantitative measure of its irreversibility.

To calculate the dissipative function, two calculation methods are used.

The first method is based on the integration of the local dissipative function over the entire volume of the thermodynamic system. The second method is to use the integral equations of energy and matter balance for a fixed control volume of a nonequilibrium thermodynamic system.

1.2 Quantification of irreversible processes based on the concept of local dissipation

Engineering applications of irreversible processes are very diverse. The most complex are the transformations of the system, accompanied by changes in the chemical composition. A comprehensive description of the phenomena accompanying irreversible changes in the state of a macroobject begins with the combination of the laws of conservation of mass and energy with the corresponding kinetic laws that describe the mechanisms of these phenomena. Local changes in the rates of heat transfer, a component of matter and momentum, near equilibrium are characterized by linear relationships between the rate and the driving force of the process:

$$\vec{q} = -\lambda \cdot \vec{\nabla} T \text{ – Fourier's law} \tag{1.8}$$

where: $\vec{q}$ – heat flux density, λ – coefficient of thermal conductivity of the medium.

$$J_j^{\text{дф}} = -D_j \vec{\nabla} C_j \text{ – Fick's law} \tag{1.9}$$

where: $J_j^{\text{дф}}$ – diffusion flux density of the j-th component, D_j – diffusion coefficient of the j-th component.

$$\sigma_{ij} = -\mu \left(\frac{\partial \vartheta_i}{\partial x_j} + \frac{\partial \vartheta_j}{\partial x_i} \right) \text{ – Newton's law of outflow of a viscous medium} \tag{1.10}$$

where: μ – dynamic viscosity coefficient, σ_{ij} – stress tensor in a moving medium.

$J_{jr}^V = L_{rr} A_r$ – the rate of the resulting process, accompanied by a change in the chemical composition equal to the difference between the rates of the forward and reverse reactions (1.11)

(1.11)

where: L_{rr} – phenomenological reaction coefficient associated with the forward and backward reaction constants. In particular, we have for the reaction of ideal gases

$$L_{rr} = \left(K^{+} \prod_i C_i^{V_i}\right)(RT)^{-1} \tag{1.11a}$$

where: K^{+} – direct reaction constant, C_i, V_i – concentrations and stoichiometric coefficients of initial reagents.

$A_r = -\sum_{k=1}^{n} \nu_{kr} \cdot \mu_k$ - chemical affinity, reaction driving force (1.11b)

where: ν_{kr} – stoichiometric coefficients of substance k in the considered reaction r, μ_k – chemical potential of all components.

In general, the local rate of the i-th process of energy and mass transfer J_i near equilibrium can be represented as a linear relationship:

$$J_i = \sum_{k=1}^{n} L_{ik} \cdot X_k \tag{1.12}$$

where J_i – the local speed of the i-th process, which generally depends on all driving forces X_k; L_{ik} – are phenomenological proportionality coefficients, which can be functions of the state of the object, but do not depend on the driving forces X_k.

For non-conjugate processes, the speed depends only on its own driving force X_i:

$$J_i = L_{ii} \cdot X_i \tag{1.13}$$

where L_{ii} – phenomenological coefficients of proportionality, which are related to the thermal conductivity coefficients λ, diffusion coefficient D_j of the j-th component, the viscosity of the medium μ, and the rate constants of the forward and backward reactions k^{+} and k^{-}.

In equations (1.8 - 1.11), the values of J_i are identical to the heat flux density $\vec{q}$, diffusion flux $J_j^{дф}$, momentum flux σ_{ij}, chemical reaction rate J_{jr}^{V} in a unit volume of the reaction mixture.

The magnitude of the driving force X_i is proportional to the temperature gradients $\vec{\nabla}T$, the concentration of the j-th component $\vec{\nabla}C_j$, the strain rate of the medium in two-dimensional space $X_{ij} = -\frac{1}{2}\left(\frac{\partial \upsilon_i}{\partial x_j} + \frac{\partial \upsilon_j}{\partial x_i}\right)$, chemical affinity A_r.

In an equilibrium system, $J_i = 0$ and $X_i = 0$, that is, there is a uniform distribution of intense characteristics in the absence of external fields and, as a consequence, the absence of relaxation flows. The irreversibility of real processes in the macrobody means the fundamental impossibility of returning all the bodies involved in the process to their original state, even if the return process is equilibrium. For example, in a compressor, the temperature and pressure of the gas increase due to an external source (electrical network). If the compression is in equilibrium, then the process is certainly thermodynamically reversible, that is, it is possible to return all bodies to their original state: gas and transfer work to an external source in the same amount. If the process is non-equilibrium and is accompanied by friction ($\sigma_{ij} \neq 0$) and heat transfer ($\vec{q} \neq 0$), then, basically, it is possible to return the gas to its original state. However, the amount of work returned to an external source will be noticeably less than the original cost. The cause of irreversibility is the dissipation (depreciation) of energy, i.e. transformation of its convertible part into a ballast part due to entropy. The second law of thermodynamics for equilibrium processes introduces entropy as a function of state, which makes it possible to isolate its convertible part from the internal energy U: the Gibbs-Helmholtz functions $A = U - TS$ and Gibbs $G = H - TS$. In non-equilibrium processes, an internal flow arises (entropy production) $in\dot{S} \geq 0$, which acts as a quantitative measure of energy dissipation.

For processes of flow of a viscous medium, the local dissipative function (the Rayleigh function) $\dot{\Psi}^V_{\text{тр}}$ is equal to the product of the momentum flux σ_{ij} in the velocity profile plane of the moving medium and the cause - the strain rate. Taking into account the tensor nature of the momentum, we obtain (1.14):

$$\dot{\Psi}^V_{\text{тр}} = -(\sigma \cdot \vec{\nabla}) \cdot \vec{\upsilon} = \sum_{i=1}^{n=3} \sum_{j=1}^{m=3} \sigma_{ij} \left[-\frac{1}{2}\left(\frac{\partial \upsilon_i}{\partial x_j} + \frac{\partial \upsilon_j}{\partial x_i} \right) \right], \tag{1.14}$$

where: $X_{ij} = -\frac{1}{2}\left(\frac{\partial \upsilon_i}{\partial x_j} + \frac{\partial \upsilon_j}{\partial x_i} \right)$.

The dissipation value $\dot{\Psi}$ for the thermodynamic system as a whole can be determined by integration, i.e. distribution of the local dissipative function $\dot{\Psi}^V$ on the entire volume V under consideration (1.15):

$$\dot{\Psi} = \int_0^V \dot{\Psi}^V dV = \int_0^V T \cdot J_S^V dV = \bar{T}_{\text{т/д}} \cdot {}_{in}\dot{S}, \tag{1.15}$$

where: $\bar{T}_{\text{т/д}}$ – average thermodynamic temperature, K; ${}_{in}\dot{S}$ – is the rate of entropy increment in the entire system under consideration due to internal irreversible processes, W/K.

Differential equations for the balance of mass, energy, entropy, kinetic ratios of local rates of energy and mass transfer processes and the Gibbs equation for fixed parameters make it possible to obtain a calculated ratio for the dissipation density:

$$\dot{\Psi}^V = \vec{q} \cdot \left(-\frac{\vec{\nabla} T}{T} \right) + \sum_{j=1}^{m} \overrightarrow{J_J^{\text{дф}}} \cdot \left(-\vec{\nabla}\mu \right)_{T,p} + \sum_{r=1}^{f} J_{jr}^V \cdot A_r + \left(-\sigma \cdot \vec{\nabla} \right) \cdot \vec{\upsilon} \tag{1.16}$$

So, the driving force for heat transfer is $X = -\frac{\vec{\nabla} T}{T}$; for mass transfer – $X_j = \left(-\vec{\nabla}\mu \right)_{T,p}$, where μ_j – where μ_j is the chemical potential of the j-th component of the system; for chemical transformations $X_r = A_r$, where A_r – where A_r is the chemical affinity characterizing the degree of completeness of the chemical reaction.

To calculate the local and integral values of the dissipative function, information is needed on the distribution of intensive parameters: temperature $T(x,y,z,\tau)$, presseure $P(x,y,z,\tau)$, concentration $C_j(x,y,z,\tau)$, velocity distributions $v(x,y,z,\tau)$ and kinetic relations for calculating the thermophysical properties of the system.

1.2.1 Dissipation of kinetic energy in viscous flow

Task 1. To btain the calculated ratio of the local dissipative function for the hydrodynamic steady motion of an incompressible fluid (50% aqueous solution of glycerin). To determine the value of the dissipative function over the entire volume in a smooth round pipe with an inner diameter D and length L under isothermal conditions. The motion mode is stationary, laminar $Re = 1800$. Process parameters: $P = 20$ bar, t = 40 ºC. Let's compute at $\frac{r}{R} = 1$. Let us imagine the flow profile of a medium in a round pipe of constant diameter under laminar conditions (Fig. 1).

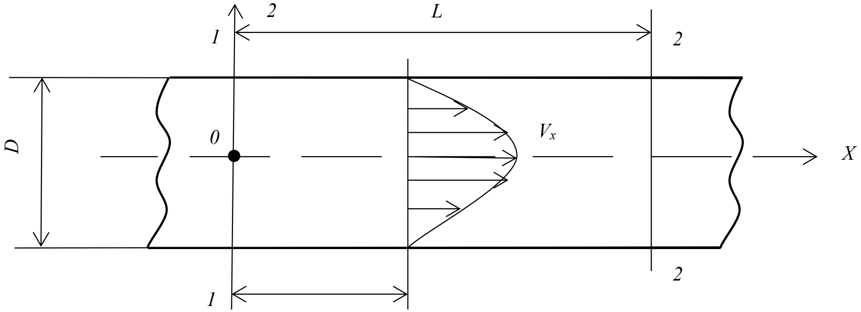


Figure 1. Medium flow profile in a round pipe of constant diameter under laminar conditions.

Under these conditions, the velocity distribution is parabolic:

$$\upsilon_X = \upsilon_0 \left[1 - \left(\frac{r}{R}\right)^2\right] \tag{1.17}$$

where: r - the current value of the flow radius, R - the inner radius of the pipe, $\upsilon_0 = 2\bar{\upsilon}$ – the speed on the pipe axis equal to the double value of the average speed.

Relation (1.17) describes the profile of velocities in the cross section of the pipe and is a parabola equation. However, it should be borne in mind that, in fact, the velocity distribution is a three-dimensional figure and, with laminar motion in a round pipe, is a paraboloid of revolution. Equation (1.17) is written for any longitudinal section of this paraboloid for a plane passing through the pipe axis. The local dissipative function is determined according to expression (1.14).

The viscous stress tensor generally has nine components (1.18).

$$\sigma_{ij} = \begin{vmatrix} \sigma_{xx} & \sigma_{xy} & \sigma_{xz} \\ \sigma_{yx} & \sigma_{yy} & \sigma_{yz} \\ \sigma_{zx} & \sigma_{zy} & \sigma_{zz} \end{vmatrix} \tag{1.18}$$

where: *i,j* = 1 – corresponds to the x-axis; *i,j* = 2 - corresponds to the y axis; *i,j* = 3 - corresponds to the z axis; σ_{ij} – component of the viscous stress tensor, N/m^2.

For an incompressible medium, the diagonal terms are equal to zero $\sigma_{xx} = \sigma_{yy} = \sigma_{zz} = 0$, in addition, taking into account the symmetry $\sigma_{ij} = \sigma_{ji}$. In the example under consideration, only one velocity component υ_x has a non-zero value. Therefore, the local dissipation of kinetic energy is equal to:

$$\dot{\Psi}^V_{\text{тр}} = 2\sigma_{xr}\left[-\frac{1}{2}\left(\frac{\partial \upsilon_x}{\partial r}\right) + \left(\frac{\partial \upsilon_r}{\partial x}\right)\right] = \mu\left(\frac{\partial \upsilon_x}{\partial r}\right)^2 \tag{1.19}$$

where: the radial velocity component for a stabilized flow is zero $\upsilon_r = 0$; viscous stress in the radial plane xr $\sigma_{xr} = -\mu\frac{\partial \upsilon_x}{\partial r}$, where μ – the dynamic viscosity of the medium.

$$\dot{\Psi}^V_{\text{тр}} = 4\mu\upsilon_0^2\frac{r^2}{R^2} = 16\mu\bar{\upsilon}^2\frac{r^2}{R^4} \tag{1.20}$$

Obviously, in the center of the pipe (r = 0) $\dot{\Psi}^V_{\text{тр}} = 0$, in turn, directly at the pipe wall (r = R) the dissipation of kinetic energy has a maximum value:

$$\dot{\Psi}^V_{\text{тр}} = 16\mu\bar{\upsilon}^2R^2 \tag{1.21}$$

Integrating $\dot{\Psi}^V_{\text{тр}}$ on the volume bounded by the inner surface of the pipe with diameter D and length L, between sections 1-1 and 2 – 2, we obtain an expression for calculating the total dissipation $\dot{\Psi}_{\text{тр}}$ under the given conditions (1.22a), (1.22b):

$$\dot{\Psi}_{\text{тр}} = \int_0^V \dot{\Psi}^V_{\text{тр}}dV = \iint_{0\,0}^{L\,R} 16\mu\bar{\upsilon}^2\frac{r^2}{R^4}2\pi r dr dx \tag{1.22a}$$

$$\dot{\Psi}_{\text{тр}} = 32\pi\mu\bar{\upsilon}^2\frac{1}{R^4}\int_0^L dx\int_0^R r^3\,dr = 8\pi\mu\bar{\upsilon}^2L \tag{1.22b}$$

The integration used the condition of an incompressible medium, which makes it possible to assume that the distribution of the flow velocity along the entire length of the pipe is unchanged. The numerical values of the parameters and thermophysical characteristics of the outflow of the solution in the pipe section included in the formula (1.22b) are calculated according to the laminar hydrodynamic regime Re=1800 in a pipe with a diameter D=0.06m and a length L=25m:

$$\bar{\upsilon} = \frac{\mu Re}{D\rho}\text{ , at } \rho = 1116\ \frac{\text{кг}}{\text{м}^3}\ [3]\ \mu = 3{,}5\cdot10^{-3}\ \text{Па}\cdot\text{с}$$

$$\bar{\upsilon} = \frac{3{,}5\cdot10^{-3}\cdot1800}{0{,}06\cdot1116} = 0{,}094\ m/s$$

$$\dot{\Psi}_{тр} = 8 \cdot 3{,}5 \cdot 10^{-3} \cdot 0{,}094^2 \cdot 3{,}14 \cdot 25 = 1{,}942 \cdot 10^{-2} \text{ W}$$

It should be noted that an analytical calculation of dissipation during the motion of a medium under conditions of a turbulent regime is impossible in this case, since there is no information on turbulent fluctuations.

1.2.2 Dissipation of internal energy in heat conduction processes

In engineering, the problems of heat transfer at a constant heat flux density of the walls are encountered in many cases: in electric heating, radiation heating, heating in nuclear reactors and in counterflow heat exchangers, when the mass flow heat capacities (the product of the mass flow rate and the heat capacity) of the heat carriers are the same. In our problem, there is also one boundary condition is a constant temperature of the outer surface of the pipe along the entire length of the reactor. Such a boundary condition is also often encountered in practice, for example, in such heat exchangers as evaporators, condensers and in all heat exchangers, when the mass flow heat capacity of one heat carrier is much greater than that of another.

The method used is based on the integration of the local dissipative function on the entire volume of the system under consideration. The advantage of this methodological approach lies in its clarity and the ability to reveal the internal logic of the derivation of the main patterns of the process. Let us analyze this case on a concrete example of the dissipation of internal energy in the process of heat conduction.

Task 2. To obtain an analytical and numerical solution for the magnitude of local and integral energy dissipation in the wall of a tubular reactor, if the value of the heat flux density on the outer surface of the pipe $q' = -60\ \frac{\text{kW}}{\text{m}^2}$, the temperature of this surface is $T_{ст} = T_{нар} = 873$ К, external diameter $d_н$=0,1 м and internal diameter $d_{вн}$=0,08 м of the wall, pipe length L = 40 m, thermal conductivity value of the wall

$$\lambda = 23{,}8\ \frac{\text{W}}{\text{m·K}}.$$

It should be noted that the relation given in the problem for the dissipative function (1.16) is the simplest linear combination of the terms of each individual gradient (temperature, concentration, velocity), although it is known that various forms of energy transfer are interconnected. In the problem under consideration, there is only a temperature gradient, therefore, according to the Fourier's law, the heat flux density is determined by equation (1.8):

$$\vec{q} = -\lambda \cdot \vec{\nabla} T,$$

where the temperature gradient in the one-dimensional (radial) problem is defined in a simplified way as:

$$\vec{\nabla}T = \overrightarrow{l_r} \cdot \frac{\partial T}{\partial r}$$

Let us depict the temperature distribution along the pipe radius for the selected boundary conditions (Fig. 2).

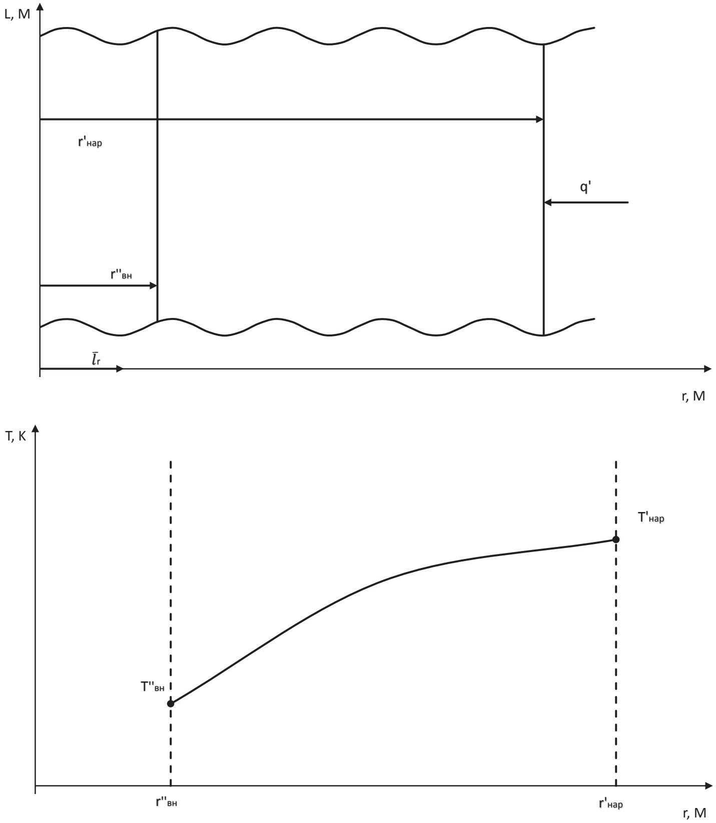


Figure 2. *Temperature distribution along the radius of the pipe under the condition of constancy q',* $T'_{\text{нар}}$ *u λ*

The local dissipative function according to equation (1.8) is equal to:

$$\dot{\Psi}^V = \vec{q} \cdot \left(-\frac{\vec{\nabla}T}{T}\right) = \frac{\lambda}{T}\left(\vec{\nabla}T\right)^2 \geq 0 \qquad (1.23)$$

For a one-dimensional problem with a boundary condition of the second kind, the temperature distribution in a cylindrical pipe of radii *r* has the form (1.24):

$$T(r) = T'_{\text{нар}} - \frac{q' \cdot r'_{\text{нар}}}{\lambda} ln \frac{r}{r'_{\text{нар}}}, \qquad (1.24)$$

where the condition $q' = -\lambda \left(\frac{\partial T}{\partial r}\right)_{r=r'_{\text{нар}}} = const$, is satisfied regardless of the medium flow regime along the entire wall surface of length *l*.

Differentiating and solving equation (1.24) with respect to $\frac{\partial T}{\partial r}$, we get (1.25):

$$\frac{\partial T}{\partial r} = -\frac{q' \cdot r'_{\text{нар}}}{\lambda} \cdot \frac{1}{r} \tag{1.25}$$

Substituting the expression $\frac{\partial T}{\partial r}$ into equation 1.23, we obtain (1.26):

$$\dot{\psi}^V = \frac{\lambda}{T}\left[-\frac{q' \cdot r'_{\text{нар}}}{\lambda} \cdot \frac{1}{r}\right]^2 \left(\overrightarrow{l_r}\right)^2, \tag{1.26}$$

where the square of the unit radiant vector is 1 $\left(\overrightarrow{l_r}\right)^2 = 1$. The final expression for local dissipation has the following form (1.27):

$$\dot{\psi}^V = \frac{\lambda}{T}\left[-\frac{q' \cdot r'_{\text{нар}}}{\lambda} \cdot \frac{1}{r}\right]^2 = \frac{\left[-q' \cdot r'_{\text{нар}}\right]^2}{T \cdot \lambda \cdot r^2} \tag{1.27}$$

We see that the resulting expression for directly determining the numerical value of the local dissipative function on the outer and inner walls of the reactor tube is very convenient. Let's do these calculations:

$$r = r'_{\text{нар}} = 0{,}05 \text{ м} \qquad T = T'_{\text{нар}} = 873 \text{ К}$$

$$\dot{\psi}^V = \frac{\left[-q' \cdot r'_{\text{нар}}\right]^2}{T'_{\text{нар}} \cdot \lambda \cdot {r'_{\text{нар}}}^2} = \frac{(q')^2}{T'_{\text{нар}} \cdot \lambda} = \frac{(-60 \cdot 10^3)^2}{873 \cdot 23{,}8} = 173{,}27 \ \frac{\text{kW}}{\text{m}^3}$$

$$r = r'_{\text{вн}} = 0{,}04 \text{ м} \qquad T = T''_{\text{вн}}$$

$$T''_{\text{вн}} = T'_{\text{нар}} - \frac{q' \cdot r'_{\text{нар}}}{\lambda} ln \frac{r''_{\text{вн}}}{r'_{\text{нар}}} = 873 - \frac{-60 \cdot 10^3 \cdot 0{,}05}{23{,}8} ln \frac{0{,}04}{0{,}05} = 844{,}88 \text{ К}$$

$$\dot{\psi}^V = \frac{(q')^2}{T'_{\text{нар}} \cdot \lambda} \cdot \left(\frac{r'_{\text{нар}}}{r''_{\text{вн}}}\right)^2 = \frac{(-60 \cdot 10^3)^2}{844{,}88 \cdot 23{,}8} \cdot \left(\frac{0{,}05}{0{,}04}\right)^2 = 279{,}72 \ \frac{\text{kW}}{\text{m}^3}$$

To determine the integral value of dissipation in the wall of a pipe with a length of $l = 1$ m, under the condition $T'_{\text{нар}} = 873$ К, $q' = -60 \ \frac{\text{кВт}}{\text{м}^2}$ and $\lambda = 23{,}8 \ \frac{W}{\text{м} \cdot \text{К}}$ we get:

$$\dot{\Psi}_l = \int_0^V \dot{\psi}^V dV = \int_{r'_{\text{нар}}}^{r''_{\text{вн}}} \frac{\left[-q' \cdot r'_{\text{нар}}\right]^2}{T \cdot \lambda \cdot r^2} 2\pi r \cdot l \cdot dr$$

$$\dot{\Psi}_l = \frac{\left[-q' \cdot r'_{\text{нар}}\right]^2}{\lambda} 2\pi l \int_{r'_{\text{нар}}}^{r''_{\text{вн}}} \frac{dr}{T \cdot r}$$

To determine the integral $\int_{r'_{\text{нар}}}^{r''_{\text{вн}}} \frac{dr}{T \cdot r}$ it is necessary to make a change of variable from (1.26):

$$\frac{dr}{r} = -\frac{\lambda}{q' \cdot r'_{\text{нар}}} dT$$

Then:

$$\dot{\Psi}_l = \frac{\left[-q' \cdot r'_{\text{нар}}\right]^2}{\lambda} 2\pi l \int_{\text{Т}'_{\text{нар}}}^{\text{Т}''_{\text{вн}}} \left(-\frac{\lambda}{T \cdot q' \cdot r'_{\text{нар}}}\right) dT$$

Dissipation per 1 meter of pipe:

$$\dot{\Psi}_l = -q' 2\pi r'_{\text{нар}} l \cdot ln \frac{\text{Т}'_{\text{нар}}}{\text{Т}''_{\text{вн}}} = -(-60 \cdot 10^3 \cdot 2 \cdot 3{,}14 \cdot 1 \cdot 0{,}05) \cdot ln \frac{873}{844{,}88} =$$

=617,84 W/m

The dissipation along the entire length of the reactor tube is:

$$\dot{\Psi} = L \cdot \dot{\Psi}_l = 617{,}84 \cdot 40 = 24713{,}6 \text{ Вт} \approx 25 \text{ kW}$$

Internal exergy losses ${}_{in}\dot{D}_L$, due to the irreversibility of heat transfer, can be determined from the following relation (1.28):

$${}_{in}\dot{D}_L = \dot{\Psi} \frac{T_{o.c.}}{\bar{T}_{\text{т/д}}}, \qquad (1.28\ )$$

where $\bar{T}_{\text{т/д}} = \frac{T'_{cm} - T''_{cm}}{ln \frac{T'_{cm}}{T''_{cm}}}$ – average thermodynamic temperature of the process.

$$\bar{T}_{\text{т/д}} = \frac{873 - 844{,}88}{ln \frac{873}{844{,}88}} = 858{,}86\, K$$

$${}_{in}\dot{D}_L = 25 \cdot \frac{298{,}15}{858{,}86} = 8{,}68 \text{ kW}$$

Both functional quantities $\dot{\Psi}$ and ${}_{in}\dot{D}_L$ are quantitative characteristics of the efficiency of an irreversible process and have the same dimension. However, an important difference is that it is the exergy losses ${}_{in}\dot{D}_L$, due to the entropy produced inside the process, that are final and, thus, can only be compensated by external energy costs.

In turn, the difference between $\dot{\Psi}$ and ${}_{in}\dot{D}_L$ amounts to that part of the energy that can still be usefully used as a secondary (internal) energy resource. With the recovery of heat generated in the wall of the cylindrical reactor, the efficiency of the heat conduction process will increase. The value of this internal energy resource is as follows:

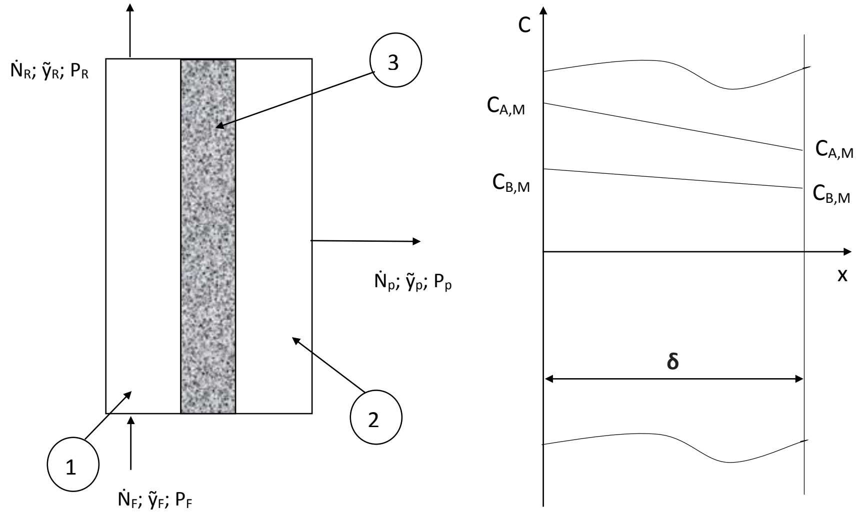
$$\dot{\Psi} - {}_{in}\dot{D}_L = 25 - 8{,}68 = 16{,}32 \text{ kW}$$

In case of qualified energy technological utilization of this hidden energy resource generated in the reactor wall, exergy losses will be minimal. The rational choice of exergy optimization of high-temperature processes depends on the characteristics of a particular energy carrier and the technological possibilities of heat recovery.

1.2.3 Energy dissipation in diffusion processes

Let us analyze the mass transfer process using a specific example of the dissipation of convertible energy in a diffusion membrane process.

Task 3. To obtain an analytical and numerical solution for the magnitude of local and integral energy dissipation in the diffusion layer of a membrane separating an ideal mixture of propane (component A) and methane (component B) hydrocarbons at a temperature T_f = 279 K and a pressure P_f = 4,6 bar in pressure channel of the device. The composition of the gas mixture in the pressure channel $\widetilde{y_A}' = 0{,}59\ \frac{\text{kmol A}}{\text{kmol of mixture}}$, in the drainage cavity $\widetilde{y_A}'' = \widetilde{y_P} = 0{,}95\ \frac{\text{kmol A}}{\text{kmol of mixture}}$. The thickness of the diffusion layer of the polymer membrane δ = 0,2 мкм, the working area A = 117 m^2; pressure in the drainage cavity P'' = 1 bar. The performance of the initial mixture is $\dot{N}_F = 0{,}023\ \frac{\text{kmol}}{\text{s}}$. The gas mixture of the initial composition $\widetilde{y_F}$ is introduced into the pressure channel 1 of the membrane apparatus at $P = P_F$ and T_F, passed through the membrane *3* flow $\dot{N}_p$ with the concentration $\widetilde{y_A}'' = \widetilde{y_P}$ is removed from the drainage cavity 2; the waste flow $\dot{N}_R$ is discharged from the pressure channel with the concentration $\widetilde{y_R} = \widetilde{y_{A0}}'$



(a) (b)

Figure 3. *a) Distribution and characteristics of flows in the apparatus:*

1 - pressure channel; 2 - drainage channel; 3 - silicone membrane.

b) Concentration profile of the components in the membrane layer of the apparatus

The process is stationary and isothermal, the gas mixture is ideal. Hydraulic resistance in the pressure and drainage channel is negligible. External diffusion resistance in the pressure and drainage channel is excluded. The structure of the gas flow in the cavity of the pressure channel *1* corresponds to the ideal mixing model (IMS). Based on these assumptions, the following values follow:

$P' = P_f = P_R$ =4,6 bar, where P' – pressure in cavity *1*;

$P'' = P_p$ = 1 bar, where P'' pressure in cavity *2*;

$$\widetilde{y_A}' = \widetilde{y_R} = 0{,}59\ \frac{\text{kmol A}}{\text{kmol of mixture}};\ \widetilde{y_A}'' = \widetilde{y_R} = 0{,}95\ \frac{\text{kmol A}}{\text{kmol of mixture}},$$

where $\widetilde{y_A}'$, $\widetilde{y_A}''$ – are the compositions of the gas phase near the membrane surfaces from the side of the pressure and drainage channel. The resistance to mass transfer is concentrated exclusively in the diffusion layer of the silicone membrane with a thickness of δ. At the boundary of the gas and the membrane, it is permissible to assume that there is a local equilibrium.

The concentrations of components A ($C_{A,m}'$) and B ($C_{B,m}'$) in the membrane at the boundary with the pressure channel are as follows:

$$\text{C}_{A,M}' = \sigma_{A,M} \cdot P' \cdot \widetilde{y_A}', \frac{\text{kmol A}}{\text{m}^3}$$

$$\text{C}_{B,M}' = \sigma_{B,M} \cdot P' \cdot \widetilde{y_\text{B}}', \frac{\text{kmol B}}{\text{m}^3},$$

where $\sigma_{A,M} = 8{,}825 \cdot 10^{-5}\ \frac{\text{mol}}{\text{m}^3 \cdot \text{Pa}}$ and $\sigma_{B,M} = 1{,}683 \cdot 10^{-6}\ \frac{\text{mol}}{\text{m}^3 \cdot \text{Pa}}$ – the values of the solubility coefficients of this membrane (material polydimethylsiloxane $[(CH_3)_2SiO]_x$) on both components at a temperature value of $T = 279$ K.

We obtain the following values of the concentrations of the components at the boundaries with the membrane:

$$\text{C}_{A,M}' = 8{,}825 \cdot 10^{-8}\ \cdot 4{,}6 \cdot 10^5 \cdot 0{,}59 = 23{,}951 \cdot 10^{-3}\ \frac{\text{kmol}}{\text{m}^3}$$

$$\text{C}_{A,M}'' = 8{,}825 \cdot 10^{-8}\ \cdot 1 \cdot 10^5 \cdot 0{,}95 = 8{,}3837 \cdot 10^{-3}\ \frac{\text{kmol}}{\text{m}^3}$$

$$\text{C}_{B,M}' = 1{,}683 \cdot 10^{-9}\ \cdot 4{,}6 \cdot 10^5 \cdot 0{,}41 = 0{,}3174 \cdot 10^{-3}\ \frac{\text{kmol}}{\text{m}^3}$$

$$\text{C}_{B,M}'' = 1{,}683 \cdot 10^{-9}\ \cdot 1 \cdot 10^5 \cdot 0{,}05 = 0{,}00841 \cdot 10^{-3}\ \frac{\text{kmol}}{\text{m}^3}$$

The values of diffusion fluxes J_A, J_B in the membrane are determined based on the condition of constant diffusion coefficients of the components in the membrane layer. For a flat and one-dimensional problem, the distribution of the substance concentration in the membrane is linear, and the concentration gradients are constant (1.9).

$$\frac{\partial \mathrm{C}_{A,m}}{\partial x}=\frac{\mathrm{C}_{A,m}{}''-\mathrm{C}_{A,m}{}'}{\delta}=\frac{(8{,}3837-23{,}951)\cdot 10^{-3}}{2\cdot 10^{-7}}$$

$$\frac{\partial \mathrm{C}_{A,m}}{\partial x}=-7{,}784\cdot 10^{4}\,\frac{\mathrm{kmol}}{\mathrm{m}^{4}}$$

$$\frac{\partial \mathrm{C}_{B,m}}{\partial x}=\frac{\mathrm{C}_{B,m}{}''-\mathrm{C}_{B,m}{}'}{\delta}=\frac{(0{,}00841-0{,}3174)\cdot 10^{-3}}{2\cdot 10^{-7}}$$

$$\frac{\partial \mathrm{C}_{B,m}}{\partial x}=-0{,}1545\cdot 10^{4}\,\frac{\mathrm{kmol}}{\mathrm{m}^{4}}$$

Diffusion flux density values on components A and B:

$$J_A=-D_{A,m}\frac{\partial C_{A,m}}{\partial x}=-5{,}558\cdot 10^{-10}\cdot(-7{,}784\cdot 10^{4})$$

$$J_A=4{,}326\cdot 10^{-5}\,\frac{\mathrm{kmol}}{\mathrm{m}^{2}\cdot\mathrm{s}}$$

$$J_{\mathrm{B}}=-D_{B,m}\frac{\partial C_{B,m}}{\partial x}=-1{,}433\cdot 10^{-9}\cdot(-0{,}1545\cdot 10^{4})$$

$$J_B=0{,}2214\cdot 10^{-5}\,\frac{\mathrm{kmol}}{\mathrm{m}^{2}\cdot\mathrm{s}}$$

The total specific flux (total density of the substance) that has penetrated through the separating membrane is:

$$J=J_A+J_B=4{,}326\cdot 10^{-5}+0{,}2214\cdot 10^{-5}$$

$$J=4{,}5474\cdot 10^{-5}\,\frac{\mathrm{kmol}}{\mathrm{m}^{2}\cdot\mathrm{s}}$$

The total flow, taking into account the active surface of the membrane, is:

$$\dot{N}_p=\int_0^A J\,dA=J\cdot A=4{,}5474\cdot 10^{-5}\cdot 117=0{,}532\cdot 10^{-2}\,\frac{\mathrm{kmol}}{\mathrm{s}}$$

Waste flow, according to the material balance equation:

$$\dot{N}_R=\dot{N}_F-\dot{N}_p=0{,}023-0{,}00532=0{,}01768\,\frac{\mathrm{kmol}}{\mathrm{s}}$$

Let's check the composition of the flow that penetrated through the diffusion layer of the membrane:

$$\widetilde{y_A}''=\frac{J_A}{J}=0{,}95\,\frac{\mathrm{kmol\ A}}{\mathrm{kmol\ of\ mixture}}$$

Initial composition:

$$\widetilde{y_F}=\frac{\dot{N}_R}{\dot{N}_F}\widetilde{y_R}+\frac{\dot{N}_P}{\dot{N}_F}\widetilde{y_P}=\frac{0{,}01768}{0{,}023}\cdot 0{,}59+\frac{0{,}00523}{0{,}023}\cdot 0{,}95=0{,}673\,\frac{\mathrm{kmol\ A}}{\mathrm{kmol\ of\ mixture}}$$

The local dissipative function in the isothermal membrane diffusion process is determined according to relation (1.16):

$$\dot{\Psi}^V = \overrightarrow{J_A} \cdot \left(-\Delta\overrightarrow{\mu_{A,M}}\right) + \overrightarrow{J_B} \cdot \left(-\Delta\overrightarrow{\mu_{B,M}}\right)$$

For ideal solutions of the component in the membrane (the value of the activity coefficient γ is equal to unity), under the conditions of the flat geometry of the membrane, we have: $\frac{\partial \mu_A}{\partial x} = \frac{\partial \mu_A}{\partial c_{A,M}} \cdot \frac{\partial c_{A,M}}{\partial x}$, where, taking into account the assumptions: $\frac{\partial \mu_A}{\partial c_{A,M}} = \frac{\partial \ln(\gamma \cdot c_{A,M})}{\partial c_{A,M}} = \frac{RT}{c_{A,M}}$

Let us present the calculated relation for local dissipation and calculate its value for the given boundary conditions ($x = 0$, $x = \delta$):

$$\dot{\Psi}^V = RT\left[D_{A,M}\left(\frac{\partial c_{A,M}}{\partial x}\right)^2 \frac{1}{c_{A,M}} + D_{B,M}\left(\frac{\partial c_{B,M}}{\partial x}\right)^2 \frac{1}{c_{B,M}}\right]$$

$$\dot{\Psi}^V_{x=0} = 8{,}314 \cdot 279 \cdot [5{,}558 \cdot 10^{-10} \cdot (-7{,}784 \cdot 10^4)^2 \cdot \frac{1}{23{,}951 \cdot 10^{-3}} +$$

$$+1{,}433 \cdot 10^{-9} \cdot (-0{,}1545 \cdot 10^4)^2 \cdot \frac{1}{0{,}3174 \cdot 10^{-3}}] = 3{,}51 \cdot 10^5\ \frac{\text{kW}}{\text{m}^3}$$

$$\dot{\Psi}^V_{x=\delta} = 8{,}314 \cdot 279 \cdot [5{,}558 \cdot 10^{-10} \cdot (-7{,}784 \cdot 10^4)^2 \cdot \frac{1}{8{,}3837 \cdot 10^{-3}} +$$

$$+1{,}433 \cdot 10^{-9} \cdot (-0{,}1545 \cdot 10^4)^2 \cdot \frac{1}{0{,}00841 \cdot 10^{-3}}] = 18{,}7525 \cdot 10^5\ \frac{\text{kW}}{\text{m}^3}$$

The local dissipation of the convertible Gibbs energy is proportional to the square of the driving force and inversely proportional to the local concentration of the components; therefore, the largest value of $\dot{\Psi}^V$ corresponds to the membrane region adjacent to the drainage cavity.

The integral value of the dissipative function, referred to the unit area of the membrane, is determined by integrating its local value along the coordinate x:

$$\dot{\Psi}_{\delta=1\text{м}^2} = \int_0^\delta \dot{\Psi}^V dx$$

For planar geometry, the linear distribution of the concentration of components in the membrane obeys the expression:

$$c_{j,M} = c_{J,M}\,(x = 0) + bx_J, \text{где } b = \frac{\partial c_j}{\partial x} = const.$$

Taking into account this assumption, we have:

$$\dot{\Psi}_{\delta=1\text{м}^2} = \int_0^\delta \dot{\Psi}^V dx = RT\left(\sum_{j=1}^{2}\left\{\int_{c'_{j,M}}^{c''_{j,M}} D_{j,M}\cdot\left(\frac{\partial c_{j,M}}{\partial x}\right)\frac{dc_{j,M}}{c_{j,M}}\right\}\right) =$$

$$= RT\left(\sum_{j=1}^{2}\left\{D_{j,M}\cdot\left(\frac{\partial c_{j,M}}{\partial x}\right) ln\frac{c''_{j,M}}{c'_{j,M}}\right\}\right)$$

$$\dot{\Psi}_{\delta=1\text{м}^2} = RT\left(\sum_{j=1}^{2}\left\{(-J_j) ln\frac{c''_{j,M}}{c'_{j,M}}\right\}\right)$$

In the conditions of our task, we assume that $j = 1$ (component A), $j = 2$ (component B):

$$\dot{\Psi}_{\delta=1\text{м}^2} = RT\left[J_A\left(-ln\frac{c''_{A,M}}{c'_{A,M}}\right) + J_B\left(-ln\frac{c''_{B,M}}{c'_{B,M}}\right)\right]$$

$$\dot{\Psi}_{\delta=1\text{м}^2} = 8{,}314\cdot 279\left[4{,}325\cdot 10^{-5}\left(-ln\frac{8{,}3837\cdot 10^{-3}}{23{,}951\cdot 10^{-3}}\right) +\right.$$

$$\left. +0{,}2214\cdot 10^{-5}\left(-ln\frac{0{,}00841\cdot 10^{-3}}{0{,}3174\cdot 10^{-3}}\right)\right] =$$

$$= 0{,}1053 + 0{,}00186 = 0{,}1239\ \frac{\text{кВт}}{\text{м}^2} > 0$$

The integral value of the dissipation of convertible energy when passing through a membrane with a contact surface area $A = 117$ м^2 is determined by integrating $\dot{\Psi}_{\delta=1\text{м}^2}$ over the entire working surface:

$$\dot{\Psi}^V = \int_0^A \dot{\Psi}_{\delta=1\text{м}^2} dA = 0{,}1239\cdot 117 = 14{,}5\ \text{kW}$$

The task is devoted to diffusion in a binary gaseous system obeying Fick's law: the intensity of mass transfer of an individual component is proportional to its concentration gradient and does not depend on the potentials of other intensive properties of the system. Often such restrictions in applied matters are justified, despite the fact that engineering applications of diffuse phenomena, in particular in gaseous systems, are very diverse, and therefore require consideration of the inseparable connection between the mechanisms of heat and mass transfer convection in each individual case.

2. Thermodynamic perfection evaluation of the energy and matter transformation processes on the basis of the concept of exergy

2.1 The method of energy balances in the analysis of an irreversible process

The second method for estimating the dissipative function is based on the use of a system of integral equations for a fixed control volume of a thermodynamic system (FTS). The configuration of the FTS boundaries is unchanged, but the boundaries themselves are permeable for convective and non-convective flows of mass and energy. These equations relate the rate of change of extensive quantities (total mass, mass of an individual k-th component, total energy, kinetic and potential energy, entropy, exergy) with the causes causing these changes: flows carried across FTS boundaries and internal sources, if any.

Gross mass balance equation:

$$\frac{\partial}{\partial \tau}\int_0^V \rho_V dV = \sum_{i=1}^{n} \dot{m}_i \qquad (2.1)$$

where $\rho_V = \lim\limits_{V\to 0}\frac{m}{V}$ – total mass density, kg/m^3; $\dot{m}_i$ – convective flow of the total mass through the permeable sections of the FTS (sections i = 1.2 ... n), kg/s.

Total energy balance equation:

$$\frac{\partial}{\partial \tau}\int_0^V (K_V + \Pi_V + U_V) dV = \sum_{i=1}^{n} \dot{m}_i \left(\bar{h} + \frac{\bar{\upsilon}^2}{2} + \bar{\varphi}_g\right)_i - \dot{W}^{tex} + \dot{Q} + \dot{H}^{\text{диф}} \qquad (2.2)$$

where $K_V = \lim\limits_{V\to 0}\frac{K}{V}$ – kinetic energy density, J/m^3;

$\Pi_V = \lim\limits_{V\to 0}\frac{\Pi}{V}$ – potential energy density, J/m^3;

$U_V = \lim\limits_{V\to 0}\frac{U}{V}$ – internal energy density, J/m^3;

h_i, $\bar{\upsilon}_i$, $\bar{\varphi}_{g_i} = g\bar{z}_i$ – mass average values of enthalpy, velocity, specific gravitational potential, J/kg;

$\dot{W}^{tex}$ – flow of external work in the form of kinetic or electrical energy, W;

$\dot{Q}$ – convective heat flux, W;

$\dot{H}^{\text{диф}}$ – enthalpy flux supplied due to the mass transfer process at the boundary of the system and the external environment, W.

The calculation of the enthalpy flow is carried out according to the ratio:

$$\dot{H}^{\text{диф}} = -\sum_{k=1}^{m}\int_0^F \bar{H}_k \vec{I}_k^{\text{нк}} \vec{n} dF , \qquad (2.3)$$

where $\overline{H}_k$ – partial value of the enthalpy of the k component, determined by local values of temperature, pressure, composition at the boundary, J/mol;

$\bar{I}_k^{нк} = \lim\limits_{F\to 0}\frac{\dot{N}_k^{нк}}{F}$ – density of the mass flux of k component (k = 1, 2 … m) $\dot{N}_k^{нк}$ through the boundaries of the FTS, determined by the conditions of the mass transfer process at the boundary, (mole of component k)/(m^2·s);

$\vec{n}$ – normal to the surface;

F – surface area, m^2.

Balance equation of kinetic and potential energy:

$$\frac{\partial}{\partial\tau}\int\limits_0^V (K_V+\Pi_V)dV = \sum_{i=1}^{n}\dot{m}_i\left(\frac{\upsilon^2}{2}+\varphi_g\right)_i - \dot{W}^{tex} + \int\limits_0^V -\vec{\upsilon}\cdot(\overrightarrow{grad}\ \ P)dV - \dot{\Psi}^{тр} \quad (2.4)$$

where $\int\limits_0^V -\vec{\upsilon}\cdot(\overrightarrow{grad}\ \ P)dV \approx \int\limits_{P_1}^{P_2}\dot{m} - \frac{dP}{\rho}$ – functional that determines the source or sink of kinetic energy during one-dimensional motion of a continuous medium with a finite velocity in the field of pressure forces P, W; $\dot{\Psi}^{тр}$ – dissipation of kinetic energy, W.

Entropy balance equation:

$$\frac{\partial}{\partial\tau}\int\limits_0^V S_V dV = \sum_{i=1}^{n}\dot{m}_i\bar{S}_i + e\dot{S}^{т/о} + e\dot{S}^{диф} + in\dot{S}, \quad (2.5)$$

where $S_V = \lim\limits_{V\to 0}\frac{S}{V}$ – entropy density, J/(m^3·K);

$\bar{S}_i$ – mass-average value of entropy in the i-th section of the FTS apparatus (i = 1, 2 … n), J/(kg·K);

$e\dot{S}^{т/о}$ – entropy flux introduced together with the heat flux at equilibrium heat transfer at the boundary, W/K. It is determined by the ratio (1.4);

$e\dot{S}^{диф}$ – entropy flux introduced with the convective flow of matter in an equilibrium mass transfer process at the boundary, W/K. It is determined by relation (1.5);

$in\dot{S}$ – rate of entropy production within the FTS due to the irreversibility of the process, W/K.

Mass balance equation for each component k:

$$\frac{\partial}{\partial\tau}\int\limits_0^V \rho_k dV = \sum_{i=1}^{n}\dot{m}_i\bar{y}_{ik} + M_k\dot{N}_k^{нк} + \int\limits_0^V I_k^V dV, \quad (2.6)$$

where ρ_k – partial density of component k in the mixture (k = 1, 2 … m), mol/m^3;

$\bar{y}_{ik}$ – mass fraction of component k in the mixture (k = 1, 2 … m), (kg of k component)/kg of mixture;

M_k – molecular weight of k component, kg/kmol;

$\dot{N}_k^{нк}$ – nonconvective flow of the k component through the FTS boundary, mol/s;

I_k^V – rate of the reaction resulting in the appearance or disappearance of k component, (mol of k component)/(m^3·s).

These relations (2.1 - 2.6) connect the rate of change of extensive quantities with the causes that cause them, both external, due to the convective flows of the substance through the permeable sections of the fixed control volume of the apparatus, and internal sources, if any. The left side of equations (2.1 - 2.6) is the rate of accumulation of extensive characteristics, i.e. changes per unit time of the values of the mass of the component, total mass, entropy, total energy, kinetic and potential energy. The values of these quantities are determined by the distribution of the density of the corresponding substance over the entire volume of the thermodynamic system. On the right side of equations (2.1 - 2.6), the first terms represent the resulting flow of substance due to visible movement due to convective transfer through the open sections of the apparatus. To calculate these quantities, the mass-average values of enthalpy $\overline{h}_i$, entropy $\overline{S}_i$, velocity $\overline{\upsilon}_i$, mass fraction of the k component are used $\overline{y}_{ik}$. External flows of substance include flows of heat $\dot{Q}$, substance $\dot{H}^{диф}$, due to reversible heat and mass transfer processes at the boundary, the flow of external work, $\dot{W}^{tex}$, as well as flows of entropy $e\dot{S}^{т/о}$ and $e\dot{S}^{диф}$, and introduced together with heat and substance flows through the permeable sections of the apparatus under reversible heat and mass transfer conditions at the boundary.

To the inner sources of substance, i.e. generated within the process due to its irreversibility, include: the reaction rate, as a result of which the k component appears or disappears I_k^V; the rate of entropy production within the process; $in\dot{S}$; dissipation of kinetic energy $\dot{\Psi}^{тр}$, i.e. the rate at which kinetic energy is converted into internal energy due to overcoming frictional forces.

We illustrate the application of the method using the example of task 1 on page 9.

Task 1

Calculate the dissipation of kinetic energy during the movement of a 50% aqueous solution of glycerol in a smooth pipe with a radius R = 0.03 m and a length L = 25 m under isothermal conditions. Laminar motion mode Re = 1800. Glycerin parameters: pressure P = 20 bar, temperature t = 40 °C.

Solution:

We tackle the task using the differential equation for the balance of kinetic and potential energy (2.4) for the FST of a thermodynamic system:

$$\dot{\Psi}_{12}^{тр} = \dot{m}\int_1^2 -\frac{dP}{\rho} = \dot{m}\frac{P_1 - P_2}{\rho} = \dot{m}\frac{\Delta P_{тр}}{\rho},$$

where $\Delta P_{тр}$ can be calculated using the Hagen–Poiseuille equation, since the mode of motion is laminar.

$$\Delta P_{тр} = 32\frac{\mu \upsilon L}{d^2} = \frac{32 \cdot 10^{-3} \cdot 25}{0{,}062} = 73{,}11 \text{ Pa},$$

$$\dot{\Psi}_{12}^{тр} = \frac{\pi D^2}{4}\bar{\upsilon}\Delta P_{тр} = \frac{3{,}14 \cdot 0{,}06^2}{4} \cdot 0{,}094 \cdot 73{,}11 = 0{,}01942 \text{ W},$$

$$\dot{\Psi}_{12}^{тр} = 1{,}942 \cdot 10^{-2} \text{ Вт}.$$

Calculation of pressure loss in straight smooth pipes during isothermal flow can be carried out according to the following formula:

$$\Delta P_{тр} = \lambda \frac{L}{d_э}\frac{\bar{\upsilon}^2 \rho}{2}, \tag{2.7}$$

where λ – dimensionless coefficient of friction, in laminar motion $\lambda = \frac{64}{\text{Re}}$.

We substitute the values into the calculation formula (2.7):

$$\Delta P_{тр} = \frac{64}{1800} \cdot \frac{25}{0{,}06} \cdot \frac{0{,}094^2 \cdot 1116}{2} = 73{,}044 \text{ Pa}$$

$$\dot{\Psi}^{тр} = \frac{\pi D^2}{4}\bar{\upsilon}\Delta P_{тр} = \frac{3{,}14 \cdot 0{,}06^2}{4} \cdot 0{,}094 \cdot 73{,}044 = 1{,}94 \cdot 10^{-2} \text{ W}$$

The result shows the equivalence of both approaches to the analysis of dissipation during the motion of viscous media.

2.2 Exergy method for the analysis of a thermodynamic system

The main tasks that should be solved when analyzing the efficiency of energy consumption in chemical production are as follows:

1) a generalized assessment of material and energy resources based on classical thermodynamics;

2) analysis of the efficiency of energy and substance consumption in non-equilibrium processes;

3) search for ways to improve the efficiency of the process, i.e. development of general principles for the rational use of material and energy resources;

4) the applied part of the energy-chemical-engineering system (ECES), i.e. assessment of technological aspects of production.

Evaluation of the effectiveness of the use of matter is based on the law of conservation of matter. Below we present the mass balance equation for a real process (2.8).

$$\sum_{i=1}^{n} \nu iMi = \sum_{j=1\div a}^{m} \nu jMj + \sum_{j=1}^{a} \nu jMj = \sum_{j=1}^{m} \nu jMj \qquad (2.8)$$

где:

$\sum_{i=1}^{n} \nu iMi$ – total flow of incoming substances, including the main components and auxiliary ones. Catalysts can be an example of the additive.

$\sum_{j=1}^{m} \nu jMj$ – total flow of outgoing substances, including:

$\sum_{j=1\div a}^{m} \nu jMj$ – by-products (semi-products, waste);

$\sum_{j=1}^{a} \nu jMj$ – target products.

In chemical production, there are solid wastes, gas emissions, and liquid effluents. The processing of these flows (crushing, filtering, settling) consumes a significant amount of energy, but the cost of these costs is justified by the guarantee of the technospheric safety of the regime. Waste may be negligible, but their harmfulness, i.e. toxicity is very high. This circumstance is taken into account by environmental indicators and sanitary and epidemic standards, i.e. indicators that are directly focused on the cost of the final product.

Based on the value of the mass utilization factor η_M, the actual cost of the target product (2.8) is estimated:

$$\eta_M = 1 - \frac{\sum_{j=a+1}^{m} \nu jMj}{\sum_{i=1}^{n} \nu iMi} \qquad (2.9)$$

Energy consumption is estimated by the energy utilization factor η_E (2.10):

$$\eta_E = \frac{\sum_{j=1}^{a} v_j \widetilde{H}_j + \dot{Q}_k(\overline{T}_k) + \dot{w}}{\sum_{i=1}^{n} v_i \widetilde{H_\imath} + \dot{Q}_k(T_k > T_c) - \dot{w}} \qquad (2.10)$$

Equation (2.9) is derived from the total energy balance equation (2.2):

$$\sum_{i=1}^{n} v_i \widetilde{H}_i - \sum_{j=1}^{m} v_j \widetilde{H_J} + \Sigma \dot{Q}_k(\overline{T}_k) - \dot{w}^{Э} = 0$$

где: $\sum_{j=1}^{m} v_j \widetilde{H}_j$ – resulting convective enthalpy fluxes;

$\sum \dot{Q}_k(\overline{T}_k)$ – convective heat fluxes that have their own thermal potential;

$-\dot{w}^{Э}$ – the total flow of electricity that is supplied (removed) to the ECES.

Relationship (2.9) does not take into account the mass transfer processes occurring on the control surface with the environment, since they occur, as a rule, within the system itself, as well as the kinetic and potential (gravitational) components of energy. The contribution of these types of energies is evaluated in specific processes and devices of a given design under certain conditions (for example, a jet apparatus).

In general, energy consumption is estimated by the relation (2.11):

$$\eta_E \cong 1 - \frac{\dot{E}_{cp}^{noT}}{\dot{E}'} = \frac{\text{losses to the environment}}{\text{supplied energy input}} \qquad (2.11)$$

where: $\dot{E}_{\text{ср}}^{\text{пот}}$ – loss of energy (heat, matter) to the environment due to poor tightness (thermal insulation), which go away with waste (gas emissions, liquid effluents, solid waste);

$\dot{E}'$ - flows of energy that are supplied to the system.

Evaluation of energy efficiency, i.e. a qualitative indicator of energy consumption is carried out using the Gibbs component of energy, i.e. exergy.

In general terms, the integral exergy balance equation is as follows (2.12):

$$\dot{E}'_x = \dot{E}''_{x\,\text{целевые}} + ex\dot{D} + in\dot{D} \tag{2.12}$$

where:

$\dot{E}'_x$ – incoming exergy flows; $\dot{E}''_{x\,\text{целевые}}$ - target exergy flows;

$ex\dot{D}$ – external exergy losses that go into the environment. These losses are due to the waste with which they take place;

$in\dot{D}$ – internal exergy losses, which are caused by inS˙_12 (the flow of entropy produced within the process itself due to irreversibility.

The model of the analytical apparatus with indication of exergy flows is represented by in fig. 4.

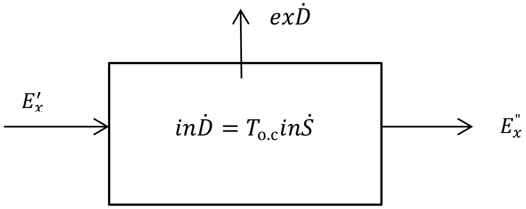


Figure 4. *Functional diagram of a real energy converter*

We introduce the concept of exergy efficiency of η_{ex}, i.e. target efficiency in terms of the Gibbs energy (2.13):

$$\eta_{ex} = \frac{\text{goal}}{\text{costs}} = \frac{\dot{E}''_{x\,\text{goal}}}{\dot{E}'^{\,\text{costs}}_{x}} = 1 - \frac{ex\dot{D} + in\dot{D}}{\dot{E}_x'^{\,\text{costs}}} \tag{2.13}$$

Attention should be paid to the expediency of adding an exergy balance and determining the value of the exergy efficiency of the entire installation, as well as the influence on its value of the contribution of its individual links. Such an analytical approach makes it possible to determine not only quantitative indicators of the consumption of material and energy resources, but also to compare and evaluate the efficiency of converting various types of energy.

The technological process should be divided into stages, since it is important to obtain the values of both the overall efficiency and the efficiency, as well as for each individual stage, so that these quantities (2.14) can be analyzed.

$$\eta_{ex\,\text{whole installation}} = 1 - \frac{\sum_{i=1}^{f} \dot{D}_{i\,\text{whole installation}}}{E'_{x\,\text{whole installation}}} \tag{2.14}$$

Estimation of the total loss of exergy of the i-th section is determined by the Gouy-Stodola formula

$$\text{in}\dot{D} = T_{\text{o.c.}}\,\text{in}\dot{S}_{12}$$

$$\frac{\dot{D}_i}{E'_x} = \frac{\dot{D}_i \cdot \dot{E}'_{x_i}}{E'_x \cdot \dot{E}'_{x_i}} = \left(1 - \eta_{ex_i}\right) \cdot d_i \qquad (2.15)$$

where:

$a_i = \frac{\dot{E}'_{x_i}}{E'_x} = d_i$ – share of exergy introduced in the i-th stage from the total exergy flow introduced into the entire installation as a whole;

$\frac{\dot{D}_i}{\dot{E}'_{x_i}} = \left(1 - \eta_{ex_i}\right)$, where η_{ex_i}- the exergy efficiency of i-th stage

Let's analyze the expression in comparison with the previously obtained formula:

$$\eta_{ex\,уст-ки} = 1 - \frac{\sum_{i=1}^{f} \dot{D}_i}{\dot{E}'_{x\,уст-ки}} = 1 - \sum_{i=1}^{f} \left(1 - \eta_{ex_i}\right) d_i \qquad (2.16)$$

where:

$\dot{D}_{уст-ки} = \sum_{i=1}^{f} \dot{D}_i$ - losses of the entire installation as a whole equal to the sum of losses in all stages (with i=1…до f).

We have presented a formula for dividing the installation by space, and perhaps division by stages, i.e. different processes occurring in the same space. There are also variable parameters (eg reflux ratio). If there are many such parameters, then multiparameter optimization is carried out.

External exergy losses $\text{ex}\dot{D}$ – due to the loss of matter and energy into the environment (imperfection of thermal insulation, waste, effluents, emissions). For the final choice of the technology mode, the economic side of the problem, environmental indicators and hardware design are taken into account. Technospheric security is a unifying problem for all civilized countries and excites the minds of both world-famous scientists and young scientists.

3. Engineering applications of the method of dissipative functions

3.1 Analysis of the energy perfection of non-ideal gas compression processes

Compressors are machines designed to increase the pressure of a gas flow. The whole variety of existing compressors can be divided into two broad classes: positive displacement and blade (dynamic) type machines. Volumetric compressors include reciprocating, rotary, membrane compressors. In dynamic compressors, due to the supply of mechanical energy, some kinetic energy is imparted to the gas, which is largely converted into pressure energy. The main varieties of this class are: centrifugal, axial compressors.

When studying the real process of gas compression, the following tasks should be solved:

1. Determination of actual energy costs for real, i.e. irreversible process.
2. Determination of the relative efficiency of the process, i.e. calculation of efficiency (efficiency factor).
3. Evaluation of the energy perfection of the compressor unit for energy and resource saving.

Equilibrium processes do not take into account the loss of kinetic energy due to friction and correspond to the minimum energy costs. To estimate the actual power of a non-ideal compressor, the values of the indicator efficiency are used, which are the results of bench tests of the compressor. The values of the indicator efficiency depend on the degree of pressure increase and are the passport characteristic of the compressor.

The indicator efficiency is the ratio of the reference power corresponding to the ideal process to the value of the actual power consumed in the real process:

$$\eta_i = \frac{\dot{W}_{эт}}{\dot{W}_{12_{вн}}}, \tag{3.1}$$

where η_i – indicator efficiency; $\dot{W}_{эт}$ – reference power of an ideal, equilibrium process, W; $\dot{W}_{12_{вн}}$ – internal power of the compressor consumed in the real process, W.

Let us perform a thermodynamic analysis of the non-equilibrium process of compression of non-ideal gases in a single-stage compressor using specific examples.

Task 3.1

Ammonia is compressed in the adiabatic stage of an uncooled compressor. Ammonia parameters at the compressor inlet:

T_1 = 306 K, P_1 = 1 bar, gas pressure at the outlet of the compressor stage P_2 = 5 bar.

Gas consumption $\dot{m}$ = 1 kg/s, non-equilibrium compression process. The calculation of the characteristics and functions of the state of the gas is carried out according to the Bogolyubov-Mayer virial equation in a truncated form [1].

Define:

1. Internal power consumed by the uncooled compressor stage $\dot{W}_{12_{вн}}$, kW.
2. Gas temperature at the outlet of the compressor stage T_2, K.
3. Exergy efficiency of the compressor unit, $\eta_{ex_{к.у.}}$

Let us present a schematic diagram of a compressor unit (Fig. 5).

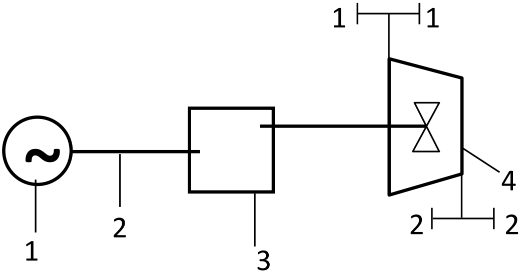


Figure 5. Schematic diagram of the turbocharger:
1 - electric motor, 2 - compressor shaft,
3 - multiplier, 4 - compressor stage.

Thermophysical characteristics of ammonia [1], annex (Tables P-3, P-6)

M = 17,031 kg/kmol, T_C = 405,6 K; P_C = 111,3 atm

virial coefficients [b_{1j}], needed to calculate the compressibility factor z of a non-ideal gas.

$$b_{10} = 4{,}53564 \cdot 10^{-3} \text{ m}^3/\text{kg};$$

$$b_{11} = -1{,}46817 \cdot 10^{-2} \text{ m}^3/\text{kg};$$

$$b_{12} = 1{,}35886 \cdot 10^{-2} \text{ m}^3/\text{kg};$$

$$b_{13} = -9{,}91264 \cdot 10^{-3} \text{ m}^3/\text{kg}.$$

Heat capacity series constants $C_P = \sum_{i=0}^{n=3} d_i T^i$

$$d_0 = 1{,}605 \text{ kJ/(кJ} \cdot \text{К)};$$

$d_1 = 1{,}4003 \cdot 10^{-3}$ kJ/(кJ · К^2);

$d_2 = 1{,}00328 \cdot 10^{-6}$ kJ/(кJ · К^3);

$d_3 = -6{,}962 \cdot 10^{-10}$ kJ/(кJ · К^4).

1st stage:

Calculation of the power consumed by an ideal compressor stage compressing ammonia.

The minimum compression work corresponds to a reversible adiabatic process. The calculation formula has the following form:

$$\dot{W}_{12S} = -m\int_{T_1}^{T_{2S}} dh = -\dot{m}\left[\int_{T_1}^{T_{2S}} Cp_{u\partial}dT + (\Delta h^{\partial}_{2S} - \Delta h^{\partial}_{1})\right]. \qquad (3.2)$$

The results of calculations of the density, as well as isothermal deviations of the thermodynamic functions of ammonia from the ideal gas state according to the parameters of the gas at the inlet to the compressor stage are presented in Table. 3.1.

Table 3.1.

Thermodynamic functions of the state of ammonia at the compressor inlet

T_1, K	$B_1 \cdot 10^2$, m^3/kg	$\rho_1^{u\partial}$, kg/m^3	$z^{(0)}$	z	ρ_1, kg/m^3	$\Delta s_T^{\partial} \cdot 10^3$, kJ/(kg·K)	Δh_T^{∂}, kJ/kg
306	–1,416	0,6694	0,9905	0,9904	0,676	–13,581	–5,574

The search for the temperature T_{2S} at the outlet of the uncooled ideal compressor stage is carried out based on the process condition

X = S = const according to the equation:

$$f(T_{2S}) = S_{2S} - S_1 = \int_{T_1}^{T_{2S}} \frac{C_{Pu\partial}}{T}dT - \mathrm{R_M}\ln\frac{P_2}{P_1} + \Delta s^{\partial}_{2S} - \Delta S^{\partial}_{1} = 0, \qquad (3.3)$$

where Δs^{∂}_{1}, Δs^{∂}_{2S} – isothermal deviations of the entropy of ammonia from the ideal gas state in terms of gas parameters at the inlet and outlet of the compressor stage.

The zero approximation $T_{2S}^{(0)}$ is set from the condition $\Delta s^{\partial}_{2} = 0$, further values are found from the calculated relation (2.3) by the method of successive approximation.

The calculation results are presented in Table. 3.2

Table 3.2.

Calculation of the temperature of ammonia T_{2S} at the outlet of an ideal compressor

T_{2S}, K	$\rho_{u\partial}$, kg/m³	$B\cdot 10^3$, kg/m³	$z^{(0)}$	z	ρ_{2S}, kg/m³	$\int_{T_1}^{T_{2S}} \frac{C_{Pu\partial}}{T} dT$, kJ/(kg·K)	$\Delta s_T^{\partial}\cdot 10^3$, kJ/(kg·K)	$f(T_{2S}^i)\cdot 10^3$, kJ/(kg·K)
436,022	2,349	–5,342	0,987	0,987	2,3799	0,7862	–16,295	–6,4977
436,798	2,345	–5,316	0,9875	0,987	2,376	0,7904	–16,467	–2,47

Let's choose T_{2S} based on the condition:

$$\varepsilon_T = \left| \frac{T_{2S}^i - T_{2S}^{i-1}}{T_{2S}^i} \right| \le 1\cdot 10^{-2} \text{ (table 3.2).}$$

T_{2S} = 436,798 K with precision

$$\varepsilon_T = \left| \frac{436{,}022 - 436{,}798}{436{,}798} \right| = 0{,}177\cdot 10^{-2}$$

The calculation of the disposable work W_{12S}^P of the ideal ammonia compression process is carried out according to the calculated ratio (3.2). The calculation results are presented in Table. 3.3.

Table 3.3.

Functions of the state of ammonia at the outlet of an ideal compressor

T_{2S}, K	$\rho_{2Su\partial}$, kg/m³	$B\cdot 10^3$, m³/kg	$z^{(0)}$	z	ρ_{2S}, kg/m³	$\int_{T_1}^{T_{2S}} C_{Pu\partial} dT$, kJ/kg	Δh_{2S}^{∂}, kJ/kg
436,798	2,345	–5,316	0,9875	0,987	2,376	291,434	–9,789

$$W_{12S} = -\left[\int_{T_1}^{T_{2S}} C_{Pu\partial} dT + (\Delta h_{2S}^{\partial} - \Delta h_1^{\partial}) \right] = -[291{,}434 - (9{,}789 - 5{,}574)] = -287{,}219 \text{ kJ/kg.}$$

The power consumed by an uncooled compressor stage in an ideal process, without taking into account frictional forces, is:

$$\dot{W}_{12S} = \dot{m} W_{12S} = -287{,}219 \text{ kW}.$$

2^{nd} stage:

Calculation of the actual power absorbed by a stage of a non-ideal compressor compressing ammonia, $\dot{W}_{12_{вн}}$.

The power absorbed by a stage of a non-ideal compressor $\dot{W}_{12_{вн}}$, is calculated by the relation (3.1):

$$\dot{W}_{12_{вн}} = \frac{\dot{W}_{12S}}{\eta_S},$$

where $\dot{W}_{12S} = \dot{m} W_{12S} = -287{,}219$ kW – power of an ideal compressor; η_S – adiabatic internal efficiency determined from the diagram, fig. 6.

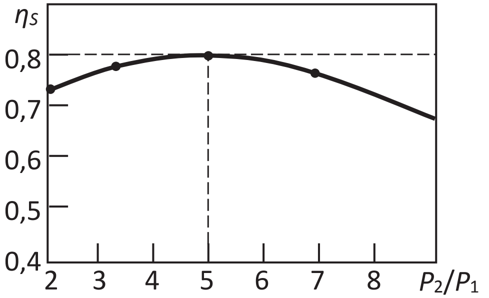


Figure 6. *Dependence of the adiabatic internal efficiency on the degree of pressure increase P_2/P_1*

Analyzing the data of the presented diagram, we choose the value of the indicator efficiency, $\eta_S = 0{,}8$ at the degree of pressure increase $P_2/P_1 = 5$.

The power consumed by an uncooled compressor stage in a real process is:

$$\dot{W}_{12_{вн}} = \frac{\dot{W}_{12S}}{\eta_S} = \frac{-287{,}219}{0{,}8} = -359{,}024 \text{ kW}.$$

3^{rd} stage:

Calculation of the ammonia temperature at the outlet of the uncooled compressor stage in the actual compression process T_2, K.

The temperature of ammonia at the outlet of the uncooled compressor stage in a real compression process is determined from the total energy balance equation for a non-equilibrium process (2.2) Analyzing this equation, we obtain the following relation:

$$\dot{W}_{12_{вн}} = -\dot{m}\int_{1}^{2} dh = -m\left[\int_{T_1}^{T_2} C_{Pu\partial}dT + (\Delta h_2^{\partial} - \Delta h_1^{\partial})\right], \qquad (3.4)$$

where Δh_1^{∂}, Δh_2^{∂} – isothermal deviation of the enthalpy of ammonia from the ideal gas state in terms of gas parameters at the inlet and outlet of the compressor stage; $\dot{W}_{12_{вн}} = -359{,}024$, kW.

The temperature T_2 is calculated according to the following relation by the method of successive approximation:

$$T_2^{(i)} = T_1 - \frac{1}{\overline{C}_{Pu\partial}}\left[W_{12_{вн}} - \Delta h_1^{\partial} + \Delta h_2^{\partial}\right], \qquad (3.5)$$

where $\overline{C}_{Pu\partial} = \dfrac{\int_{T_1}^{T_2^{(i-1)}} C_{Pu\partial}dT}{T_2^{(i\text{-}1)} - T_1}$ – the average heat capacity of ammonia in the temperature range from T_1 to $T_2^{(i-1)}$, kJ/(kg·K); $\Delta h_1^{\partial} = -5{,}574$ kJ/kg (table 3.1).

The calculation of the zero approximation of the final temperature $T_2^{(0)}$ is carried out according to relation (3.5) based on the condition that $\Delta h_2^{\partial} = 0$:

$$T_2^{(0)} = T_1 - \frac{1}{C_{Pu\partial}}\left[W_{12_{вн}} + \Delta h_1^{\partial}\right], \qquad (3.5, a)$$

where $C_{Pu\partial} = 4{,}5R_M$; $W_{12_{вн}} = -359{,}024$; kJ/kg; $\Delta h_1^{\partial} = -5{,}574$; kJ/kg; (table 3.1).

$$T_2^{(0)} = 306 - \frac{1}{4{,}5 \cdot 488{,}169 \cdot 10^{-3}} \cdot \left[-359{,}024 - 5{,}574\right] = 466{,}88 \text{ K}.$$

Further calculations T_2^i are carried out according to relation (3.4) with an accuracy:

$$\varepsilon_T = \left|\frac{T_2^i - T_2^{i-1}}{T_2^i}\right| \le 1 \cdot 10^{-2} \text{ (table 3.4)}.$$

Table 3.4.

Calculation of the temperature of ammonia T_2 at the outlet of a non-ideal compressor

T_2^i, K	B·10³, m³/kg	$\rho_{2u\partial}$, kg/m³	$z^{(0)}$	z	ρ_2, kg/m³	Δh_2^{∂}, kJ/kg	$\dfrac{\int_{T_1}^{T_2} C_{Pu\partial}dT}{T_2^{(i\text{-}1)} - T_1}$, kJ/(kg·K)
466,88	–4,464	2,194	0,9903	0,9901	2,216	–8,186	2,2557
466,32	–4,477	2,196	0,9902	0,99	2,218	–8,208	2,256

Let's choose T_2 = 466.32 K with accuracy:

$$\varepsilon_T = \left|\frac{466{,}32 - 466{,}88}{466{,}32}\right| = 0{,}12 \cdot 10^{-2}.$$

4th stage:

Calculation of kinetic energy losses due to friction in a non-equilibrium process, i.e. dissipation estimate $\dot{\Psi}_{12}^{\text{тр}}$, kW.

The dissipation is estimated using the following equation, obtained from the balance equation of kinetic and potential energy for a nonequilibrium process (2.4):

$$\dot{\Psi}_{12}^{\text{тр}} = -\dot{W}_{12_{\text{вн}}} + \dot{m}\int_1^2 -\frac{dP}{\rho}, \qquad (3.6)$$

where $\dot{W}_{12_{\text{вн}}}$ = –359,024 kW – internal power expended in the actual process of ammonia compression, taking into account friction forces;

$\dot{m}\int_1^2 -\frac{dP}{\rho} = \dot{W}_{12_{\text{пол}}}$ – the power of the polytropic equilibrium process, calculated by the formula (3.6 a):

$$\dot{W}_{12пол} = \dot{m}\frac{\bar{n}}{\bar{n}-1}\frac{P_1}{\rho_1}\left[1-\left(\frac{P_2}{P_1}\right)^{\frac{\bar{n}-1}{n}}\right] \qquad (3.6\text{ a})$$

where: $\bar{n} = \dfrac{\ln\frac{P_1}{P_2}}{\ln\frac{\rho_2}{\rho_1}}$ – the average value of the polytropic index for the initial and final parameters characterizing the state of the gas at the inlet and outlet of the compressor stage.

In the introduction, we emphasized that the real process cannot be represented on a thermodynamic diagram; a real process can be quantified only if the working fluid at the beginning and at the end of the process is in certain equilibrium states. The polytropic process acts as such a process, as the closest to the real one, since the final and initial states of the working fluid. In our case, ammonia, of both processes completely coincide. The calculation of the power of the polytropic process is carried out according to the relation (3.6a). The average value of the polytropic index is calculated from the initial and final parameters characterizing the state of ammonia at the inlet and outlet of the compressor stage.

$$\bar{n} = \frac{\ln\frac{P_2}{P_1}}{\ln\frac{\rho_2}{\rho_1}} = \frac{\ln 5}{\ln\frac{2{,}218}{0{,}676}} = 1{,}354;$$

$$\dot{W}_{12пол} = 1\cdot\frac{1,354}{1,354-1}\cdot\frac{10^5}{0,676}\cdot\left[1-5^{\frac{1,354-1}{1,354}}\right] = -295,94 \text{ кВт.}$$

The dissipation value $\dot{\Psi}_{12}^{тр}$ is determined by relation (2.6)

$$\dot{\Psi}_{12}^{тр} = -\dot{W}_{12_{вн}} + \dot{W}_{12пол} = -(-359,024) - 295,94 = 63,084 \text{ kW}$$

The dissipation $\dot{\Psi}_{12}^{тр} = 63,084$ kW is numerically equal to that part of the kinetic energy flow that is converted into internal energy due to overcoming the friction forces that impede the process.

The value of the polytropic efficiency of the process $\eta_{пол}$ is determined by the relation (3.1):

$$\eta_{пол} = \frac{\dot{W}_{12пол}}{\dot{W}_{12_{вн}}} = \frac{-295,94}{-359,024} = 0,82$$

The polytropic efficiency of the process $\eta_{пол}$ = 0,82 characterizes the degree of energy perfection of the ammonia compression process.

Let us represent the polytropic equilibrium process $1-2\bar{n}$ of ammonia compression in the coordinates (*PV*), (*TS*) (Fig. 7).

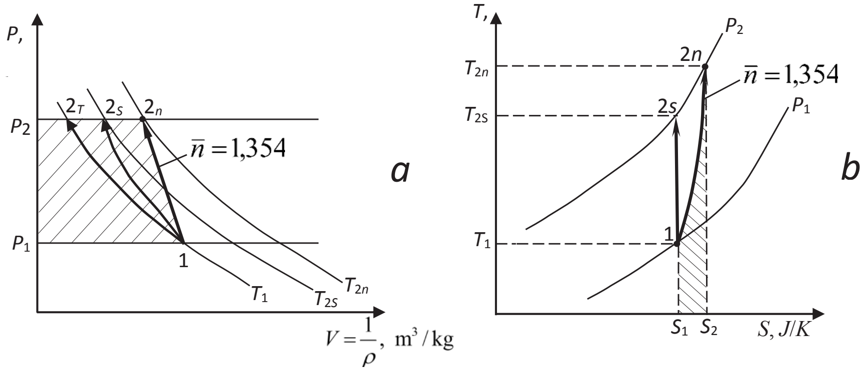


Figure 7. *Polytropic equilibrium compression process of ammonia in the coordinates PV(a), TS(b)*

5th stage:

Analysis of the exergy perfection of the compression process.

The exergy analysis of the process is carried out on the basis of the exergy balance equation:

$$\sum_{i=1}^{n} \dot{m}_i e_{xi} + \dot{E}_{X}^{T/o} - \dot{W}^{tex} + \dot{E}x^{диф} - in\dot{D}_{12} = 0, \tag{3.7}$$

where $\sum_{i=1}^{n}\dot{m}_i e_{xi}$ – resulting exergy flow due to the visible, convective movement of the medium, W;

$\dot{E}x^{т/о}$ – exergy flux due to heat transfer, W;

$\dot{W}^{tex}$ – flow of mechanical work, W;

$in\dot{D}_{12}$ – exergy losses due to the irreversibility of the process, are estimated by the Gouy-Stodola formula:

$$in\dot{D}_{12} = T_{o.c.} in\dot{S}_{12}. \tag{3.8}$$

The exergy efficiency of the uncooled compressor stage is estimated by the following relation:

$$\eta_{ex} = \frac{\dot{E}x_{вых} - \dot{E}x_{транзит}}{\dot{E}x_{вх} - \dot{E}x_{транзит}}, \tag{3.9}$$

where $\dot{E}x_{вых}$ – exergy flow at the outlet of the compressor stage;

$\dot{E}x_{вх}$ – exergy flow at the inlet to the compressor stage;

$\dot{E}x_{транзит}$ – transit flows of exergy, i.e. those that make up that part of the exergy of the incoming flows, which passes invariably through the entire apparatus.

In this case:

$$\dot{E}x_{вых} = \dot{m}e_{x2}$$

$$\dot{E}x_{вх} = \dot{m}e_1 - \dot{W}_{12_{вн}}$$

$$\dot{E}x_{транзит} = m\dot{e}_{x1}$$

We obtain the following calculation formula for the exergy efficiency of the compressor stage:

$$\eta_{ex} = \frac{\dot{m}(e_{x2} - e_{x1})}{me_{x1} - \dot{W}_{12}^{tex} - \dot{m}e_{x1}} = \frac{\dot{m}(e_{x2} - e_{x1})}{-\dot{W}_{12_{вн}}} = 1 - \frac{in\dot{D}_{12}}{-\dot{W}_{12_{вн}}}, \tag{3.10}$$

where $in\dot{D}_{12} = \dot{m}T_{o.c.}\left[\int_{T_1}^{T_2} C_{Pид}\frac{dT}{T} - R_M \ln\frac{P_2}{P_1} + (\Delta s_2^{д} - \Delta s_1^{д})\right]$ – exergy losses due to the irreversibility of the process.

The results of calculations of the exergy efficiency of the compressor stage are presented in table. 3.5.

Table 3.5.

Calculation results of the exergy efficiency (η_{ex}) of the compressor stage

T_1, K	T_2, K	$\int\limits_{T_1}^{T_2}\frac{C_{Pu\partial}}{T}dT$, kJ/kg·K	$\Delta s_1^{\partial}10^3$, kJ/kg·K	$\Delta s_2^{\partial}10^3$, kJ/kg·K	$in\dot{D}_{12}$, kW	η_{ex}
306	466,32	0,9459	–13,581	– 12,812	47,999	0,866

The final loss of exergy, which can only be compensated by external energy carriers (steam, fuel, electricity) $in\dot{D}_{12} = 46{,}716$ kW. The dissipation of the ammonia compression process was $\dot{\Psi}_{12}^{\text{тр}} = 63{,}084$ kW. The difference between these values is the part of the energy that can be used in the future. In this case, this value has the following value:

$$\dot{\Psi}_{12}^{mp} - in\dot{D}_{12} = 63{,}084 - 47{,}999 = 15{,}085 \text{ kW}.$$

The possibility of utilizing this energy is limited by climatic and temporal conditions and depends on the average temperature of the process. Since dissipation is quantitatively equal to the heat that is supplied in a polytropic equilibrium process without friction, the average temperature of a polytropic process can be determined from the following relation:

$$\dot{\Psi}_{12}^{mp} = \dot{Q}_{12n} = \dot{m}\overline{T}\int\limits_{1}^{2n} dS, \tag{3.11}$$

where $\overline{T}$ – average temperature of the process, K.

Based on the data obtained (Table 2.5), we determine the value of the average process temperature $\overline{T}$, K:

$$\overline{\mathrm{T}}_{\text{cp}} = \frac{\dot{\Psi}_{12}^{mp}}{\dot{m}[S_2 - S_1]} = \frac{\dot{\Psi}_{12}^{mp}}{\dot{m}\left[\int\limits_{T_1}^{T_2}\frac{C_{Pu\partial}}{T}dT - R_M\ln\frac{P_2}{P_1} + \Delta s_2^{\partial} - \Delta s_1^{\partial}\right]};$$

$$\overline{\mathrm{T}}_{\text{cp}} = \frac{63{,}084}{1\cdot[0{,}9459 - 488{,}169\cdot 10^{-3}\ln 5 - 12{,}812\cdot 10^{-3} + 13{,}581\cdot 10^{-3}]} = 391{,}085 \text{ K}.$$

The difference between the dissipation of kinetic energy for friction and the loss of exergy due to the irreversibility of the process is the part of the energy that can be used in the future. An example of energy recovery is its use in heating systems.

Another example of the utilization of this energy can be the transfer to subsequent links in the technological chain, i.e. its use as a secondary energy resource. This is especially effective in large-scale and mass production [9].

6th stage:

Evaluation of the energy perfection of the compressor unit.

The following formula is used to determine the capacity of the compressor unit:

$$\dot{W}^{э}_{\text{к.у.}} = \frac{\dot{W}_{12S}}{\eta_S \eta_{\text{мех}} \eta_{\text{пер}} \eta_{\text{дв}}}, \qquad (3.12)$$

where $\dot{W}_{12S}$ – power consumed by the ideal compressor stage;

η_S – adiabatic internal efficiency of the compressor stage;

$\eta_{\text{мех}}$ – mechanical efficiency of the shaft, which takes into account friction losses between the moving parts of the compressor;

$\eta_{\text{пер}}$ – transfer efficiency, which takes into account the presence of a multiplier as an intermediate device between the shaft and the electric motor;

$\eta_{\text{дв}}$ – The efficiency of the electric motor.

In practice, the values of these efficiency factors are as follows [7, 11]:

$\eta_{\text{мех}} = 0{,}98$; $\eta_{\text{пер}} = 0{,}94$; $\eta_{\text{дв}} = 0{,}95$.

The power of the compressor plant according to formula (3.12) has the following value:

$$\dot{W}^{э}_{\text{к.у.}} = \frac{-287{,}219}{0{,}8 \cdot 0{,}98 \cdot 0{,}94 \cdot 0{,}95} = -410{,}25 \text{ кВт.}$$

The exergy efficiency of the compressor unit is as follows:

$$\eta_{ex_{\text{к.у.}}} = 1 - \frac{in\dot{D}_{12}}{-\dot{W}^{э}_{\text{к.у.}}} = 1 - \frac{47{,}999}{410{,}25} = 0{,}883.$$

Task 3.2

Ammonia is compressed in a cooled reciprocating compressor, the gas flow rate is $\dot{m}$ = 1 kg/s. The compression process is non-equilibrium, it is known that the ratio of the heat removed and the compression work expended is φ = 0,6. Gas parameters at the compressor inlet: T_1 = 306 K, P_1 = 1 bar, gas parameters at the compressor outlet: P_2 = 5 bar. The functions of the state of ammonia are calculated according to the truncated virial Bogolyubov-Mayer equation. The initial data for ammonia (T_c, K; P_c atm; $C_p = \sum_{i=0}^{n=3} d_i T^i$; kJ/(kg · K)) are given in the condition of problem 3.1 and are presented in Annex (P-1, P-2). The internal efficiency is η_T = 0,7.

To define:

1. Internal power consumed by the compressor, $\dot{W}_{12_{\text{вн}}}$, kW.
2. The final temperature of ammonia at the outlet of the compressor, T_2, *K*.
3. Dissipation of kinetic energy into friction, $\dot{\psi}^{\text{тр}}_{12}$, kW.
4. Exergy efficiency of the compressor unit in the nominal mode of operation η_{ex}.

Let's imagine a schematic diagram of a reciprocating compressor (Fig. 8).

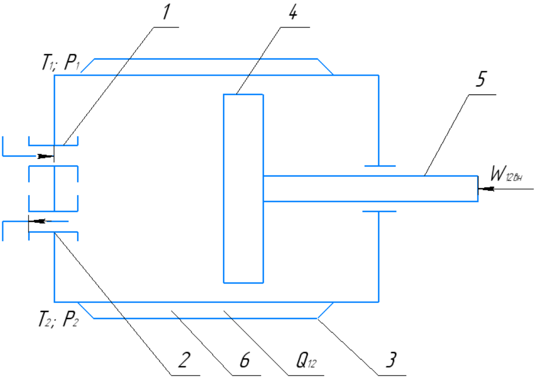


Figure 8. *Schematic diagram of a reciprocating compressor:*
1 - suction valve; 2 - discharge valve; 3 - cylinder body; 4 - piston; 5 - shaft; 6 - cooling jacket.

1st stage:

We determine the internal power of the compressor spent in the real process of gas compression $\dot{W}_{12_{\text{вн}}}$, kW.

The internal power of an isothermal compressor, taking into account the indicated efficiency, is calculated according to the following relationship:

$$\dot{W}_{12_{\text{вн}}} = \frac{\dot{m} W^P_{12_T}}{\eta_T}, \tag{3.12}$$

where $W^P_{12_T} = -R_M T_1[\ln\frac{\rho_{2_T}}{\rho_1} + 2B(\rho_{2_T} - \rho_1)]$ - work of an ideal compressor compressing 1 kg of gas in isothermal mode [1];

$\eta_T = 0{,}7$ – internal isothermal efficiency of the reciprocating compressor;

$\dot{m} = 1$ kg/s – gas consumption.

The results of calculations of the internal power of the real ammonia compression process are presented in Table. 3.6.

Table 3.6.

Calculation of the internal power of a non-ideal compressor

T_1, K	ρ_1, kg/m³	B · 10², kg/m³	ρ_{2_T}, kg/m³	$W^P_{12_T}$, kJ/kg	$\dot{W}_{12_{\text{вн}}}$, kW
306	0,676	–1,416	3,523	–234,56	–335,086

The power consumed by the compressor in the real process is: $\dot{W}_{12_{\text{вн}}} = -335{,}086$ kW.

2nd stage:

We determine the temperature of ammonia at the outlet of the compressor T_2, K. The temperature T_2 is calculated by the method of successive approximation from the relation obtained on the basis of the equation for the balance of the total energy of the nonequilibrium

$$f(T_{2n}) = -\int_{T_1}^{T_2} C_{pи\partial}dT + (\Delta h_1^{\partial} - \Delta h_2^{\partial}) - (1-\varphi)W_{12_{вн}} = 0, \tag{3.13}$$

where $\int_{T_1}^{T_2} C_{pи\partial}dT = \sum_{i=0}^{n=3} \frac{d_i\left(T_2^{i+1} - T_1^{i+1}\right)}{i+1}$, kJ/kg;

$\Delta h_1^{\partial} = -5{,}574$ kJ/kg (table 3.1);

Δh_2^{∂} – isothermal deviation of the enthalpy of ammonia from the ideal gas state according to the parameters of the gas at the outlet of the compressor, kJ/kg;

$W_{12_{вн}} = -335{,}086$ kJ/kg (table 3.6).

The value of the zero approximation $T_{2n}^{(0)}$ is determined based on the condition: $\Delta h_2^{\partial} = 0$; $C_{pи\partial} = 4{,}5R_M$.

We get the following value:

$$T_{2n}^{(0)} = 364{,}48 \text{ K}.$$

Subsequent values $T_{2n}^{(i)}$ are determined based on the value $f(T_{2n}^{(i)})$, obtained from equation (3.13). We solve by numerical approximation method with accuracy:

$$\varepsilon_T = \left|\frac{T_{2n}^{(i)} - T_{2n}^{(i-1)}}{T_{2n}^{i}}\right| \le 1\cdot 10^{-2}.$$

The calculation results are presented in Table. 3.7.

Table 3.7.

Calculation of the temperature of ammonia $\boldsymbol{T_{2n}}$ *at the outlet of a non-ideal compressor*

ρ_1, kg/m³	Δh_1^{∂}, kJ/kg	ρ_{2T}, kg/m³	$W_{12_T}^{P}$, kJ/kg	T_{2n}, K	ρ_{2n}, kg/m³	$\int_{T_1}^{T_{2n}} C_{pи\partial}dT$, kJ/kg	Δh_2^{∂}, kJ/kg	$f(T_{2n})$, kJ/kg
0,676	–5,574	3,523	–234,56	373,087	2,8099	145,493	–15,423	–1,6087
0,676	–5,574	3,523	–234,56	372,345	2,816	143,8562	–15,525	0,1291

Selecting a value $T_{2n} = 372{,}345$ K with precision

$$\varepsilon_T = \left|\frac{372{,}345 - 373{,}087}{372{,}345}\right| = 0{,}2 \cdot 10^{-2}.$$

3rd stage:

Calculation of dissipation of kinetic energy on friction $\dot{\Psi}_{12}^{\text{тр}}$, k$W$.

We estimate dissipation using the balance equation for kinetic and potential energy for a nonequilibrium process (2.4).

$$\dot{\Psi}_{12}^{\text{тр}} = -\dot{W}_{12_{\text{вн}}} + \dot{m}\int_{T_1}^{T_{2n}} -\frac{dP}{\rho},$$

where $\dot{W}_{12_{\text{вн}}} = -335{,}086$ kW – internal power of the nonequilibrium process;

$\dot{m}\int_{T_1}^{T_{2n}} -\frac{dP}{\rho} = \dot{m}\frac{\bar{n}}{\bar{n}-1}\cdot\frac{P_1}{\rho_1}\left[1-\left(\frac{P_2}{P_1}\right)^{\frac{\bar{n}-1}{\bar{n}}}\right]$ – power of the equilibrium polytropic process, kW,

$\bar{n} = \dfrac{ln\frac{P_2}{P_1}}{ln\frac{\rho_{2_n}}{\rho_1}}$ - the average value of the polytropic index for the initial and final parameters characterizing the state of the gas at the inlet and outlet of the compressor.

The results of calculations of viscous dissipation $\dot{\Psi}_{12}^{\text{тр}}$, k$W$ are presented in Table. 3.8.

Table 3.8.

Analysis of the polytropic equilibrium process of ammonia compression

ρ_1, kg/m^3	T_2, K	$\rho_{2\bar{n}}$, kg/m^3	$\bar{n}$	$\dot{W}_{12_{\text{пол}}}$, kW	$\dot{W}_{12_{\text{вн}}}$, kW	$\dot{\Psi}_{12}^{\text{тр}}$, kW
0,676	372,345	2,815	1,128	–261,27	–335,086	73,816

Let us represent the equilibrium polytropic process of ammonia compression (1–2$\bar{n}$) in coordinates (PV), (TS) (table 9).

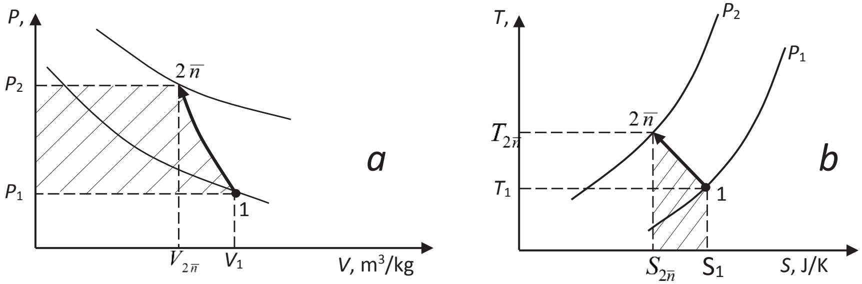


Figure 9. *Polytropic equilibrium process of ammonia compression with heat removal in the coordinates PV(a), TS(b).*

We determine the polytropic efficiency, the value of which characterizes the degree of energy perfection of the gas compression process $\eta_{пол}$.

$$\eta_{пол} = \frac{\dot{W}_{12_{пол}}}{\dot{W}_{12_{вн}}} = \frac{-261,27}{-335,086} = 0,78 .$$

4th stage:

Exergy analysis of ammonia compression process with heat removal.

The exergy analysis of the ammonia compression process with heat removal is carried out on the basis of the exergy balance equation (3.7). Heat flow exergy is calculated based on equation (3.14):

$$\dot{E}x_{12}^{Q} = \dot{Q}\left(1 - \frac{T_{о.с.}}{\overline{T}_{пр}}\right), \tag{3.14}$$

where $\dot{Q} = \varphi \dot{W}_{12_{вн}}$ – heat flow removed from gas, kW,

$T_{о.с.}$= 298,15 K – ambient temperature, under standard conditions,

$\overline{T}_{пр}$ – average thermodynamic temperature of the process, K,

The exergy balance equation (2.7), taking into account relation (3.14), takes the following form:

$$\dot{m}(e_{x1} - e_{x2}) + \dot{Q}(1 - \frac{T_{о.с.}}{\overline{T}_{пр}}) - \dot{W}_{12_{вн}} - in\dot{D}_{12} = 0 , \tag{3.15}$$

where $\dot{m}(e_{x1} - e_{x2}) = -\dot{m}\left[\begin{array}{c} \int_{T_1}^{T_2} C_{рид} dT + (\Delta h_2^{д} - \Delta h_1^{д}) - \\ - T_{о.с.}\left(\int_{T_1}^{T_2} \frac{C_{рид}}{T} dT - R_M \ln\frac{P_2}{P_1} + (\Delta s_2^{д} - \Delta s_1^{д})\right) \end{array}\right]$ – exergy flow due to the convective movement of the gas flow, kW,

$\dot{E}x^{Q} = \dot{Q}\left(1-\frac{T_{\text{о.с.}}}{\overline{T}_{\text{пр}}}\right)$ – heat flow exergy, kW,

$\dot{W}_{12_{\text{вн}}} = \frac{\dot{W}^{P}_{12_T}}{\eta_T}$ – internal power of the compressor, kW,

$in\dot{D}_{12}$ – internal exergy losses due to irreversibility of the process, kW.

The calculation of internal exergy losses $in\dot{D}_{12}$, due to the irreversibility of the process itself is carried out on the basis of the following relationship:

$$in\dot{D}_{12} = \dot{\Psi}_{12}^{\text{тр}}\frac{T_{\text{о.с.}}}{\overline{T}_{\text{пр}}}, \tag{3.16}$$

where $\dot{\Psi}_{12}^{\text{тр}}$ – dissipation of kinetic energy due to friction, kW,

$T_{\text{о.с.}}$ = 298,15 K – ambient temperature under standard conditions,

$\overline{T}_{\text{пр}}$ – average thermodynamic temperature of the process, K.

The results of calculations of the ammonia exergy flux due to convection are presented in Table. 3.9. The calculation of isothermal deviations of the enthalpy Δh_T^{∂} entropy Δs_T^{∂} of ammonia from the ideal gas state is carried out according to the calculated ratios presented in the author's manual [1].

Table 3.9.

Calculation of loss of specific exergy of ammonia due to convection

T_1, K	T_2, K	$\int_{T_1}^{T_2} C_{pид}dT$, kJ/kg	$\int_{T_1}^{T_2} \frac{C_{pид}}{T}dT$, kJ/(kg·K)	Δh_1^{∂},	Δh_2^{∂},	$\Delta s_1^{\partial}\cdot 10^3$,	$\Delta s_2^{\partial}\cdot 10^3$,	$(e_{x1}-e_{x2})$, kJ/kg
				kJ/kg		kJ/(kg·K)		
306	372,345	144,00794	0,4255	–5,574	–15,522	–13,581	–30,657	–246,538

The calculation of the average thermodynamic temperature of the process $\overline{T}_{\text{пр}}$ is carried out on the basis of the exergy balance equation (3.15).

$$\overline{T}_{\text{пр}} = T_{o.c.}\left[\frac{\dot{Q}_{12}+\dot{\Psi}_{12}^{\text{тр}}}{\dot{Q}_{12}+\dot{m}(e_{x1}-e_{x2})-\dot{W}_{12_{\text{вн}}}}\right]=$$

$$=298{,}15\left[\frac{-201{,}0516+73{,}816}{-201{,}0516-246{,}538+335{,}086}\right]=337{,}19\ \text{K}.$$

The calculation of exergy losses due to the irreversibility of the process $in\dot{D}_{12}$ is carried out according to the calculated ratio (3.16):

$$in\dot{D}_{12} = \dot{\Psi}_{12}^{\text{тр}} \frac{T_{\text{о.с.}}}{\overline{T}_{\text{пр}}} = 73{,}816\frac{298{,}15}{337{,}19} = 65{,}27 \text{ кВт}.$$

The value $in\dot{D}_{12} = 65{,}27$ кВт is the final loss of exergy, which can only be compensated by external energy carriers.

The difference between the values of dissipation of kinetic energy and internal losses of exergy, due to the irreversibility of the process, is that part of the exergy that can still be usefully used in the future.

$$\dot{\Psi}_{\text{тр}} - in\dot{D}_{12} = 73{,}816 - 65{,}27 = 8{,}546 \text{ кВт}.$$

The calculation of the exergy efficiency of the ammonia compression process in a cooled compressor, provided that the removed heat flow is usefully used in the future, is carried out according to the relation (3.10):

$$\eta_{ex} = 1 - \frac{in\dot{D}_{12}}{-\dot{W}_{12_{\text{вн}}}} = 1 - \frac{65{,}27}{335{,}086} = 0{,}805.$$

In the event that the thermal energy of the cooling agent is not used, then the exergy losses are calculated according to the equation:

$$\dot{D}_{12} = in\dot{D}_{12} + ex\dot{D}_{12}, \tag{3.17}$$

where $\dot{D}_{12}$ – total exergy losses, kW,

$in\dot{D}_{12}$ – internal exergy losses, kW,

$ex\dot{D}_{12} = -\dot{E}x^{Q} = -\dot{Q}\left(1 - \frac{T_{\text{о.с.}}}{\overline{T}_{\text{пр}}}\right)$ – external exergy losses, kW.

The calculation of the exergy efficiency of the compression process with heat removal in this case is carried out according to the following relation:

$$\eta_{ex} = 1 - \frac{in\dot{D}_{12} + ex\dot{D}_{12}}{-\dot{W}_{12_{\text{вн}}}}. \tag{3.18}$$

The results of calculations of the exergy efficiency of the ammonia compression process without utilization of the removed heat are presented in Table. 3.10.

Table 3.10.

Calculation of the exergy efficiency of of the process
η_{ex} without disposal of removed heat flux

$\dot{Q}_{12}$, kW	$\dot{W}_{12_{\text{вн}}}$, kW	$ex\dot{D}_{12}$, kW	$in\dot{D}_{12}$, kW	$\overline{T}_{\text{пр}}$, K	η_{ex}
–201,0516	–335,086	23,278	65,27	337,19	0,736

The exergy analysis of the process indicates that it is advisable to use the thermal energy of the cooling agent as a secondary energy resource. The possibility of practical implementation of heat exergy in each specific case is decided individually. [9, 10].

Task 3.3

To obtain high-pressure gas, multistage compressors are used, between the stages of which heat exchangers are installed to provide cooling of the gas compressed in the previous stage. Let's consider the process of gas compression in a two-stage turbocharger with intermediate cooling in a refrigerator at constant pressure. The schematic diagram of the compressor unit is shown in fig. 10.

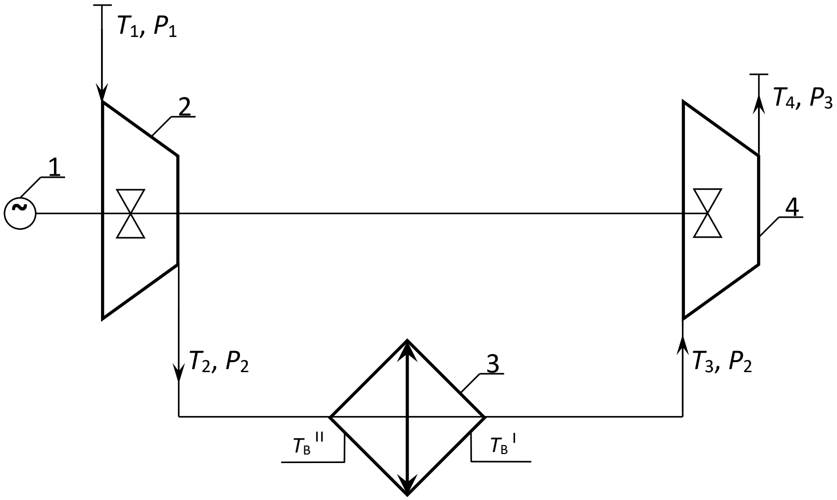


Figure 10. *Schematic diagram of a two-stage adiabatic compressor with intercooling:*
1 - electric motor; 2 - the first stage of the compressor;
3 - heat exchanger cooled by recycled water;
4 - the second stage of the compressor

Gas with initial parameters T_1, P_1 enters the first stage 2 of the compressor unit, where an adiabatic compression process takes place from the initial pressure P_1 to the intermediate pressure P_2. Then, the gas with temperature T_2, pressure P_2 is sent to the intermediate cooler 3, where it is cooled to the initial temperature T_1 at a constant pressure with water from the circulating water supply. The resistance of the cooler along the gas path is made small in order to save energy spent on compression, which makes it possible to consider the gas cooling process as isobaric. After the cooler 3, the gas is sent to the second stage 4 of the compressor unit, where the adiabatic compression process takes place from the intermediate pressure P_2 to the specified final pressure P_3.

In multi-stage compression, in order to select the optimal intermediate pressures at which the work would be the least, the distribution of the load on each stage is calculated according to the following relation:

$$\varepsilon_1 = \varepsilon_2 = \ldots = \varepsilon_n = \sqrt[n]{\varepsilon_{\text{общ}}}, \tag{3.19}$$

where ε_1, ε_2, … ε_n – the degree of gas pressure increase in the first stage, second stage, n-th stage of the multistage compressor;

n – number of compressor stages;

$\varepsilon_{\text{общ}} = \dfrac{P_{\text{кон}}}{P_{\text{нач}}}$ – total degree of increase in gas pressure in the compressor with the number of steps n from the initial pressure $P_{\text{нач}}$ to the specified final pressure $P_{\text{кон}}$.

If this condition (3.19) is met, the pressure ratio in all stages is the same, which is favorable not only for the power consumption, but also for the discharge temperatures in reciprocating compressors, which in this case are lower than with different pressure ratios in the stages. With an increase in the number of stages and intermediate coolers of the compressor unit, the compression process is more and more close to isothermal, i.e. to the most advantageous in terms of energy consumption. This does not exhaust the advantages of the multi-stage compression process. In reciprocating compressors, a decrease in the discharge temperature is achieved, and the risk of ignition of lubricating oils is reduced [10]. In the practice of compressor construction, there are very different relationships between the number of stages and the final pressure. The number of compressor stages of medium and high efficiency should be chosen so that the pressure ratio in each stage of the turbocharger does not exceed the value ε, equal to four. When compressing polyatomic gases, such as ammonia, it is advantageous to take higher pressure ratios than for compressors compressing diatomic gases, such as nitrogen. In compressors for gases with a low specific gravity, such as hydrogen, it is advantageous to adopt a reduced pressure ratio ($\varepsilon = 1{,}5 \div 2$), since the pressure losses between stages are below average. To increase the efficiency of compressors, they strive for the most complete cooling of the gas in intermediate coolers. The limit of possible cooling is determined by the initial temperature of the cooling water. When using water from a circulating water supply system, this temperature is determined by climatic and weather conditions. In modern designs of multistage compressors, the difference between the final and initial temperatures of the cooling water is 5–10 °C. The choice of the most advantageous number of stages should be carried out, guided not only by the desire for the minimum energy consumption, but also by considerations of a general economic nature.

Let us give a specific example of calculating gas compression in a two-stage compressor unit.

A two-stage ammonia turbocompressor with an intermediate isobaric cooler 3 serves to compress ammonia to a final pressure P_3 = 7 bar (fig. 10). The compression process is non-equilibrium, the consumption of ammonia is $\dot{m}_\text{г} = 1$ kg/s. The parameters of ammonia at the inlet to the first stage 2 of the compressor unit are as follows: P_1 = 1 bar, T_1 = 306 K. Cooling water from the circulating water supply is used to cool the ammonia compressed in the first stage 2 to an intermediate pressure P_2. The cooling of ammonia after the first stage 2 in the isobaric cooler 3 is achieved to the initial temperature $T_3 = T_1$ = 306 K. Water heating is $\Delta T_\text{В}$ = 5 K. The value of the adiabatic efficiency of each stage of the compressor unit is assumed to be the same and equal to the following value:

$$\eta_{S_{12}}^{\text{I}} = \eta_{S_{34}}^{\text{II}} = 0{,}8.$$

The functions and parameters of the state of ammonia are calculated according to the equation of state of an ideal gas with a constant heat capacity.

To define:

1. The power absorbed by each stage of the compressor unit $\dot{W}_{12\text{вн}}^{\text{I}}$, $\dot{W}_{34\text{вн}}^{\text{II}}$, kW.
2. Heat flow removed in the intermediate heat exchanger 3 $\dot{Q}_\text{В}$, kW.
3. Exergy efficiency of the compressor unit η_{ex}.

Stage 1:

Calculation of the power absorbed by the first stage of the compressor unit, $\dot{W}_{12\text{вн}}^{\text{I}}$, kW.

The determination of the power spent on compressing the gas from the initial pressure P_1 to the intermediate pressure P_2 in the first stage of the compressor unit is carried out according to the calculated ratio:

$$\dot{W}_{12_\text{I}}^{\text{tex}} = -\dot{m}\int_{T_1}^{T_2} C_{p_{\text{ид}}}\,dT = \dot{m}C_{p_{\text{ид}}}(T_1 - T_2), \tag{3.20}$$

where: $C_{p_{\text{ид}}} = 4{,}5R_M$

$T_2 = T_1 - \dfrac{T_1 - T_{2S}}{\eta_S}$ – ammonia temperature at the end of the nonequilibrium compression process in the first stage, K;

$T_{2S} = T_1\left(\dfrac{P_2}{P_1}\right)^{\frac{k-1}{k}}$ – temperature of ammonia at the end of the adiabatic equilibrium compression process in the first stage, K.

The determination of the intermediate pressure of ammonia after the first stage P_2 is carried out according to the calculated relation (3.19).

$$\frac{P_2}{P_1} = \frac{P_2}{P_3} = \sqrt{\frac{P_3}{P_1}} = \sqrt{7} = 2{,}646$$

The calculation results $\dot{W}^{\text{I}}_{12\text{вн}}$, $\dot{W}^{\text{II}}_{34\text{вн}}$ are presented in Table. 3.11.

Table 3.11.

Calculation of internal power of compressor steps **($\dot{W}^{\text{I}}_{12\text{вн}}$; $\dot{W}^{\text{II}}_{34\text{вн}}$)**

$T_1 = T_3$, K	$T_{2S} = T_{4S}$, K	$T_2 = T_4$, K	$C_{рид}$ kJ/(kg · K)	$\dot{W}^{\text{I}}_{12\text{вн}}$, kW	$\dot{W}^{\text{II}}_{34\text{вн}}$, kW	$\dot{Q}_{\text{В}}$, kW	$\frac{P_2}{P_1} = \frac{P_3}{P_2}$
306	380,82	399,53	2,196	–205,39	–205,39	–205,39	2,646

Since the ammonia compression ratio in both stages of the compressor unit is the same, the gas temperatures at the inlet to the first and second stages are equal to each other, and the adiabatic efficiency of the stages is the same, the internal power in both stages will be the same and equal to the following value :

$$\dot{W}^{\text{I}}_{12\text{вн}} = \dot{W}^{\text{II}}_{34\text{вн}} = –205{,}39 \text{ kW}.$$

2nd stage:

Calculation of the heat flow $\dot{Q}_{\text{В}}$, removed in the intermediate isobaric cooler 3, cooled by circulating water, kW.

The calculation is carried out in accordance with the total energy balance equation (2.2) for heat exchanger 3 with respect to ammonia:

$$\dot{m}(h_2 - h_3) + \dot{Q}_{\text{В}} = 0, \tag{3.21}$$

where $\dot{m}(h_2 - h_3) = \dot{m}C_p(T_2 - T_3)$ – the enthalpy flow of ammonia, kW;

$\dot{Q}_{\text{В}}$ – heat flow supplied by the cooling circulating water, kW;

$\dot{Q}_{\text{В}}$ = –205,39 кВт – heat flow removed from ammonia in the intermediate cooler 3.

To determine the flow rate of cooling water, we use the total energy balance equation (1.25) for heat exchanger 3 with respect to water:

$$\dot{m}_{\text{В}}(h'_{\text{В}} - h''_{\text{В}}) + \dot{Q}_{23} = 0, \tag{3.22}$$

where $\dot{Q}_{23}$ = 205,39 kW – heat flow supplied by ammonia compressed in the first stage;

$\dot{m}_{\text{В}}(h'_{\text{В}} - h''_{\text{В}}) = \dot{m}_{\text{В}} \cdot \overline{C}_{p_{\text{В}}}(T'_{\text{В}} - T''_{\text{В}})$ – enthalpy flow of water coming from the circulating water supply to heat exchanger 3, kW;

$\overline{C}_{p_{\text{В}}} = 4{,}19 \dfrac{\text{kJ}}{\text{kg} \cdot \text{K}}$ [3, nomogram XI] – the value of the heat capacity of water at an average temperature $\overline{T}$ = 302,5 K.

The cooling water flow is as follows:

$$\dot{m}_{\text{В}} = \frac{\dot{Q}_{23}}{m\overline{C}_p(T_{\text{В}}^{'} - T_{\text{В}}^{"})} = \frac{205{,}39}{4{,}19 \cdot 5} = 9{,}81 \text{ kg/s.}$$

Stage 3:

Calculation of the exergy efficiency of the compressor unit, η_{ex}.

The calculation of the exergy efficiency of the compressor unit is carried out according to relation (3.12):

$$\eta_{ex} = \frac{\dot{E}x_{\text{вых}} - \dot{E}x_{\text{тр}}}{\dot{E}x_{\text{вх}} - \dot{E}x_{\text{тр}}} = \frac{\dot{m}_{\text{г}}[C_p(T_4 - T_1) - T_{\text{о.с.}}(s_4 - s_1)]}{-\dot{W}^{\text{I}}_{12_{\text{вн}}} - \dot{W}^{\text{II}}_{34_{\text{вн}}} - \dot{W}_{\text{В}}},$$

where $\dot{E}x_{\text{вых}} = \dot{m}_{\text{г}} e_{x4}$ – ammonia exergy flow at the outlet of the compressor unit, kW;

$\dot{E}x_{\text{тр}} = \dot{m}_{\text{г}} e_{x1}$ – transit flow of exergy, kW;

$-\dot{W}^{\text{I}}_{12_{\text{вн}}}$; $-\dot{W}^{\text{II}}_{34_{\text{вн}}}$ – technical power of the first and second stages, kW;

$-\dot{W}_{\text{В}} = 9{,}81$ кВт – power absorbed by the pump for pumping circulating water;

$T_{\text{о.с.}} = 298{,}15\ K$ – ambient temperature.

$$\eta_{ex} = \frac{1[2{,}196(399{,}53 - 306) - 298{,}15(2{,}196 ln\dfrac{399{,}53}{306} - 0{,}489\ln 7)]}{-(-2 \cdot 205{,}39) - (-9{,}81)} = 0{,}764\,.$$

In the quasi-static approximation, the actual compression process in the compressor can be represented as an equilibrium polytropic process, the initial and final states of which completely coincide with the real process. Let's represent this process in the coordinates *PV*(*a*) and *TS*(*b*), (Fig. 11).

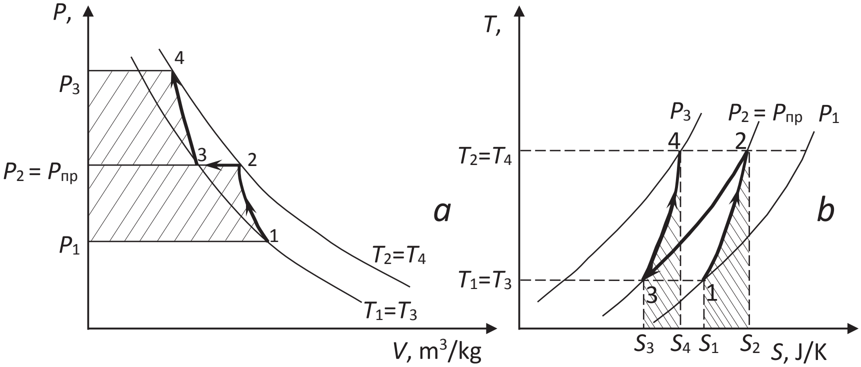


Figure 11. *Polytropic equilibrium process of two-stage ammonia compression with intermediate cooling in coordinates PV(a) and TS(b)*

Due to the cooling of ammonia in the intermediate cooler 3 at a constant pressure (isobaric process 2→3) the overall compression process in the compressor unit approaches isothermal, i.e. most advantageous in terms of energy savings.

For a non-ideal gas, the problem of economic distribution of compression between stages becomes more complicated. In this case, when the pressure ratios are equal, the work flow in individual stages is different. It is greater in the last stages if the final pressure is high enough.

In order to save energy in compressor installations, an automatic process control system is used, which ensures the regulation of those parameters, the deviation from which requires the compressor to be stopped in order to protect against an accident.

3.2 Analysis of the energy perfection of the processes of expansion and cooling of non-ideal gases.

3.2.1 Features of the use of low-temperature thermal resources

In most cases, the analysis of the effectiveness of actual processes in a particular device is carried out using the method of classical thermodynamics, which is able to determine predictions that are important for practice at the initial design stage. In this case, it is possible not only to predetermine the consumption of energy and material resources in a real unit, but also to get an idea of a number of engineering factors, such as the weight of the apparatus, the dimensions of individual components and the cost of their manufacture. The desire to reduce the cost of primary and traditional energy sources (consumption of fuel, electricity) without reducing or even increasing the return of energy to the end consumer due to its more rational transformation is the main trend of modern technology. The questions raised are reflected both when considering the features of the energy of low-temperature processes in chemical technology, and in a wide variety of industries. The need to use artificial cold arises in all cases when the task is to remove heat from a technological object at temperatures below ambient T_{cp}. The variety of chemical industries, in which almost all known physical and chemical processes are carried out using substances with a wide variety of properties, gives rise to a variety of specific technological problems that can be solved with the help of cold. However, some typical applications of cold in chemical technology can be identified:

1. The need for cooling in exothermic reactions of chemical interaction, and we are talking not only about preliminary cooling of the starting materials to a given temperature and ensuring the removal of process heat, but also about direct control of the rate and direction of the reaction. The highest yield of dichloroethane in the reaction of ethylene chlorination is fixed in the range from 243 to 253 K. Even the production of polymers of high molecular weight is expedient in the low-temperature regime of the polymerization reaction, in particular, butyl rub-

ber with desired properties is formed during the catalytic copolymerization of isobutylene and isoprene at 173 K.

2. Liquefaction of low-temperature gases and gas mixtures and associated with these processes low-temperature distillation and fractional condensation.

3. In the processes of polythermal crystallization, by changing the temperature, one can control the speed of the process, adjust the size and shape of the crystals. Fractionated crystallization at low temperatures is used in the production of aromatic compounds in the separation of para- and meta-xylenes (200 K), in the production of mineral fertilizers in the freezing of calcium nitrate (263 K), in dewaxing processes in the production of petroleum oils (240 K).

4. Large consumers of cold in the chemical industry are drying processes, including freeze-drying, industrial air conditioning systems.

In all cases, the introduction of cold allows you to create new technological processes, intensify production, increase product yield and quality. In addition, it becomes possible to reduce the level of toxicity of industrial emissions and create more comfortable working conditions. It should be noted that obtaining artificial cold is an expensive and energy-intensive process, in connection with which the questions of the economic justification of technological processes using cold arise very sharply. Modern chemical industries are the largest consumers of cold. At the same time, chemical enterprises have a huge amount of secondary energy resources (SER) in the form of flue and waste gases, gas flares, waste steam of low parameters. The use of these types of energy to produce cold in absorption refrigeration machines can drastically reduce electricity consumption, which is an important way to create economical (energy-saving) chemical-technological systems.

More than half a century of history of the development of refrigeration has led to its differentiation and specialization. Currently RU are classified:

- according to the temperature range of work;
- by type of energy used;
- according to the state of aggregation of the working fluid;
- by methods of obtaining a cooling effect.

There are areas distinguished:

1) Moderate cold 150 K $<T_{х}<T_{ос}$:

$T_{сублимации}CO_2 = 195$ K; $T_{нв}NH_3 = 240$ K; $T_{нв}SO_2 = 263$ K;

2) Deep cold 70 K$<T_{х}<$ 150 K:

$T_{нв}\,O_2 = 90$ K; $T_{нв}Ar = 87$ K; $T_{нв}N_2 = 72$ K;

3) Cryogenic cold 3–5 K$<T_{х}<$ 70 K:

$T_{нв}\,H_2 = 20$ K; $T_{нв}$ He = 27 K.

In refrigeration engineering, to obtain low temperatures, phase transformations, adiabatic expansion of gases and vapors with the return of external work, throttling, vortex effect, thermoelectric and thermomagnetic effect, and desorption are used.

Consider the energy consumption of the most used methods for obtaining the effect of cooling gases and vapors in the chemical industry.

3.2.2 Analysis of the adiabatic expansion of gases in the expander

The processes of expansion of gases and vapors are widely practiced in equipment of chemical industries, for example, in turbines (steam, gas, hydraulic), expanders (piston, turbo expanders), nozzles, throttle devices. Adiabatic expansion of gases and vapors with the return of external work in expanders is the most effective method of internal cooling. Expanders are low-temperature expansion machines that serve to produce cold by expanding the working fluid with a decrease in temperature and the return of external work (energy). The term "expander" comes from the French word "de'tendre", which means to reduce pressure. In practice, there are basically two classes of expanders:

1. Expansion machines of volumetric action, such as piston, screw and rotary expanders (Fig. 12).

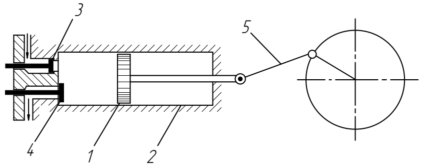


Figure 12. *Scheme of a piston expander:*
1 - piston; 2 - cylinder; 3 - inlet valve; 4 - exhaust valve;
5 - crank mechanism

2. Expansion machines of dynamic type (kinetic action), turbo-expanders (Fig. 13).

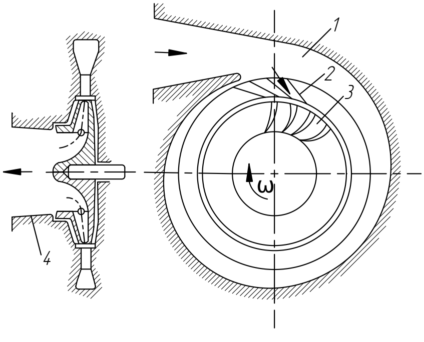


Figure 13. *Scheme of a centripetal jet turboexpander:*
1 - spiral gas supply; 2 - directing nozzle apparatus; 3 - rotor; 4 - outlet diffuser

In volumetric expanders, gas expansion occurs due to a direct change in the volume of the working fluid by moving a piston or some other device. In kinetic action expanders (turbo-expanders), gas expansion occurs due to the force interaction of the expanding gas with the blades of the impeller when the gas flow moves in a specially profiled channel in which a rotating lattice of the blade apparatus (impeller) is installed. With the help of the rotating blades of the impeller, the internal and kinetic energy of the gas flow is converted into mechanical energy of the rotating lattice of the blade apparatus. This mechanical energy is converted into electrical or thermal energy, and then transferred to the rotation of the blower or compressor impeller. Expanders of both volumetric and kinetic action, depending on the pressure of the working fluid used at the inlet, are divided into high, medium and low pressure expanders. High pressure expanders have an inlet pressure of more than 10 MPa; low-pressure apparatus, not more than 1.5 MPa. In accordance with the working gas used, the devices are divided into air, nitrogen, hydrogen, helium. Structurally, both volumetric expanders and expansion machines of kinetic action are very diverse.

Main energy characteristics of expanders

Expanders are characterized by an adiabatic (internal) coefficient of performance (COP), which is the ratio of real power, i.e. non-equilibrium process in the nominal mode of operation (excluding mechanical losses), to the reference power of the ideal expansion process:

$$\eta_{\mathrm{S}} = \frac{\dot{W}_{12}^{\mathrm{tex}}}{\dot{W}_{12\mathrm{S}}^{P}} = \frac{\dot{m} W_{12}^{\mathrm{tex}}}{\dot{m} W_{12\mathrm{S}}^{P}}, \tag{3.23}$$

where η_{S} – the adiabatic (internal) efficiency of the expander; $\dot{W}_{12}^{\mathrm{tex}}$ – the power of the expander in the actual expansion process, kW; $\dot{m}$ – the mass flow rate of the working fluid, kg/s; W_{12}^{tex} – the external work of the expander in the nonequilibrium expansion process, kJ/kg; $W_{12\mathrm{S}}^{\mathrm{P}}$ – external, disposable work of the expander in an ideal, equilibrium expansion process, kJ/kg; $\dot{W}_{12\mathrm{S}}^{\mathrm{P}}$ – the power of the expander in the ideal expansion process, kW.

For piston expanders, the values of the indicator (internal) efficiency are $\eta_{\mathrm{S}} = 0{,}6 - 0{,}75$, for turbo-expanders the higher values are $\eta_{\mathrm{S}} = 0{,}75 - 0{,}83$. Expanders are widely used in refrigeration cycles, i.e. reverse circular processes designed to transfer heat from less heated bodies to more heated ones. Expander refrigeration cycles are characterized by the following energy characteristics: refrigeration capacity, power absorbed by refrigeration unit, and coefficient of performance (Fig. 14).

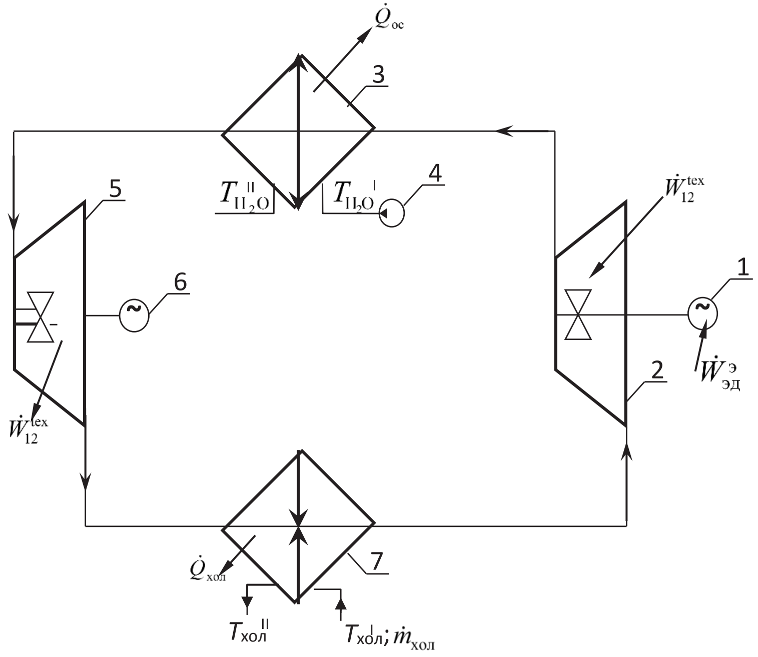


Figure 14. *Schematic diagram of the expander refrigeration cycle:*
1 - electric motor; 2 – adiabatic compressor; 3 – heat exchanger for cooling compressed gas; 4 - pump for water supply from the circulating system; 5 – turbo expander; 6 - electric motor; 7 - heat exchanger, where heat is removed from the cooling object

The principle of operation of the expander refrigeration cycle is based on the adiabatic expansion in the turboexpander 5 of pre-compressed gas (for example, nitrogen) in the adiabatic compressor 2 and then cooled in the isobaric heat exchanger 3. As a heat carrier that removes heat from the working fluid of the refrigeration cycle (for example, nitrogen) in the heat exchanger 3, the water supplied by the pump 4 from the circulating water supply circuit is used. There is a transfer of heat taken from nitrogen to the external environment. The technology provides for control of the water temperature at the inlet to the cooled heat exchanger. The prepared nitrogen enters the turbo expander 5, where the adiabatic expansion of nitrogen takes place with the return of external work W^{tex}_{12}. The expansion of the gas is accompanied by a decrease in its temperature, external work is removed in the form of electricity in the electric motor 6, sitting on the same shaft with the expander 5. Next, nitrogen is sent to the isobaric heat exchanger 7, where it is used as a coolant that removes heat from the cooling object $Q_{хол}$. Hydrocarbons (ethane, propane), air can serve as coolants.

Let us consider in detail the real process of adiabatic expansion of nitrogen in an expander. When studying the expansion process in expanders, the following tasks should be solved:

1. Determination of the integral cooling effect, i.e. degree of cooling of the working fluid;
2. Determination of the technical capacity of the expander in the non-equilibrium expansion process;
3. Determination of the exergy efficiency of the process.

The following is a specific example of calculating and analyzing the adiabatic expansion of gas in a turboexpander:

Task 3.4. Nitrogen expands in an adiabatic turboexpander. The expansion process is non-equilibrium, the nitrogen consumption is $\dot{m}$ = 1 kg/s. Parameters of nitrogen at the expander inlet: T_1 = 300 K, P_1 = 5 bar, nitrogen pressure at the expander outlet P_2 = 1,5 bar, nitrogen state functions are calculated according to the truncated Bogolyubov-Mayer virial equation (Fig. 15).

To define:

1. Integral temperature effect of the expansion process ΔT_{12}, K.
2. Technical capacity of the expander, $\dot{W}_{12}^{\text{tex}}$, kW.
3. Exergy efficiency of the nitrogen expansion process in the expander, η_{ex}.

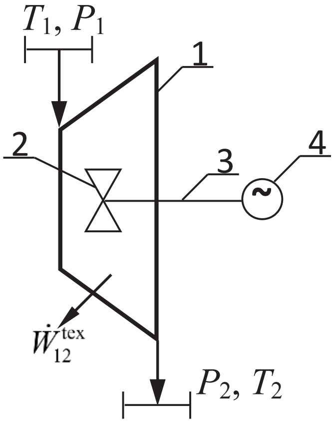


Figure 15. *Schematic diagram of the turboexpander stage:*

1 – expander stage body;
2 - rotating elements (blades) of the impeller;
3 - shaft;
4 - electric motor

The thermophysical characteristics and reference data of nitrogen are as follows (Annex, Table P.1):

$M = 28{,}013$ kg/kmol; $T_c = 126{,}2$ K; $P_c = 33{,}5$ atm; $\omega = 0{,}04$; virial coefficients b_{1j} of nitrogen:

$$b_{10} = 1{,}623 \cdot 10^{-3} \text{ m}^3\text{/kg};$$

$$b_{11} = -3{,}642 \cdot 10^{-3} \text{ m}^3\text{/kg};$$

$$b_{12} = -1{,}383 \cdot 10^{-3} \text{ m}^3\text{/kg};$$

$$b_{13} = -3{,}203 \cdot 10^{-4} \text{ m}^3\text{/kg}.$$

The heat capacity series constants $C_\text{p} = \sum_{i=0}^{n=3} d_i T^i$ have the following values (Annex, Table P.3):

$$d_0 = 1{,}113 \ \frac{\text{kJ}}{\text{kg}\cdot\text{K}};$$

$$d_1 = -4{,}846 \cdot 10^{-4} \ \frac{\text{kJ}}{\text{kg}\cdot\text{K}^2};$$

$$d_2 = 9{,}573 \cdot 10^{-7} \ \frac{\text{kJ}}{\text{kg}\cdot\text{K}^3};$$

$$d_3 = -4{,}173 \cdot 10^{-10} \ \frac{\text{kJ}}{\text{kg}\cdot\text{K}^4}.$$

1st stage: calculation of the integral effect of nitrogen cooling in an ideal equilibrium expansion process, $\Delta T_{12\text{S}}$, K.

The integral effect of nitrogen cooling of an ideal expansion process is determined by the equation:

$$\Delta T_{12\text{S}} = T_1 - T_{2\text{S}}, \tag{3.24}$$

where T_1 – nitrogen temperature at the expander inlet, K; $T_{2\text{S}}$ – nitrogen temperature at the end of the adiabatic equilibrium expansion process, K.

The temperature $T_{2\text{S}}$ is calculated based on the process condition $X = S = \text{const}$, according to the equation:

$$f(T_{2\text{S}}) = S_{2\text{S}} - S_1 = \int_{T_1}^{T_{2S}} \frac{C_{P\text{ид}}}{T} \text{d}T - R_\text{M} \ln \frac{P_2}{P_1} + \Delta S_{2\text{S}}^\text{д} - \Delta S_1^\text{д} = 0 \tag{3.25}$$

The zero approximation $T_{2\text{S}}^{(0)}$ is given from the condition that the isothermal entropy deviation from the ideal gas state of nitrogen $\Delta S_{2\text{S}}^\text{д} = 0$. The calculation of nitrogen density, isothermal deviations of enthalpy and entropy is carried out according to the Bogolyubov-Mayer equation in a truncated form.

The calculation results are presented in Table 3.12.

Table 3.12.

T_1, K	$B_1 \cdot 10^4$, m³/kg	$\rho_{1ии}$, kg/m³	$Z^{(0)}$	Z	ρ_1, kg/m³	$\Delta S_1^{д} \cdot 10^3$, $\frac{\text{kJ}}{\text{kg} \cdot \text{K}}$	$\Delta h_1^{д}$, $\frac{\text{kJ}}{\text{kg}}$
300	–1,778	5,616	0,999	0,999	5,622	–3,493	–1,137

$T_{2S}^{(0)}$ is calculated by the formula (3.25), provided $\Delta S_2^{д} = 0$.

$$T_{2S}^{(0)} = T_1 \exp\left[\frac{R_{М} \ln \frac{P_2}{P_1} + \Delta S_1^{д}}{C_p}\right] = 300 \ .\exp\left[\frac{296{,}791 \ln \frac{1{,}5}{5} - 3{,}493}{3{,}5 \cdot 296{,}791}\right] = 211{,}97 \ \text{K}$$

Further approximations T_{2S} are found from the calculated relation (3.25) by the method of successive approximations. The value is chosen T_{2S}, based on the condition:

$$\varepsilon_{T_{2s}} = \left|\frac{T_{2S}^{i} - T_{2S}^{i-1}}{T_{2S}^{i}}\right| \le 1 \cdot 10^{-2}$$

The calculation results are presented in Table. 3.13.

Table 3.13.

T_{2S}, K	$\rho_{2S\ ид}$, kg/m³	$B \cdot 10^3$, m³/kg	$Z^{(0)}$	Z	ρ_{2S}, kg/m³	$\int_{T_1}^{T_{2S}} \frac{C_{рид}}{T} \mathrm{d}T$, $\frac{\text{кJ}}{\text{kg} \cdot \text{K}}$	$\Delta S_{2S}^{д} \cdot 10^3$, $\frac{\text{кJ}}{\text{kg} \cdot \text{K}}$	$f(T_{2S}^{i}) \cdot 10^3$, $\frac{\text{кJ}}{\text{kg} \cdot \text{K}}$
211,97	2,384	–1,024	0,997	0,997	2,391	– 0,3631	–3,348	–5,635
212,65	2,377	–1,0925	0,997	0,997	2,384	– 0,3597	–2,478	–1,355

T_{2S} = 212,65 K is chosen with accuracy

$$\varepsilon_{T} = \left|\frac{212{,}65 - 211{,}97}{212{,}65}\right| = 0{,}322 \cdot 10^{-2}$$

The integral temperature effect of cooling of ΔT_{12S} of an ideal equilibrium expansion process is calculated by formula (3.24).

$$\Delta T_{12S} = T_1 - T_{2S} = 300 - 212{,}65 = 87{,}35 \text{ K}.$$

2^nd^ stage: the power of the expander is determined under the conditions of the adiabatic equilibrium process of nitrogen expansion.

The calculation of the power of the equilibrium adiabatic process of nitrogen expansion in the expander is carried out according to the relation:

$$\dot{W}^{\text{P}}_{12\text{S}} = \dot{m}W^{\text{P}}_{12\text{S}} = -\dot{m}\left[\int\limits_{T_1}^{T_{2\text{S}}} C_{pид}\,\text{d}T + (\Delta h^{\text{д}}_{2\text{S}} - \Delta h^{\text{д}}_{1})\right], \tag{3.26}$$

where $\dot{m}W^{\text{P}}_{12\text{S}} = \dot{W}^{\text{P}}_{12\text{S}}$ – power of equilibrium adiabatic (isoentropic) expansion of nitrogen in the expander, kW;

$\int\limits_{T_1}^{T_{2\text{S}}} C_{pид}\,\text{d}T = h_{2\text{S ид}} - h_{1\text{ ид}}$ – difference between the enthalpy values of nitrogen in terms of parameters at the inlet and outlet of the expander in an ideal gas state, $\frac{\text{kJ}}{\text{kg}}$;

$(\Delta h^{\text{д}}_{2\text{S}} - \Delta h^{\text{д}}_{1})$ – difference between the isothermal deviations of the nitrogen enthalpy from the ideal gas state in terms of gas parameters at the inlet and outlet of the expander working zone (stage), $\frac{\text{kJ}}{\text{kg}}$

The calculation results are presented in Table. 3.14.

Table 3.14.

T_1, K	$T_{2\text{S}}$, K	ρ_1, kg/m³	$\rho_{2\text{S}}$, kg/m³	$\int\limits_{T_1}^{T_{2S}} C_{pид}dT$, $\frac{\text{kJ}}{\text{kg}}$	$\Delta h^{\text{д}}_1$, $\frac{\text{kJ}}{\text{kg}}$;	$\Delta h^{\text{д}}_{2\text{S}}$, $\frac{\text{kJ}}{\text{kg}}$	$W^{\text{P}}_{12\text{S}}$, $\frac{\text{kJ}}{\text{kg}}$	$\dot{W}^{\text{P}}_{12\text{S}}$, kW
300	212,65	5,622	2,384	–91,287	–1,137	– 0,666	90,816	90,816

3^rd^ stage: calculation of the integral effect of nitrogen cooling in real, i.e. non-equilibrium expansion process in the expander, ΔT_{12}, K.

The determination of the integral effect of cooling is carried out according to the calculated ratio:

$$\Delta T_{12} = T_1 - T_2, \tag{3.27}$$

where T_1 – the nitrogen temperature at the expander inlet, K; T_2 – the nitrogen temperature at the expander outlet during the nonequilibrium expansion process, K; ΔT_{12} – the integral cooling effect, K.

The determination of the temperature value T_2 is carried out by the method of successive approximation based on the relationship:

$$\eta_{\text{S}} = \frac{\dot{W}_{12}^{\text{tex}}}{\dot{W}_{12S}^{P}} = \frac{\dot{m}\left[\int_{T_1}^{T_2} C_{\text{рид}} \mathrm{d}T + (\Delta h_2^{\text{д}} - \Delta h_1^{\text{д}})\right]}{\dot{m}\left[\int_{T_1}^{T_2} C_{\text{рид}} \mathrm{d}T + (\Delta h_{2\text{S}}^{\text{д}} - \Delta h_1^{\text{д}})\right]}, \tag{3.28}$$

where η_{S} – the adiabatic efficiency of the expander (η_{S} = 0,83); $\dot{m}$ – mass flow rate of nitrogen, kg/s; $\Delta h_1^{\text{д}}$, $\Delta h_{2\text{S}}^{\text{д}}$, $\Delta h_2^{\text{д}}$ – isothermal deviation of the nitrogen enthalpy value from the ideal gas state in terms of the parameters at the inlet and outlet of the expander, $\frac{\text{kJ}}{\text{kg}}$;

$$\int_{T_1}^{T_2} C_{\text{рид}} \mathrm{d}T = d_0(T_{2\text{S}} - T_1) + d_1\left(\frac{T_{2\text{S}}^2 - T_1^2}{2}\right) + d_2\left(\frac{T_{2\text{S}}^3 - T_1^3}{3}\right) + d_3\left(\frac{T_{2\text{S}}^4 - T_1^4}{4}\right)$$

The determination of the zero approximation of the temperature value $T_2^{(0)}$ is carried out according to the relation below, provided that $\Delta h_2^{\text{д}}$=0.

$$T_2^{(0)} = T_1 - \frac{1}{C_{\text{рид}}}[\eta_{\text{S}} \cdot W_{12\text{S}}^{\text{P}} - \Delta h_1^{\text{д}}], \tag{3.29}$$

where $\text{C}_{\text{рид}} = 3{,}5\, R_{\text{M}}$.

$$T_2^{(0)} = 300 - \frac{1}{3{,}5 \cdot 296{,}791 \cdot 10^{-3}}[0{,}83 \cdot 90{,}816 + 1{,}137] = 226{,}34\ \text{K}$$

Further calculations T_2^i are carried out with accuracy:

$$\varepsilon_T = \left|\frac{T_2^i - T_2^{i-1}}{T_2^i}\right| \le 1 \cdot 10^{-2}$$

according to the ratio:

$$T_2^i = T_1 - \frac{1}{\bar{C}_{\text{рид}}}[\eta_{\text{S}} \cdot W_{12\text{S}}^{\text{P}} - \Delta h_1^{\text{д}} + \Delta h_2^{\text{д}}], \tag{3.30}$$

where $\overline{C}_{\text{рид}} = \dfrac{\int\limits_{T_1}^{T_2^{(i-1)}} C_{\text{рид}} \mathrm{d}T}{T_2^{i-1} - T_1}$ – the average heat capacity within the temperature range from T_1 to T_2^{i-1}, $\dfrac{\text{kJ}}{kg \cdot \text{K}}$.

The results of calculating the temperature T_2 at the outlet of the expander are presented in table 3.15.

Table 3.15.

T_1, K	T_2^i, K	ρ_2, $\dfrac{\text{kg}}{\text{m}^3}$	$\overline{C}_p \Big\|_{T_1}^{T_2^i}$, $\dfrac{\text{kJ}}{\text{kg} \cdot \text{K}}$	Δh_1^{∂}, $\dfrac{\text{kJ}}{\text{kg}}$	$\Delta h_2^{\partial} \cdot 10^3$, $\dfrac{\text{kJ}}{\text{kg}}$;	W_{12S}^{P}, $\dfrac{\text{kJ}}{\text{kg}}$
300	226,34	2,237	1,0444	–1,137	–593,37	90,816
300	227,33	2,227	1,0444	–1,137	–588,98	90,816

The value T_2 = 227,33 K is selected with an accuracy of:

$$\varepsilon_{\text{T}} = \left| \frac{227{,}33 - 226{,}34}{227{,}33} \right| = 0{,}435 \cdot 10^{-2}.$$

The integral effect of cooling is the following value according to the formula (3.27):

$$\Delta T_{12} = T_1 - T_2 = 300 - 227{,}33 = 72{,}67\ \text{K}.$$

The actual power of the turboexpander in the real expansion process in the nominal operating mode is determined by the following ratio:

$$\dot{W}_{12}^{\text{tex}} = -\dot{m} \int\limits_{T_1}^{T_2} \mathrm{d}h = -\dot{m} \left[\int\limits_{T_1}^{T_2} C_{\text{рид}} \mathrm{d}T + (\Delta h_2^{\text{д}} - \Delta h_1^{\text{д}}) \right] \tag{3.31}$$

The calculation results are presented in Table 3.16.

Table 3.16.

T_1, K	T_2, K	Δh_1^{∂}, $\frac{\text{kJ}}{\text{kg}}$	$\Delta h_2^{\partial}\cdot 10^3$, $\frac{\text{kJ}}{\text{kg}}$;	$\int_{T_1}^{T_2} C_{pu\partial}\mathrm{d}T$, $\frac{\text{kJ}}{\text{kg}}$	W_{12}^{tex}, $\frac{\text{kJ}}{\text{kg}}$;	$\dot{W}_{12}^{tex}$, kW
300	227,33	–1,137	–588,98	–75,899	75,351	75,351

$$\dot{W}_{12}^{tex} = -1\cdot[-75{,}899 - 588{,}98\cdot 10^{-3} + 1{,}137] = 75{,}351\ \text{kW}$$

4th stage: calculation of kinetic energy losses due to friction in a non-equilibrium expansion process, i.e. evaluation of the viscous dissipation of the process $\dot{\Psi}_{12}^{\text{тр}}$, kW.

The calculation of kinetic energy losses due to friction in a non-equilibrium process is carried out according to the equation obtained on the basis of the integral equation for the balance of kinetic and potential energy of a non-equilibrium process:

$$\dot{\Psi}_{12}^{\text{тр}} = -\dot{W}_{12}^{\text{tex}} + \dot{m}\int_1^2 -\frac{\mathrm{d}P}{\rho}\ , \tag{3.32}$$

where $\dot{W}_{12}^{\text{tex}}$ =75,351 kW – technical power removed in the actual process of nitrogen expansion, i.e. taking into account friction forces;

$\dot{m}\int_1^2 -\frac{\mathrm{d}P}{\rho} = \dot{W}_{12\text{пол}}$ – power of the polytropic equilibrium process, calculated by the formula:

$$\dot{W}_{12\text{пол}} = \dot{m}\frac{\bar{n}}{\bar{n}-1}\frac{P_1}{\rho_1}\left[1-\left(\frac{P_2}{P_1}\right)^{\frac{\bar{n}-1}{n}}\right], \tag{3.33}$$

where $\bar{n} = \frac{\ln\frac{P_2}{P_1}}{\ln\frac{\rho_2}{\rho_1}}$ – average value of the polytropic index for the initial and final parameters characterizing the state of the gas at the inlet and outlet of the expander stage.

$$\dot{W}_{12\ \text{пол}} = \dot{m}\frac{\bar{n}}{\bar{n}-1}\frac{P_1}{\rho_1}\left[1-\left(\frac{P_2}{P_1}\right)^{\frac{\bar{n}-1}{\bar{n}}}\right] = 1\cdot\frac{1{,}3}{1{,}3-1}\cdot\frac{5\cdot 10^5}{5{,}622}\left[1-\left(\frac{1{,}5}{5}\right)^{\frac{1{,}3-1}{1{,}3}}\right] = 93{,}57\ \text{кВт}$$

The calculation results are presented in Table 3.17.

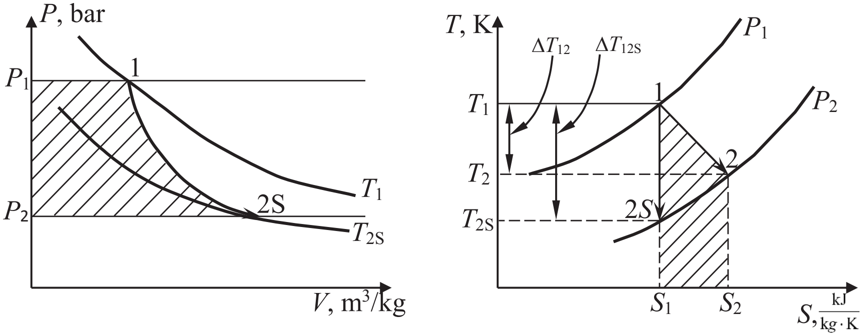
Table 3.17.

T_1, K	T_2, K	P_1, bar	P_2, bar	ρ_1, kg/m³	ρ_2, kg/m³	$n=\dfrac{\ln\frac{P_2}{P_1}}{\ln\frac{\rho_2}{\rho_1}}$	$\dot{W}_{12\ \text{пол}}$, kW	$\dot{W}_{12}^{tex}$, kW	$\dot{\Psi}_{12}^{\text{тр}}$, kW
300	227,33	5	1,5	5,622	2,227	1,3	93,57	75,351	18,219

The viscous dissipation of the nonequilibrium expansion process $\dot{\Psi}_{12}^{\text{тр}}$, according to the calculated relation (3.32), is the following value:

$$\dot{\Psi}_{12}^{\text{тр}}=-\dot{W}_{12}^{tex}+\dot{W}_{12\ \text{пол}}=-75{,}357+93{,}57=18{,}219\ \text{kW}$$

The polytropic equilibrium process of nitrogen expansion, carried out through the initial and final states, which completely coincide with the real non-equilibrium process, can be represented in the form of diagrams (*PV*), (*TS*) (fig. 16).



a b

Figure 16. *Polytropic equilibrium process of nitrogen expansion in PV (a), TS (b) diagrams*

5th stage: exergy analysis of the nitrogen expansion process in the expander.

The calculation of the exergy efficiency of the nitrogen expansion process is based on the calculated ratio, excluding transit exergy flows:

$$\eta_{ex}=\frac{\dot{Ex}_{\text{вых}}-\dot{Ex}_{\text{транзит}}}{\dot{Ex}_{\text{вх}}-\dot{Ex}_{\text{транзит}}},\qquad(3.34)$$

where $\dot{Ex}_{\text{вых}}$ – exergy flux at the output of the expander stage;

$\dot{E}x_{\text{вх}}$ – exergy flux at the inlet to the expander stage;

$\dot{E}x_{\text{транзит}}$ – transit flows of exergy, i.e. constituting that part of the exergy of incoming flows, which passes invariably through the entire apparatus.

The nitrogen exergy flux at the expander inlet is calculated by the relation:

$$\dot{E}x_{\text{вх}} = \dot{m}ex_1 = \dot{m}\left[\int_{T_0}^{T_1} C_{\text{рид}}dT + \Delta h_1^{\partial} - T_{\text{ос}}\left(\int_{T_0}^{T_1}\frac{C_{\text{рид}}}{T}dT - R_{\text{M}}\ln\frac{P_1}{P_0} + \Delta S_1^{\partial}\right)\right], \quad (3.35)$$

where $T_{\text{ос}}$ = 298,15 K – temperature value in the reference state;

$\int_{T_0}^{T_1} C_{\text{рид}}\mathrm{d}T = \sum_{i=0}^{n=3}\frac{d_i(T_1^{i+1} - T_0^{i+1})}{i+1}$ – change in the enthalpy of nitrogen in the ideal gas state, ;

$\int_{T_0}^{T_1}\frac{C_{\text{рид}}}{T}\mathrm{d}T$ – change in the entropy of an ideal gas depending on the change in temperature, $\frac{\text{kJ}}{\text{kg}\cdot\text{K}}$

Δh_1^{∂} – isothermal deviation of the nitrogen enthalpy from the ideal gas state according to the parameters at the expander inlet, $\frac{\text{kJ}}{\text{kg}}$;

ΔS_1^{∂} – isothermal deviation of the entropy of nitrogen from the ideal gas state in terms of the parameters T_1, P_1, $\frac{\text{kJ}}{\text{kg}\cdot\text{K}}$.

The calculation results $\dot{E}x_{\text{вх}}$ are presented in Table 3.18.

Table 3.18.

T_1, K	P_1, Bar	$\int_{T_0}^{T_1} C_{\text{рид}}\mathrm{d}T$, $\frac{\text{kJ}}{\text{kg}}$	$\int_{T_0}^{T_1}\frac{C_{\text{рид}}}{T}\mathrm{d}T\cdot 10^3$, $\frac{\text{kJ}}{\text{kg}\cdot\text{K}}$	Δh_1^{∂}, $\frac{\text{kJ}}{\text{kg}}$	$\Delta S_1^{\partial}\cdot 10^3$, $\frac{\text{kJ}}{\text{kg}\cdot\text{K}}$	ex_1, $\frac{\text{kJ}}{\text{kg}}$	$\dot{E}x_{\text{вх}}$, W
300	5	1,928	6,448	–1,137	–3,439	141,16	141,16

The exergy flow at the outlet is the sum of the exergy flows due to the movement of nitrogen $\dot{E}\text{x}_2$ and the flow of mechanical work $\dot{W}_{12}^{\text{tex}}$. According to the exergy balance equation, the calculated ratio for the exergy of nitrogen at the outlet of the expander is obtained:

$$\dot{E}x_{\text{вых}} = \dot{m}ex_2 + \dot{W}_{12}^{\text{tex}} = \dot{m}ex_1 - in\dot{D}_{12}, \tag{3.36}$$

where $\dot{m}ex_1 = 141{,}16$ кВт – nitrogen exergy flow at the expander inlet, kW;

$in\dot{D}_{12}$ – exergy loss due to irreversibility of the process, kW.

The calculation formula for the exergy efficiency η_{ex} taking into account relations (3.34), takes the form:

$$\eta_{ex} = \frac{\dot{E}x_{\text{вых}}}{\dot{E}x_{\text{вх}}} = \frac{\dot{m}ex_1 - in\dot{D}_{12}}{\dot{m}ex_1} = 1 - \frac{in\dot{D}_{12}}{\dot{m}ex_1}, \tag{3.37}$$

Exergy losses due to the irreversibility of the process are calculated using the Hui-Stodola formula:

$$in\dot{D}_{12} = \dot{m} \cdot T_{\text{o.c.}} \cdot \left[\int_{T_1}^{T_2} \frac{C_{рид}}{T} \mathrm{d}T - R_M \ln \frac{P_2}{P_1} + (\Delta S_2^{\partial} - \Delta S_1^{\partial}) \right]$$

The results of calculations of the exergy efficiency η_{ex} of the process are presented in Table 3.19.

Table 3.19.

T_1, K	T_2, K	$\int_{T_1}^{T_2} \frac{C_{рид}}{T} \mathrm{d}T \cdot 10^3$, $\frac{\text{kJ}}{\text{kg} \cdot \text{K}}$	$\Delta S_1^{\partial} \cdot 10^3$, $\frac{\text{kJ}}{\text{kg} \cdot \text{K}}$	$\Delta S_2^{\partial} \cdot 10^3$, $\frac{\text{kJ}}{\text{kg} \cdot \text{K}}$	$in\dot{D}_{12}$, kW	$\dot{m}ex_1$	η_{ex}
300	227,33	–289,7	–3,493	–2,021	20,602	141,16	0,85

$$\eta_{\text{ex}} = 1 - \frac{in\dot{D}_{12}}{\dot{m}ex_1} = 1 - \frac{20{,}602}{141{,}16} = 0{,}85.$$

To increase the exergy efficiency of the nitrogen expansion process, it is advisable to increase the efficiency of the expander, i.e. reduce exergy losses due to the irreversibility of the process itself. This is achieved by improving the gas dynamics of motion, namely by reducing friction losses in the gas path of the turboexpander.

3.2.3 Analysis of throttling processes

The problem of internal cooling can be solved using the throttling process. Throttling is an irreversible adiabatic process of reducing the pressure of a gas (steam) flow, when passing through a narrowed hole (throttle), and the working fluid does not perform external work. For such a process, the total energy of the flow remains unchanged. Let us consider this process during the outflow of a gas flow using the example of a diaphragm as a throttle device (Fig. 17).

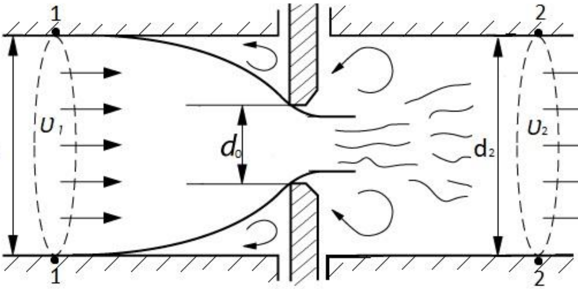


Figure 17. *Scheme of throttling the working flow along the channel profile*

In the figure, d_1, d_2 – the cross section of the channel before and after the diaphragm; d_0 – diaphragm (throttle) section of the channel.

Since throttling is carried out without external work $\dot{W}$ and without supply (removal) of heat flow $\dot{Q}$ when gas (steam) flows through local hydraulic resistance, then for sections (1–1) and (2–2) that are sufficiently remote from the throttle, where the values of the parameters can be assumed to be steady, is obtained from the total energy balance equation (2.2):

$$\frac{\partial}{\partial\tau}\int_0^V (K_V + \Pi_V + U_V)dV = \sum_{i=1}^{n}\dot{m}_i(\bar{h} + \frac{\bar{\upsilon}^2}{2} + \bar{\varphi}_g)_i - \dot{W}^{\text{tex}} + \dot{Q} + \dot{H}^{\text{диф}}$$

the following equation:

$$\dot{m}(h_1 + \frac{\bar{\upsilon}_1^2}{2} + \varphi_1) = \dot{m}(h_2 + \frac{\bar{\upsilon}_2^2}{2} + \varphi_2) \qquad (3.38)$$

This equation is a special case of the integral total energy balance equation.

Let's analyze the presented ratio. Since the mass flow rate of the working fluid in each section remains constant and the area of the flow section of the channel (1–1) does not change before and after local constriction (2–2), the flow velocities remain practically unchanged $\bar{\upsilon}_1 \approx \bar{\upsilon}_2$. Assuming the specific gravitational potential over the flow cross section to be constant $\varphi_1 = \varphi_2$, since the area of the control section is insignificant, the following integral throttling condition is obtained:

$$h_1 = h_2 \qquad (3.39)$$

In essence, this is a non-equilibrium process of poorly organized outflow in non-profiled channels, when the kinetic energy of the expanded gas is not converted into external work, as happens in expanders, but is transformed due to turbulent eddies into dissipation energy ψ, which, in turn, is used to restore enthalpy. For analysis, it is advisable to replace such an outflow process with a quasi-static irreversible process that has the same integral result of the constancy of enthalpy (3.39). In this case, however, it becomes possible to apply the analytical apparatus of classical equilibrium thermodynamics.

Task 3.5. Determine the integral effect of throttling ΔT_{12h}, K of nitrogen during the outflow of the flow through the diaphragmatic constriction of the pipeline (Fig. 17). The process is irreversible, adiabatic. Nitrogen parameters in the section (1–1), before the throttle device: $P_1 = 5$ bar, T_1=300 K, nitrogen parameters in the cross section and (2–2), after the throttle device: $P_2 = 3$ bar.

The functions of the state of nitrogen in the section (1–1) should be determined by the truncated virial equation of Bogolyubov – Mayer. In the section (2–2) it is permissible to use the ideal gas model.

Based on equation (3.39), taking into account the equation of state of the gas, the following expression is obtained:

$$c_p(T_1 - T_{2h}) + \Delta h_1^\partial - \Delta h_2^\partial = 0 \tag{3.40}$$

The integral throttling effect is determined by the relation:

$$\Delta T_{12h} = T_1 - T_{2h}, \tag{3.41}$$

where T_1, K – temperature of nitrogen in the section (1–1), before entering the diaphragmatic constriction; T_{2h}, K – the temperature of nitrogen in the section (2–2), after the diaphragmatic constriction, when the pipeline section becomes full again. The integral effect ΔT_{12h}, K can be found on the basis of the thermal equation of state for a nonideal gas. For an ideal gas, the throttling effect is zero. The calculation of T_{2h} should be carried out using equation (3.40) together with the truncated Bogolyubov-Mayer equation of state in virial form. From the above it follows:

$$T_{2h} = T_1 + \frac{\Delta h_1^\partial - \Delta h_2^\partial}{c_p}, \tag{3.42}$$

where Δh_1^∂, Δh_2^∂ – the isothermal deviation of the nitrogen enthalpy from the ideal gas state in terms of the parameters in the section (1–1) and (2–2), kJ/kg.

The change in gas temperature during throttling, due to the deviation from the ideal gas state, is called the Joule-Thompson effect. To reduce the load on the numerical apparatus and gain qualitative experience in the analysis of the problem described, it is proposed to calculate nitrogen in the cross section (2–2) according to the ideal gas equation. Therefore, the temperature value T_{2h} is determined by the following relation:

$$T_{2h} = T_1 + \frac{\Delta h_1^\partial}{c_p} \tag{3.43}$$

The results of the calculation are given in table 3.20.

Table 3.20.

T_1, K	$B_1 \cdot 10^4$, m^3/kg	$\rho_{1ид}$, kg/m^3	$Z^{(0)}$	Z	ρ_1, kg/m^3	$C_p(T_1)$, $\frac{\text{kJ}}{\text{kg} \cdot \text{K}}$	Δh_1^{∂}, $\frac{\text{kJ}}{\text{kg}}$
300	–1,778	5,616	0,999	0,999	5,622	1,0425	–1,137

The value of the gas temperature T_{2h} is found:

$$T_{2h} = 300 + \frac{-1{,}137}{1{,}0425} = 298{,}9 \text{ K}.$$

The intensity of temperature change, called the differential Joule-Thompson effect, is characterized as $a_h = (\partial T/\partial P)_h$. The value of this characteristic is determined from the energy balance of the closed system dh = TdS – VdP together with the Maxwell equation $(\partial S/\partial P)_T = - (\partial V/\partial T)_P$:

$$\alpha_h = \frac{T\left(\frac{\partial V}{\partial T}\right)_p - V}{C_p} = \alpha_S - \frac{V}{C_p}\,, \tag{3.44}$$

where a_S – differential cooling effect for an ideal reversible gas (steam) expansion.

Relationships for determining the magnitude of the differential and integral effects of gas cooling in the adiabatic equilibrium process of gas expansion from pressure P_1 to the final value P_2:

$$\alpha_S = \frac{k-1}{k} \cdot \frac{T_1}{P_1}, \tag{3.45a}$$

$$\Delta T_{12S} = T_1 \cdot \left[1 - \left(\frac{P_2}{P_1}\right)^{\frac{k-1}{k}}\right] \tag{3.45b}$$

Attention should be paid to the dependence of the value of the adiabatic exponent k in relation (3.45) on the thermal equation of state of the gas. The calculation of this quantity for a non-ideal gas, considered in the example of the model of the truncated virial Bogolyubov – Mayer equation, is carried out according to the formula:

$$k = \frac{1+2B\rho}{1+B\rho} \cdot \frac{C_p}{C_V}, \tag{3.4b}$$

where B – the second virial coefficient.

The calculation results are given in Table 3.21.

Table 3.21.

T_1, K	$B_1 \cdot 10^4$, m³/kg	ρ_1, kg/m³	$C_p(T_1)$, $\frac{\text{kJ}}{\text{kg} \cdot \text{K}}$	k	$\alpha_{\text{s}} \cdot 10^4$, $\frac{\text{K}}{Pa}$	$\alpha_{\text{h}} \cdot 10^6$, $\frac{\text{K}}{Pa}$
300	–1,778	5,622	1,0425	1,3966	1,717	1,078

Since $\alpha_S > 0$ and $\frac{V}{C_p} > 0$, then $\alpha_S > \alpha_{\text{h}}$, which is a consequence of the irreversibility of the process. Obviously, for an ideal gas, $\alpha_{\text{h}} = 0$. If the gas is not ideal and the value $\alpha_{\text{S}} < \frac{V}{C_p}$, then the throttling effect will be negative $\alpha_{\text{h}}<0$, i.e., the gas will heat up during expansion. If the gas is not ideal and $\alpha_{\text{S}} > \frac{V}{C_p}$, then the throttling effect will be positive $\alpha_{\text{h}}>0$, i.e., the expansion of the gas will be accompanied by cooling. The state of matter in which the differential effect changes sign is called the point of inversion. The set of such points forms an inversion line, the equation of which is as follows:

$$T(\partial V/\partial T)_P = V \tag{3.47}$$

An analysis of this relationship shows that in the *T–P* coordinates the inversion line has a maximum and limits the range of *T* and *P*, where $\alpha_{\text{h}} > 0$ and gas cooling occurs. For any pressure $P<P_{\text{max}} \approx 10\ P_{\text{кр}}$ there are two inversion temperatures $T_{\text{ин}}^{\text{в}}$ – the lower inversion temperature in the liquid region, $T_{\text{ин}}^{\text{в}}$ – the upper inversion temperature in the gaseous region. If $T_{\text{ин}}^{\text{н}} < T < T_{\text{ин}}^{\text{в}}$, then gas cooling will occur during throttling. Obviously, in order to use the substance as a working fluid of throttling refrigeration devices, it is necessary to cool the gas flow below $T_{\text{ин}}^{\text{в}}$ before throttling. Gases at low pressures and high temperatures are ineffective as refrigerants. For example, for air at temperature $T = 300$ K and pressure $P = 10$ bar, their ratio $\alpha_{\text{h}}/\alpha_{\text{S}} \approx 0{,}2$. However, as the temperature decreases and the pressure increases, the specific volume decreases and the ratio $\alpha_{\text{h}}/\alpha_{\text{S}}$ increases. For example, at temperature $T = 100$ K and pressure $P = 100$ bar for air, their ratio $\frac{\alpha_{\text{h}}}{\alpha_{\text{S}}} \approx 0{,}98$. Changes in the state of the throttled nitrogen flow are shown in fig. 18.

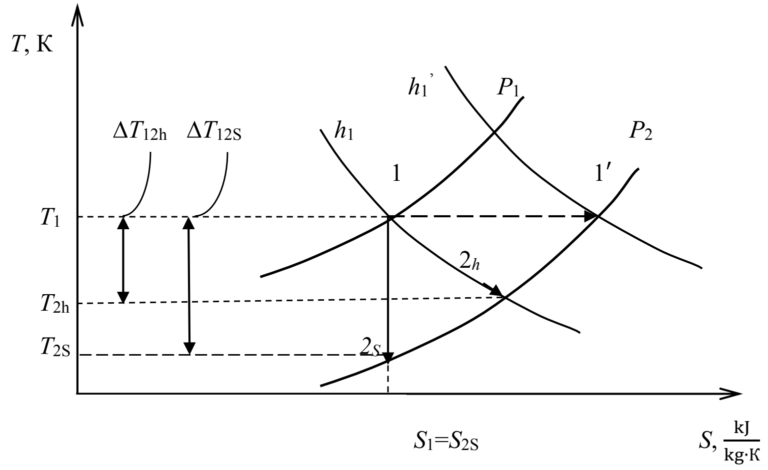


Figure 18. *Isothermal effect of Joule-Thompson throttling*

The choice of nitrogen as an object for a comparative analysis of methods for obtaining a cooling effect is due to the fact that this gas is the main component of atmospheric air and, in its properties in a wide range of thermal parameters, corresponds to the model of an ideal gas. The integral thermal effect characterizing the cooling capacity of the throttle device Q_x is determined by the following relationship:

$$Q_x = \int_{T_{2h}}^{T_1} C_p\, dT = \bar{C}_p (T_1 - T_{2h}) = h_1' - h_2 = h_1' - h_1 = -\Delta h_T, \quad (3.48)$$

where Δh_T – the isothermal effect of cooling (throttling).

Throttle refrigeration cycles are used in cryogenic technology for liquefying and separating low-temperature gases. The use of throttle devices allows you to control the parameters of gases (vapors), as well as their mixtures; determine and vary the consumption of technological objects in the investigated section of the equipment.

3.3 Analysis of heat transfer processes in an isolated TDS

The use of any method of thermodynamic analysis poses the problem of calculating the loss of expended work or exergy according to the Gouy-Stodola formula (3.8), which is reduced to calculating the entropy produced as a result of irreversibility. Let us consider this problem for an irreversible heat transfer process and heat losses to the environment as a result of process imperfections.

Task. 3.5. Let two bodies with different temperatures T_1 and T_2, and $\Delta T = (T_1 - T_2) > 0$ are exchanged through the surface dF by the elementary heat flux δQ. Considering the set of two interacting bodies as an isolated TDS, the entropy of which must increase during irreversible heat transfer, the growth of entropy within the process or the produced entropy is determined by the entropy balance equation (2.5):

$$\dot{m}(S_1 - S_2) + in\dot{S}_{12} = 0$$

Using thermodynamic laws and heat transfer equations, we obtain a relation for calculating the entropy produced (3.49):

$$in\dot{S}_{12} = \frac{K\Delta T_{12}^2 F}{T_1 T_2} \tag{3.49}$$

where: K – heat transfer coefficient, $\frac{\text{Вт}}{\text{м}^2\text{К}}$; F – the area of the heat exchange surface.

An analysis of relation (3.49) shows that the losses from irreversible heat transfer depend on the squared temperature difference and on the temperature zone of heat exchange. The smaller the product T_1T_2, the more significant the effect of the temperature difference in low-temperature aggregates. Usually in refrigeration plants ΔT does not exceed 7-10 °C, and even lower in the cryogenic area.

Let us estimate the amount of losses arising due to the imperfection of thermal insulation. In this case, from the environment to the cold elements of the installation with a temperature T, a heat flow is supplied that does not contain exergy $\delta Q_{\text{из}}$. Considering this process as an irreversible heat transfer at $\Delta T = (T_{\text{ср}} - T)$ and $T_1 = T_{\text{ср}}$, we find

$$\delta Q_{\text{из}} = T_{\text{ср}} \cdot \delta S_{\text{из}}^{\text{н}} = \delta Q_{\text{из}} \left(\frac{T_{\text{ср}}}{T} - 1\right) \tag{3.50}$$

The lower the temperature of the machines and apparatus of the refrigeration system, the greater the loss of exergy, the more perfect the thermal insulation should be. Cryogenic systems use high-vacuum and vacuum-powder insulation with the lowest values of effective thermal conductivity.

4. Examples and tasks

4.1 Examples

Task 4.1. H_2 enters the isobaric heat exchanger at a flow rate $\dot{m}_{H2} = 0.1\ ks/s$. Initial pressure of hydrogen $P_{1H2} = 3 \cdot 10^5$ Pa and temperature $T_1 = 448$ K. The heat flow removed from hydrogen in the heat exchanger is $Q = 145.5$ kW. After the heat exchanger, the gas enters the adiabatic compressor, where the pressure H_2 increases to $P_{3H_2} = 9 \cdot 10^5$ Pa. The exergy loss rate in the compressor is $inD_{ex} = 12.8$ кВт. The ambient temperature is $T_{oc} = 298$ K , the gas is calculated according to the MIG with a constant heat capacity, $c_p^0(T) = const.$

To define:

1) Draw up an installation plan.
2) Compressor technical power, $W_{23}^{тех}$, kW;
3) η_S – adiabatic compressor efficiency;
4) η_{ex} – exergy efficiency of the compressor;
5) T_2, K – gas temperature at the compressor inlet; T_3, K- gas temperature at the outlet of the compressor.

Solution:

We will draw up an installation scheme

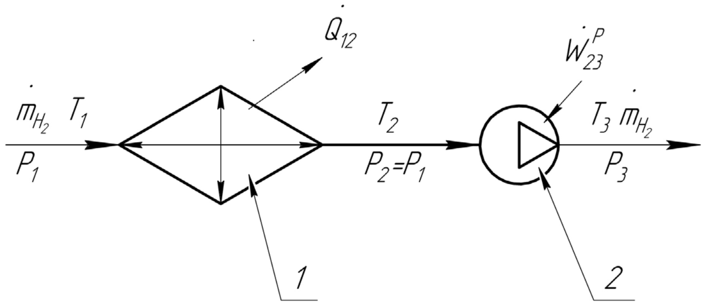


Figure 19. Technological site for preliminary preparation of raw materials

1 – heat exchanger;
2 – turbocharger;
$\dot{m}_{H_2}$ – flow H_2, entering the heat exchanger.
T_1, P_1 – temperature and pressure H_2 at the inlet from the heat exchanger;
T_2, $P_2 = P_1$ – temperature and pressure H_2 at the outlet of the heat exchanger;
T_3, P_3 – temperature and pressure H_2 at the outlet of the compressor;
$\dot{Q}_{12}$ – heat flow removed from H_2 in the heat exchanger.

I. Let us represent the total energy balance equations (2.2) for the compression zone

$$\frac{\partial Э}{\partial \tau} = \sum_i \dot{m}_i \left(h_i + \frac{\bar{v}_i^2}{2} + g\varphi_i \right) + \dot{Q} + \dot{H} - \dot{W}^{tex} = 0$$

Let us analyze equation (2.2) for the considered example:

a) Components $\frac{m_i v_i^2}{2} = 0, \quad \sum_i m_i g\varphi_i = 0$ – since there is no change in kinetic and potential energy;

b) $\dot{Q} = 0$ compressor is adiabatic, there is no heat exchange with the environment;

c) $\dot{H} = 0$ – there is no diffusion flow, no mass transfer transformations occur

Therefore, equation (2.2) takes the following form:

$\dot{m}(h_2 - h_3) - \dot{W}_{23}^{tex} = 0,$ whence the power consumed by the compressor is equal to the loss of gas enthalpy flow:

$$\dot{W}_{23}^{tex} = \dot{m}(h_2 - h_3) = \dot{m}c_p^0(T_2 - T_3)$$

where:

T_2 – gas temperature H_2 at the compressor inlet, K.

T_3 – gas temperature H_2 at the outlet of the compressor, K.

$Cp_{H_2}^0$ = 3.5 *Rm* (MIG - model of an ideal gas,i.e.the number of degrees of freedom H_2 is 5 (j=5),because it is a diatomic gas).

II. To determine the temperature T_2 we compose the total energy balance equation (2.2) for the heat exchange zone:

$$\frac{\partial Э}{\partial \tau} = \sum_i^n \dot{m}_i \left(h_i + \frac{\bar{v}_i^2}{2} + g\varphi_i \right) + \dot{Q} + \dot{H}^{ml} - \dot{W}^{tex} = 0$$

a. $\dot{H} = 0$ since there is no heat transfer;

b. $\dot{W}^{tex} = 0$ – since the technical work in the heat exchanger is not consumed, since the process is isobaric;

c. $\sum_i^n m_i \frac{\bar{v}_i^2}{2} = 0, \sum_i^n m_i g\varphi_i = 0$ – there is no change in kinetic and potential energy;

d. $\dot{Q} = -145.5$ kW, since the heat flow is removed from H_2.

Taking into account the above remarks, we obtain the expression for the total energy balance:

$$\dot{m}(h_2 - h_1) + \dot{Q}_{12} = 0$$

$$\dot{m}c_p^0(T_2 - T_1) = -\dot{Q}_{12}$$

$$T_2 = T_1 - \frac{\dot{Q}_{12}}{\dot{m} \cdot \mathrm{c}_p^0} = T_1 - \frac{\dot{Q}_{12}}{\dot{m} \cdot 2.5R_M}$$

$$R_{M_{H2}} = \frac{\tilde{R}}{MM_{H2}} = \frac{8314}{2.016} = 4.124 \frac{\mathrm{kJ}}{\mathrm{kg} \cdot \mathrm{К}}$$

$$Cp_{H_2}^0 = 3.5\, R_M = 3.5 \cdot 4.124 = 14.434 \frac{\mathrm{kJ}}{\mathrm{kg} \cdot \mathrm{К}}$$

$$MM_{H_2} = 2.016 \frac{\mathrm{kg}}{\mathrm{kmol}}$$

Substituting the data for section 1 (heat exchange) of the technological scheme for T_1 and $\dot{Q}_{12}$, we obtain the value of the temperature H_2 at the outlet from the heat exchange zone:

a. $T_2 = 448 - \frac{145.5}{0.1 \cdot 14.434} = 347.19$ К.

Now we determine the temperature H_2 at the outlet of the adiabatic compressor stage, T_{3S}.

б. $T_{3S} = T_2 \left(\frac{p_3}{p_2}\right)^{\frac{k-1}{k}} = 347.196 \left(\frac{9 \cdot 10^5}{3 \cdot 10^5}\right)^{\frac{1,4-1}{1,4}} = 475.21$ К,

where $k = 1.4$ (adiabatic exponent for 2^x atomic gases)

In order to determine the actual gas temperature at the compressor outlet T_3, we write the entropy balance equation (2.5) for the gas compressed in the turbocharger stage:

$$\frac{\partial S}{\partial \tau} = \sum_{i=1}^{n} m_i \bar{S}_i + \dot{S}^Q + \dot{S}^{mc} + in\dot{S}_{23} = 0$$

Let's analyze:

a. Since the compression in the compressor is carried out in the adiabatic mode $\dot{S}^Q = 0$, i.e. there is no change in entropy due to heat transfer.

б. $\dot{S}^{mc} = 0$ – there is no mass transfer in the compressor, i.e. there is no change in entropy due to diffusion.

As a result of the above,expression (2.5) takes the following form:

$$\dot{m}(S_2 - S_3) + in\dot{S}_{23} = 0$$

Whence the entropy growth rate due to irreversibility for the compression zone can be determined by the following formula:

$$in\dot{S}_{23} = \dot{m}\left(\dot{C}_P ln \frac{T_3}{T_2} - R_M \ln \frac{p_3}{p_2}\right)$$

Since the exergy loss rate in the compressor stage is $\dot{D}_{23} = 12.8\mathrm{kW}$, and the calculation formula for $in\dot{D}_{23}$ is as follows:

$$\dot{\imath n D}_{23} = T_{\text{o.c.}}\dot{m}\left(C_P ln\frac{T_3}{T_2} - R_M \ln\frac{p_3}{p_2}\right),$$

The gas temperature in the actual compression process T_3 can be determined by the expression:

$$C_P ln\frac{T_3}{T_2} = \frac{in\dot{D}_{23}}{T_{\text{o.c.}}\dot{m}} + R_M \ln\frac{p_3}{p_2} = \frac{12.8}{298 \cdot 0.1} + 4.124 \ln\left(\frac{9}{3}\right) = 4.9605\frac{\text{kJ}}{\text{kg} \cdot \text{К}}$$

$$\ln\frac{T_3}{T_2} = \frac{4.9605}{14.434} = 0.3437$$

Final answer T_3 = 489.54 К – temperature H_2 at the compressor outlet.

We determine the adiabatic efficiency of the compressor stage (3.1):

$$\eta_S = \frac{\dot{W}_{23S}^{TK}}{\dot{W}_{23}^{\text{вн}}} = \frac{\dot{m}C_p^0(T_2 - T_{3s})}{\dot{m}C_p^0(T_2 - T_3)}$$

$$\eta_S = \frac{T_2 - T_{3s}}{T_2 - T_3} = \frac{347.196 - 475.213}{347.196 - 489.546} = 0.899$$

III. We determine the exergy efficiency of the compressor stage:

$$\eta_{ex} = \frac{(\dot{E}_{x\ 3} - \dot{E}_{x\ 2})}{\dot{E}_{x2}\ - \dot{W}_{23}^{\text{вн}} - \dot{E}_{x2}}$$

$$\eta_{ex} = \frac{\dot{m}(e_{x\text{вых}} - e_{x\text{тр}})}{\dot{m}(e_{x\text{вх}} - e_{x\text{тр}})},$$

where:

$e_{\text{хвых}} = e_{\text{х}3}$ – specific exergy at the compressor outlet

$e_{\text{тр}} = me_{\text{х}2}$ – transit exergy H_2; i.e. that part of the input exergy with which the gas enters the compressor and passes through the compressor zone unchanged.

$$\eta_{ex} = \frac{\dot{m}(e_{x3} - e_{x2})}{-\dot{W}_{13}^{\text{вн}}} = \frac{\dot{m}((h_3 - h_2) - T_{\text{o.c.}}(S_3 - S_2))}{-\dot{W}_{23}^{\text{вн}}}$$

$$\eta_{ex} = \frac{\dot{m}\left(C_P(T_3 - T_2) - T_{0.e.}\left(C_p\ \ \ln\frac{T_3}{T_2} - R_M \ln\frac{p_3}{p_2}\right)\right)}{-\dot{m}C_p\ (T_3 - T_2)}$$

$$\eta_{ex} = 1 - \frac{\dot{\imath n D}_{ex23}}{-\dot{W}_{23}^{\text{вн}}} = 1 - \frac{12.8}{0.1 \cdot 14.434(488.546 - 347.146)} = 0.937$$

IV. The technical power of the compressor $\dot{W}_{23}^{\text{тех}}$, kW is determined by the following relation:

$$\dot{W}_{23}^{\text{тех}} = \dot{m}C_p\ (T_2 - T_3) = 0.1 \cdot 14.434 \cdot (347.196 - 489.546) == -20.402\ \text{кВт}$$

Task 4.2.

In an isobaric heat exchanger, a heat flow of 1600 kW is supplied to the air. Air consumption 4 kg/s. Heat exchanger pressure 5·10^5 Pa. After the heat exchanger, the air enters the adiabatic turbine, where it expands to 1·10^5 Pa. The air temperature after the turbine is 693 K. The exergy loss rate in the turbine is 150 kW. Ambient temperature $T_{\text{o.c.}}$ = 298 K. Air according to MIGS with $C_p^0(T) = const.$

To define:

1) To draw up an installation diagram;
2) Heat exchanger air temperature T_1, K;
3) $\dot{W}_{23}^{тех}$, kW – technical power of the turbine kW;
4) η_s, η_{ex} – adiabatic, exergy efficiency of the turbine;
5) $in\dot{S}_{23}$, $\frac{\text{kW}}{\text{K}}$ – rate of entropy production in the turbine.
6) $in\dot{D}_{23}$, kW – exergy loss due to irreversibility of the process

Solution:

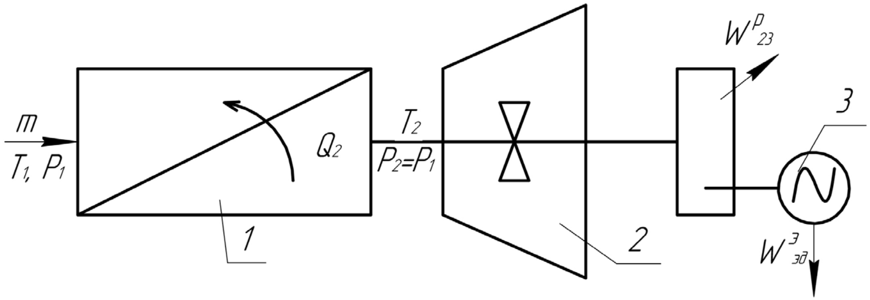


Figure 20. *Technological section*

1-isobaric heat exchanger; 2 - turbine stage; 3- electric generator

1.) Let's write the total energy balance equation for the heat exchanger:

$$\frac{\partial Э}{\partial \tau} = \sum_i^n \dot{m}_i \left(h_i + \frac{\bar{v}_i^2}{2} + g\varphi_i \right) + \dot{Q} + \dot{H}^{ml} - \dot{W}^{tex} = 0$$

$$\dot{H} = 0; \quad \dot{W}^{tex} = 0; \quad \sum_i^n m_i \frac{\bar{v}_i^2}{2} = 0; \quad \sum_i^n m_i g\varphi_i = 0$$

$$\dot{m} C_p^0 (T_1 - T_2) + \dot{Q}_{12} = 0$$

$$\Delta T_{12} = T_2 - T_1 = \frac{\dot{Q}_{12}}{\dot{m} C_p^0} = \frac{1600}{4 * 1.29} = 310{,}08\text{К}$$

where C_p^0 в − х $= 4.5 \cdot R_M = 4.5 \cdot \frac{8.314}{29} = 1.29 \frac{\text{kJ}}{\text{kg} \cdot \text{К}}$

$\Delta T_{12} = -310.08$ К

$T_1 = T_2 - \Delta T_{12}$

1) $\iota n\dot{D}_{23} = T_{\text{o.c.}} \cdot in\dot{S}_{23}$

$in\dot{S}_{23} = \frac{150}{298} = 0.503355 = 503.355 \frac{\text{Вт}}{\text{К}}$ − entropy production rate

In the turbine:

$$in\,\dot{S}_{23} = \dot{m}\left[C_p^0 \ln\frac{T_3}{T_2} - R_M \ln\frac{p_3}{p_2}\right]$$

where T_3, p_3 − pressure and air temperature at the outlet of the turbine; $T_2; p_2$ −pressure and air temperature at the turbine inlet.

$$C_p^0 \ln\frac{T_3}{T_2} - R_M \ln\frac{p_3}{p_2} = \frac{in\dot{S}_{23}}{\dot{m}},$$

$$C_p^0 \ln\frac{T_3}{T_2} = \frac{in\dot{S}_{23}}{\dot{m}} + R_M \ln\frac{p_3}{p_2},$$

finally we have:

$$\ln\frac{T_3}{T_2} = \frac{in\dot{S}_{23} + \dot{m}R_M \ln\frac{p_3}{p_2}}{\dot{m}C_p^0}$$

$$\ln\frac{T_3}{T_2} = \frac{0.503 + 40.287 \ln\frac{1}{5}}{4 \cdot 1.29} = -0.2606$$

$$\frac{T_3}{T_2} = 0.7706$$

$T_2 = \frac{T_3}{0.7706} = \frac{693}{0.7706} = 899.3$ К − air temperature at the inlet to the expansion turbine, i.e. at the outlet of the heat exchanger, so

$$T_1 = T_2 + \Delta T_{12} = 899.3 - 310.08 = 589.22 \text{ К}$$

2) Let's define η_S:

$$\eta = \frac{\dot{W}_{23}^{\text{вн}}}{W_{23s}} = \frac{\dot{m}C_p^0(T_2 - T_3)}{\dot{m}C_p^0(T_2 - T_{3s})} = \frac{T_2 - T_3}{T_2 - T_{3s}}$$

Therefore, it is necessary to calculate

T_{3S} – air temperature at the outlet of the turbine in the equilibrium adiabatic process of air expansion:

$$T_{3s} = T_2 \left(\frac{p_3}{p_2}\right)^{\frac{k-1}{k}} = 899.3 \left(\frac{1}{5}\right)^{\frac{1.29-1}{1.29}},$$

where $k = 1.29$ (because the number of degrees of freedom of air is $j = 7$)

$$T_{3s} = 626.29 \text{ К},$$

$$\eta_s = \frac{T_2 - T_3}{T_2 - T_{3s}} = \frac{899.3 - 693}{899.3 - 626.29} = 0.756$$

$$\eta_{ex} = 1 - \frac{\dot{\imath n D}_{23}}{\dot{W}_{23}} = 1 - \frac{150}{4 \cdot 1.29(899.3 - 693)} = 0.86,$$

where $\dot{W}_{p23} = \dot{m} C_p^0 (T_2 - T_3) = 4 \cdot 1.29(899.3 - 693) = 1064.51$ kW –technical power of the turbine.

Task 4.3.

He (helium) flows in a horizontal pipe (d = 0,05 m) MM_{He} = 4.003 kg/mol initial **He** parameters at the pipe inlet:

$$T_1 = 298 \text{ К}; \ p_1 = 1.2 \cdot 10^5 \text{ Pa}$$

Final parameters:

$$T_2 = 473 \text{ К}; \ p_2 = 1.0 \cdot 10^5 \text{ Pa}$$

(pressure drops due to hydraulic resistance)

Mass flow He: $\dot{m}_{He} = 0.1$ kg⁄s

$$T_{\text{о.с.}} = 298 \text{ К (ambient temperature)}$$

To calculate the gas according to MIG with constant heat capacity.

To define:

1) $\eta_{\text{пол}}$ – polytropic efficiency of the process;
2) exergy loss rate $\dot{\imath n D}_{12}$ (кВт) due to irreversibility of the process;
3) dissipation energy $\dot{\psi}_{12}$ (kJ⁄kg);
4) heat flux supplied to the gas $\dot{Q}_{12}$ (kW).

Solution:

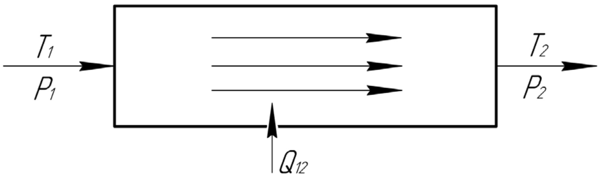


Figure 21. *Section of a horizontal pipe of constant diameter*

I. Let's make the equation of balance of kinetic energy:

$$\frac{\partial}{\partial \tau}(K+\pi)=\sum_{i=1}^{n}\dot{m}_i\left(\frac{v_i^2}{2}+gh\right)-\dot{W}_{12}^{tex}+\dot{m}\int_{1}^{2n}\frac{-dp}{\rho}-\dot{\psi}_\sigma=0$$

Let's analyze these equations:

$\dot{W}_{12}^{tex}=0$ – no technical work;

$\Delta\dot{m}gh$ – there is no change in potential energy;

$$\dot{m}\int_{1}^{2n}\frac{-dp}{\rho}=\dot{m}\frac{n}{n-1}\frac{p_1}{\rho_1}\left[1-\left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}}\right]-$$

polytropic power i.e. simulated equilibrium process,

where $n=\dfrac{\ln\left({}^{p_2}/_{p_1}\right)}{\ln\left({}^{\rho_2}/_{\rho_1}\right)}$;

$$\rho_1=\frac{p_1}{R_M T_1}=0.194\frac{\text{kg}}{\text{m}^3};$$

$$\rho_2=\frac{p_2}{R_M T_2}=0.102\frac{\text{kg}}{\text{m}^3};$$

where $n=\frac{\ln\left({}^{1}/_{1.2}\right)}{\ln\left({}^{0.102}/_{0.194}\right)}=0.283.$

The power of the polytropic pressure change process:

$$\dot{W}_{12}^{\text{пол}}=\dot{m}\frac{n}{n-1}\frac{p_1}{\rho_1}\left[1-\left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}}\right]$$

$$\dot{W}_{12}^{\text{пол}}=0.1\frac{0.283}{0.283-1}\frac{1.2\cdot 10^5}{0.194}\left[1-\left(\frac{1}{1.2}\right)^{\frac{0.283-1}{0.283}}\right]=14.335\text{ kW}$$

II. Loss of kinetic energy due to friction:

$$\dot{\psi}_\sigma=\dot{m}\frac{n}{n-1}\frac{p_1}{\rho_1}\left[1-\left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}}\right]-\dot{m}\left(\frac{v_2^2}{2}-\frac{v_1^2}{2}\right);$$

where v_1, v_2 – the He speed at the pipe inlet and outlet.

$$v_1=\frac{4\cdot\dot{m}}{\pi d_1^2\rho_1}=\frac{4\cdot 0.1}{3.14\cdot(0.05)^2\cdot 0.194}=262.66\text{ m/s}$$

$$v_2 = \frac{4 \cdot \dot{m}}{\pi d_2^2 \rho_2} = \frac{4 \cdot 0.1}{3.14 \cdot (0.05)^2 \cdot 0.102} = 499.56 \text{ m/s}$$

$$\dot{\psi}_\sigma = 0.1 \cdot 143.35 - 0.1 \cdot 10^{-3} \left(\frac{499.56^2}{2} - \frac{262.66^2}{2} \right)$$

$\dot{\psi}_\sigma = 5.307 \text{ kW}$ – kinetic energy leakage for friction.

III. We determine the polytropic efficiency of the gas flow.

$$\eta_{\text{пол}} = \frac{\dot{W}_{12}^{\text{необр}}}{\dot{W}_{12}^{\text{обр}}} = \frac{\dot{m} \frac{n}{n-1} \frac{p_1}{\rho_1} \left[1 - \left(\frac{p_2}{p_1} \right)^{\frac{n-1}{n}} \right] - \dot{\psi}_\sigma}{\dot{m} \frac{n}{n-1} \frac{p_1}{\rho_1} \left[1 - \left(\frac{p_2}{p_1} \right)^{\frac{n-1}{n}} \right]}$$

$$\eta_{\text{пол}} = \frac{14.335 - 5.307}{14.335} = 0.63$$

IV. $\dot{Q}_{12}$ – heat flux supplied to the gas.
Let us write the total energy balance equation for the pipe:

$$\frac{\partial Э}{\partial \tau} = \sum_i \dot{m}_i \left(h_i + \frac{\bar{v}_i^2}{2} + g\varphi_i \right) + \dot{Q}_{12} + \dot{H} - \dot{W}_{12}^{tex} = 0$$

$$\dot{m}(h_1 - h_2) + \dot{m} \left(\frac{v_2^2}{2} - \frac{v_1^2}{2} \right) + \dot{Q}_{12} = 0$$

$$\dot{Q}_{12} = \dot{m}(h_2 - h_1) + \dot{m} \left(\frac{v_2^2}{2} - \frac{v_1^2}{2} \right)$$

$$\dot{Q}_{12} = \dot{m} C_p^0 (T_2 - T_1) + \dot{m} \left(\frac{v_2^2}{2} - \frac{v_1^2}{2} \right)$$

$$\dot{Q}_{12} = 9.028 + 90.822 = 99.85 \text{ kW.}$$

V. $\iota\dot{n}D_{12} = \frac{T_{\text{о.с.}}}{T\text{тд}} * \dot{\psi}_\sigma = \frac{298}{378.8} * 5.307 = 4.17\text{кВт}$, where $\text{Ттд} = \frac{\text{Т1} - \text{Т2}}{ln\frac{\text{Т1}}{\text{Т2}}} = 378.8 \text{ К}$ – average thermodynamic temperature of the gas flow process.

Task 4.4. To determine the dissipation of kinetic energy during the movement of ammonia in the heat exchanger, ammonia moves from top to bottom with a flow rate of $\dot{m}$ = 0,05 kg/s. The cross-sectional diameter of the inlet pipe D_1 = 0,031 м, outlet pipe D_2 = 0,04 m, the difference in the levels of the location of the pipes is 6 m. Parameters of ammonia at the inlet to the heat exchanger; T_1 = 306 K; P_1 = 1,5 bar; parameters at the outlet of the heat exchanger T_2 = 300 K; P_2 = 1,3 bar.

Solution:

We find the dissipation of kinetic energy from the balance equation of kinetic and potential energy in the integral form (4.1):

$$\dot m\left(\frac{\upsilon_1^2}{2}-\frac{\upsilon_2^2}{2}\right)+\dot m g\left(z_1-z_2\right)+\int_{P_1}^{P_2}\dot m-\frac{dP}{\rho}=\dot\Psi_{\text{тр}}\,.$$

To calculate the kinetic energy sink functional, we use the model of an equilibrium polytropic process, the initial and final states of which completely coincide with the real process:

$$\int_{P_1}^{P_2}\dot m-\frac{dP}{\rho}=\dot m\frac{n}{n-1}\frac{P_1}{\rho_1}\left[1-\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}}\right], \tag{1.31}$$

where $n=\dfrac{\ln\frac{P_2}{P_1}}{\ln\frac{\rho_2}{\rho_1}}$ – average value of the polytropic index for the initial and final parameters characterizing the state of the gas in the real process.

To determine the density of ammonia, we use the equation of state of an ideal gas, due to a slight deviation of the parameters from standard conditions:

$$\rho_1=\frac{P_1}{R_M T_1}=\frac{1{,}5\cdot 10^5}{488{,}17\cdot 306}=1{,}0041\ \text{kg/m}^3;$$

$$\rho_2=\frac{P_2}{R_M T_2}=\frac{1{,}3\cdot 10^5}{488{,}17\cdot 300}=0{,}887\ \text{kg/m}^3$$

Let's determine the value of the polytropic index: $\bar n$:

$$\bar n=\frac{\ln\dfrac{P_2}{P_1}}{\ln\dfrac{\rho_2}{\rho_1}}=\frac{\ln\dfrac{1{,}3}{1{,}5}}{\ln\dfrac{0{,}887}{1{,}0041}}=1{,}154$$

Let's calculate the kinetic energy sink functional:

$$\dot m\int_{P_1}^{P_2}-\frac{dP}{\rho}=\dot m\frac{n}{n-1}\frac{P_1}{\rho_1}\left[1-\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}}\right]=0{,}05\frac{1{,}154}{1{,}154-1}\cdot\frac{1{,}5\cdot 10^5}{1{,}0041}\cdot$$

$$\cdot\left[1-\left(\frac{1{,}3}{1{,}5}\right)^{\frac{1{,}154-1}{1{,}154}}\right]=1{,}0583\ \text{k}W.$$

The gas velocities at the inlet and outlet of the heat exchanger are determined by the equation:

$$\dot m=\frac{\pi D_1^2}{4}\rho_1\upsilon_1=\frac{\pi D_2^2}{4}\rho_2\upsilon_2\,,$$

where: υ_1 – the gas velocity at the inlet to the heat exchanger, m/s;

υ_2 – gas velocity at the outlet from the heat exchanger, m/s.

$$\upsilon_1 = \frac{4\dot{m}}{\pi D_1^2 \rho_1} = \frac{4 \cdot 0{,}05}{3{,}14 \cdot 0{,}031^2 \cdot 1{,}0041} = 66{,}01 \text{ m/s},$$

$$\upsilon_2 = \frac{4\dot{m}}{\pi D_2^2 \rho_2} = \frac{4 \cdot 0{,}05}{3{,}14 \cdot 0{,}04^2 \cdot 0{,}887} = 44{,}88 \text{ m/s}.$$

Let's determine the dissipation of kinetic energy:

$$\dot{\Psi}_{12}^{\text{тр}} = 0{,}05\left[\left(\frac{66{,}01^2}{2} - \frac{44{,}88^2}{2}\right) + 9{,}8 \cdot 6\right] + 1058{,}3 = 1{,}12 \text{ kW}.$$

Task 4.5. In a cooled compressor, N_2 is compressed in the state of an ideal gas with a constant heat capacity. Compressor initial gas parameters T_1 = 293 К, P_1 = 1·10^5 Pa, gas mass flow rate $\dot{m}_{N2}$ = 1 kg⁄s. The compressor gas pressure is P_2=5·10^5 Pa. The compressor is cooled by circulating water entering through the compressor jacket, cooling water consumption $\dot{m}_{\mathrm{H_2O}}$ = 3.58 kg⁄s, $C_P^{H_2O} = 4.19 \frac{\text{kJ}}{\text{kg}\cdot\text{К}}$. The compressor temperature of the chilled water increases by 10 К. The technical work of the compressor is $W_{12}^{tex} = -200 \frac{\text{кДж}}{\text{кг}}$. The compression process is non-equilibrium.

To define:

1) $\boldsymbol{T_2}$ – temperature $\boldsymbol{N_2}$ at the outlet from the cooled compressor, К;
2) $\dot{\psi}_{12}$ – dissipation of kinetic energy into friction,kW;
3) $\eta_{\text{пол}}$ – polytropic efficiency of the process;
4) To present (PV), (TS) the charts.
5) $\boldsymbol{\eta_{ex}}$ – exergy efficiency of the process;

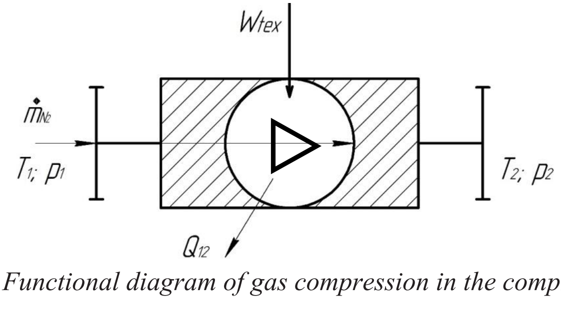


Figure 22. *Functional diagram of gas compression in the compressor stage*

Solution:

1) Let's determine the final temperature of nitrogen N_2 at the outlet of the cooling stage of the compressor T_2, К.

Let us compose the total energy balance equation (2.2) for the zone of the cooled stage with respect to nitrogen. When analyzing the compression process, we do not take into account changes in the kinetic and potential energy of the gas.

$$\dot{m}_{N_2}(h_1 - h_2) - \dot{W}_{12}^{tex} - \dot{Q}_{12} = 0,$$

where: $\dot{W}_{12}^{tex} = -200$ kW,, the available work flow supplied to the gas in the nominal operating mode.

$\dot{Q}_{12} = \dot{m}_{H2O} \cdot Cp_{H2O} \cdot \Delta T_{H2O} = -150$ kW – the heat flow removed from the gas in the cooling system.

We express the value of the final temperature of nitrogen at the outlet of the compressor stage:

$$T_2 = T_1 - \frac{Q_{12} + \dot{W}_{12}^{tex}}{\dot{m}_{N_2} \cdot Cp_{N_2}}$$

$$T_2 = 293 - \frac{150 + (-200)}{1 \cdot 1{,}038} = 341{,}17\ \text{К}$$

2) Let's determine the dissipation of kinetic energy on friction $\dot{\Psi}_{12}^{\text{тр}}$, kW.

Calculation of dissipation of kinetic energy on friction $\dot{\Psi}_{12}^{\text{тр}}$, k$W$.

We estimate dissipation using the balance equation for kinetic and potential energy for a nonequilibrium process (2.4).

$$\dot{\Psi}_{12}^{\text{тр}} = -\dot{W}_{12_{\text{вн}}} + \dot{m}\int_{T_1}^{T_{2n}} -\frac{dP}{\rho},$$

where $\dot{W}_{12}^{tex} = -200$ kW – internal power of the nonequilibrium process;

$\dot{m}\int_{T_1}^{T_{2n}} -\frac{dP}{\rho} = \dot{m}\frac{\bar{n}}{\bar{n}-1} \cdot \frac{P_1}{\rho_1}\left[1 - \left(\frac{P_2}{P_1}\right)^{\frac{\bar{n}-1}{\bar{n}}}\right]$ – the power of the equilibrium polytropic process, kW,

$\bar{n} = \dfrac{\ln\frac{P_2}{P_1}}{\ln\frac{\rho_{2_n}}{\rho_1}}$ – average value of the polytropic index for the initial and final parameters characterizing the state of the gas at the inlet and outlet of the compressor.

The results of calculations of viscous dissipation $\dot{\Psi}_{12}^{\text{тр}}$, k$W$ are presented in Table 4.1.

Table 4.1.

Analysis of the polytropic equilibrium process of nitrogen compression

ρ_1, kg/m³	T_2, K	$\rho_{2\bar{n}}$, kg/m³	$\bar{n}$	$\dot{W}_{12_{\text{пол}}}$, kW	$\dot{W}_{12_{\text{вн}}}$, кВт	$\dot{\Psi}_{12}^{\text{тр}}$, kW
1,15	341,17	4,94	1,161	–151,12	–200	48,88

Let us represent the equilibrium polytropic process of nitrogen compression ($1–2\bar{n}$) in coordinates (*PV*), (*TS*) (Fig. 23).

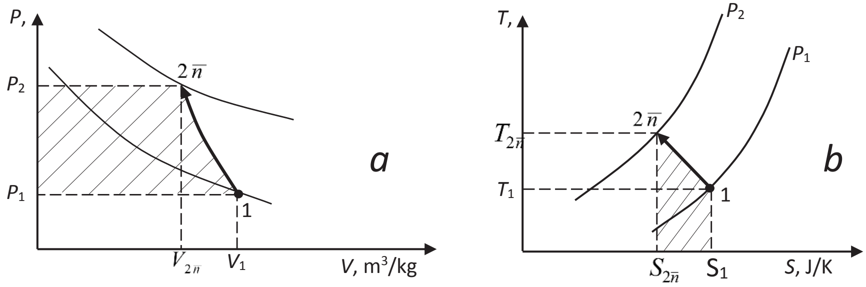


***Figure 23.** Polytropic equilibrium process of nitrogen compression with heat removal in the coordinates PV(a), TS(b).*

We determine the polytropic efficiency, the value of which characterizes the degree of energy perfection of the gas compression process $\eta_{пол}$.

$$\eta_{пол} = \frac{\dot{W}_{12_{пол}}}{\dot{W}_{12_{вн}}} = \frac{-151{,}12}{-200} = 0{,}75\,.$$

3.) Exergy analysis of the nitrogen compression process with heat removal.

The exergy analysis of the process of nitrogen compression with heat removal is carried out on the basis of the exergy balance equation (3.7). Heat flow exergy is calculated based on the equation:

$$\dot{E}_{12}^{Q} = \dot{Q}\left(1 - \frac{T_{о.с.}}{\bar{T}_{пр}}\right),$$

where $\dot{Q}$ – heat flow removed from the gas, kW,

$T_{о.с}$.= 298,15 K – ambient temperature, under standard conditions,

$\bar{T}_{пр}$ – average thermodynamic temperature of the process, K,

The exergy balance equation (3.7), taking into account the ratio, takes the following form:

$$\dot{m}(e_1 - e_2) + \dot{Q}(1 - \frac{T_{о.с.}}{\bar{T}_{пр}}) - \dot{W}_{12_{вн}} - in\dot{D}_{12} = 0\,,$$

where $\dot{m}(e_1 - e_2) = -\dot{m}\left[\int_{T_1}^{T_2} C_{pиd}\,dT - T_{о.с.}\left(\int_{T_1}^{T_2} \frac{C_{pиd}}{T}\,dT - R_M \ln\frac{P_2}{P_1}\right)\right]$ – exergy flow due to the convective movement of the gas flow, kW,

$\dot{E}^Q = \dot{Q}\left(1 - \frac{T_{\text{о.с.}}}{\overline{T}_{\text{пр}}}\right)$ – heat flow exergy, kW,

$\dot{W}_{12_{\text{вн}}} = \frac{\dot{W}^P_{12_T}}{\eta_T}$ – technical power of the compressor, kW, where η_T - the internal efficiency of the compressor stage,

$in\dot{D}_{12}$ – internal exergy losses due to irreversibility of the process, kW.

Calculation of internal losses of exergy $in\dot{D}_{12}$, due to the irreversibility of the process itself is carried out on the basis of the following relation (3.16):

$$in\dot{D}_{12} = \dot{\Psi}^{\text{тр}}_{12}\frac{T_{\text{о.с.}}}{\overline{T}_{\text{пр}}},$$

where $\dot{\Psi}^{\text{тр}}_{12}$ – the dissipation of kinetic energy due to friction, kW,

$T_{\text{о.с}}$= 298,15 K – ambient temperature under standard conditions,

$\overline{T}_{\text{пр}}$ – average thermodynamic temperature of the process, K.

The results of calculations of the exergy flux of nitrogen due to convection are presented in Table 4.2.

Table 4.2.

Calculation of nitrogen specific exergy loss due to convection

T_1, K	T_2, K	$\int_{T_1}^{T_2} C_{рид}dT$, kJ/kg	$\int_{T_1}^{T_2}\frac{C_{рид}}{T}dT$, kJ/(kg·K)	W^{tex}_{12} kJ/kg	$e_{x1} - e_{x2}$ kJ/kg
293	341,17	50	0,158	–200	147,13

The calculation of the average thermodynamic temperature of the process $\overline{T}_{\text{пр}}$ is carried out on the basis of the exergy balance equation (3.15).

$$\overline{T}_{\text{пр}} = T_{o.c.}\left[\frac{\dot{Q}_{12} + \dot{\Psi}^{\text{тр}}_{12}}{\dot{Q}_{12} + \dot{m}(e_1 - e_2) - \dot{W}_{12_{\text{вн}}}}\right] = 298{,}15\left[\frac{-150 + 48{,}88}{-150 - 147{,}13 + 200}\right] = 310{,}4\text{K}.$$

The calculation of exergy losses due to the irreversibility of the process $in\dot{D}_{12}$ is carried out according to the calculated ratio (3.16):

$$in\dot{D}_{12} = \dot{\Psi}^{\text{тр}}_{12}\frac{T_{\text{о.с.}}}{\overline{T}_{\text{пр}}} = 48{,}88\frac{298{,}15}{310{,}4} = 46{,}95\text{кВт}.$$

The value $in\dot{D}_{12} = 46{,}95\text{k}W$ is the final loss of exergy, which can only be compensated by external energy carriers.

The difference between the values of dissipation of kinetic energy and internal losses of exergy, due to the irreversibility of the process, is that part of the exergy that can still be usefully used in the future.

$$\dot{\Psi}_{\text{тр}} - in\dot{D}_{12} = 48{,}88 - 46{,}95 = 1{,}93\text{кВт}.$$

In the event that the thermal exergy of the cooling agent is not used, since its value is very small, then the exergy losses are calculated according to the equation:

$$\dot{D}_{12} = in\dot{D}_{12} + ex\dot{D}_{12},$$

where $\dot{D}_{12}$ – the total exergy losses, kW,

$in\dot{D}_{12}$ – internal exergy losses, kW,

$ex\dot{D}_{12} = -\dot{E}^{Q} = -\dot{Q}\left(1 - \frac{T_{\text{о.с.}}}{\overline{T}_{\text{пр}}}\right)$ – external exergy losses, kW.

The calculation of the exergy efficiency of the compression process with heat removal in this case is carried out according to the following relation:

$$\eta_{ex} = 1 - \frac{in\dot{D}_{12} + ex\dot{D}_{12}}{-\dot{W}_{12_{\text{вн}}}}.$$

The results of calculations of the exergy efficiency of the ammonia compression process without utilization of the removed heat are presented in Table. 4.3.

Table 4.3

Calculation of the exergy efficiency of the process without utilization of the removed heat flow

$\dot{Q}_{12}$, kW	$\dot{W}_{12_{\text{вн}}}$, кВт	$ex\dot{D}_{12}$, кВт	$in\dot{D}_{12}$, кВт	$\overline{T}_{\text{пр}}$, K	η_{ex}
–150	–200	5,92	46,95	310,4	0,74

To increase the efficiency of the process, it is advisable to provide a deeper cooling of the compressed gas. To improve the cooling system, air or oil cooling can be used.

4.2. Tasks for solitary work

Task 4.6. Reciprocating refrigerated compressor 1 sucks in nitrogen at a temperature T_1 = 306 K and pressure P_1 = 1 bar and pumps compressed gas into cylinders 3 with a volume of 50 l (Fig. 23). The gas pressure at the outlet of the compressor 1 is maintained constant and equal to P_2 = 5 bar due to the installation of a control valve 2 on the discharge lines. The compression process is non-equilibrium, it is known that the ratio of the removed heat and internal work is φ = 0,6. Gas parameters at the outlet of the compressor: P_2 = 5 bar. The compressor capacity is equal to $\dot{V}_2$ = 1,2 m^3/min in terms of compressed gas parameters. To calculate the parameters and functions of the state of nitrogen according to the Bogolyubov-Mayer virial equation of a non-ideal gas in a truncated form [1]. Critical parameters and heat capacity constants of nitrogen are presented in Annex (P-3, P-4). The internal isothermal efficiency of the compressor is taken equal to η_T = 0,7.

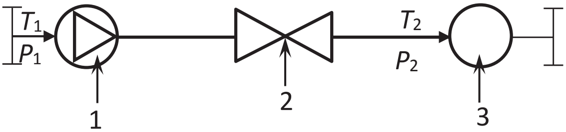


Figure 23. *Schematic diagram of the installation*
1 - piston compressor; 2 - control valve; 3 - cylinder for compressed gas

Let's determine: 1) nitrogen temperature at the compressor outlet T_2, K; 2) compressor internal power $\dot{W}_{12_{вн}}$, kW; 3) dissipation of kinetic energy on friction, $\dot{\Psi}_{12}^{тр}$, kW; 4) compressor operation time for filling thirty cylinders τ, s; 5) the rate of exergy loss due to the irreversibility of the process, $in\dot{D}_{12}$, kW; 6) exergy efficiency of the compressor unit, provided that the removed heat flow is usefully used, η_{ex}.

Answer: 1) T_2 = 386,67 K; 2) $\dot{W}_{12_{вн}} = -18{,}16$ kW; 3) $\dot{\Psi}_{12}^{тр}$ = 3,83 kW; 4) τ = 75 c; 5) $in\dot{D}_{12}$ = 3,315 kW; 6) η_{ex} = 0,82.

Task 4.7. The turbocharger is used to compress carbon dioxide to a final pressure of $P_{кон}$ = 20 bar. The state of the gas during suction is determined by the pressure $P_{нач}$ = 1 bar and temperature $T_{нач}$ = 280 K. The compression process is adiabatic, non-equilibrium. The compressor capacity is 30 m^3/h of compressed gas, the degree of pressure increase in one stage is not allowed more than ε = 4. Between the stages, the compressed gas is cooled isobarically to the initial temperature $T_{нач}$ = 280 K in heat exchangers. For cooling, water from the circulating water supply is used, the heating of the cooling water is ΔT_{H_2O} = 7 K. The value of the adiabatic efficiency of all stages of the compressor unit is assumed to be the same and equal to η_S = 0,83. In calculations, it is permissible to use the model of an ideal gas with a constant heat capacity.

Let's determine: 1) the number of stages of the compressor unit, n; 2) internal power of each stage $\dot{W}_{вн}$, kW; 3) heat flow removed in intermediate coolers $\dot{Q}_{H_2O}$, kW; 4) exergy efficiency of the compressor unit $\eta_{ex_{к.у.}}$

Answer: 1) n = 3; 2) $\dot{W}_{вн}^{I} = \dot{W}_{вн}^{II} = \dot{W}_{вн}^{III} = -17{,}44$ kW; 3) $\dot{Q}_{H_2O} = -17{,}44$ kW; 4) $\eta_{ex_{к.у.}}$ = 0,78.

Task 4.8. Oxygen is compressed in the adiabatic stage of an uncooled compressor. Oxygen parameters at the compressor inlet: $T_1 = 300$ K, $P_1 = 1$ bar, compressor outlet pressure: $P_2 = 4$ bar. After compression, the gas is sent to the refrigerator, where it is cooled isobarically to the initial temperature $T_1 = 300$ K. The gas flow rate is $\dot{m} = 1$ kg/s, the compression process is nonequilibrium. Calculation of the parameters and functions of the state of oxygen is carried out according to the virial equation of a nonideal Bogolyubov-Mayer gas in a truncated form [1]. The critical parameters and heat capacity constants of oxygen are given in [1, 4] and presented in Annex (P-1, P-2). The adiabatic internal efficiency of the compressor is $\eta_S = 0{,}8$.

To determine: 1) oxygen temperature at the outlet of the compressor T_2, K; 2) compressor internal power $\dot{W}_{12_{\text{вн}}}$, kW; 3) polytropic compressor efficiency $\eta_{\text{пол}}$; 4) heat flow removed from oxygen in the heat exchanger $\dot{Q}_{H_2O}$; 5) exergy efficiency of the compressor unit, provided that the removed heat flow $\dot{Q}_{H_2O}$ is not usefully used, $\eta_{ex_{\text{к.у.}}}$.

Answer: 1) $T_2 = 481{,}55$ K; 2) $\dot{W}_{12_{\text{вн}}} = -163{,}18$, kW; 3) $\eta_{\text{пол}} = 0{,}847$;4) $\dot{Q}_{H_2O} = -162{,}27$ kW; 5) $\eta_{ex_{\text{к.у.}}} = 0{,}87$.

Task 4.9. Oxygen is compressed in an adiabatic compressor, gas parameters at the compressor inlet: $T_1 = 300$ K, $P_1 = 1$ bar, gas pressure at the compressor outlet: $P_2 = 4$ bar. The compression process is non-equilibrium, the gas flow rate is $\dot{m} = 1$ kg/s. It is known that the internal work of the compressor is 28 kJ/kg greater in absolute value than in a reversible adiabatic process proceeding up to the same final pressure. The functions and parameters of the state of oxygen are calculated according to the Bogolyubov-Mayer virial equation of a non-ideal gas in a truncated form [1]. The critical parameters and heat capacity constants of oxygen are given in [1, 4] and presented in Annex (P-1, P-2).

To determine: 1) the final temperature of oxygen in the irreversible process of compression T_2, K; 2) compressor internal power $\dot{W}_{12_{\text{вн}}}$, kW; 3) loss of exergy due to the irreversibility of the process $in\dot{D}_{12}$, kW, kW; 4) dissipation of kinetic energy on friction $\dot{\Psi}_{12}^{\text{тр}}$, kW; 5) exergy efficiency of the compressor η_{ex}.

Answer: 1) $T_2 = 476{,}15$ K; 2) $\dot{W}_{12_{\text{вн}}} = -161{,}03$ kW; 3) $in\dot{D}_{12} = 16{,}0063$ kW; 4) $\dot{\Psi}_{12}^{\text{тр}} = 23{,}73$ кВт; 5) $\eta_{ex} = 0{,}901$.

Task 4.10. To determine the savings in internal power that can be obtained by switching from single-stage to two-stage compression with intermediate isobaric cooling in a heat exchanger to the initial temperature of a mixture of oxygen and nitrogen in equal mole fractions. The parameters of the mixture at the inlet to the compressor: $T_1 = 306$ K, $P_1 = 1$ bar, outlet pressure: $P_2 = 7$ bar. The compression process is non-equilibrium, adiabatic, mixture flow rate is $\dot{m} = 1$ kg/s. The degree of pressure increase in both stages is considered the same, the internal adiabatic efficiency of each stage is taken equal to $\eta_S = 0{,}8$. A mixture of gases can be considered ideal with a constant heat capacity.

Answer: $10{,}6 \cdot 10^{-2}$.

Task 4.11. To determine the energy dissipation during the movement of nitrogen into the reactor along the horizontal section of the gas pipeline. Inlet diameter $d_1 = 0{,}04$ m, outlet diameter $d_2 = 0{,}03$ m. Nitrogen parameters at the gas pipeline inlet: $T_1 = 300$ K, $P_1 = 1{,}5$ bar; parameters at the outlet of the gas pipeline; $T_2 = 302$ K, $P_2 = 1{,}3$ bar. Gas consumption $\dot{m} = 0{,}1$ kg/s. The process is unbalanced. Nitrogen can be considered an ideal gas with a constant heat capacity due to a slight deviation of the parameters from the standard ones.

Answer: $\dot{\Psi}_{12}^{mp} = 0{,}36\ \text{kW}$.

Task 4.12. The compressor sucks in air at a pressure of 1 atm and a temperature of 20°C and compresses it isothermally up to 8 atm. (compressor internal efficiency = 0,85). To determine the capacity of the compressor m^3/h, if the theoretical engine power to drive the compressor is 40 kW. Also determine the hourly consumption of cooling water if its temperature rises by 10°C when the compressor cylinder is cooled. The heat capacity of water is taken equal to 4.19 kJ/kg K.

Conclusion

The monograph is a consistent continuation of previous educational publications and manuals, scientific articles written by the author and devoted to the study of the processes of obtaining, converting and using energy in gases, vapors, condensed media and their mixtures. Pedagogical experience in preparing for the publication of this work is aimed at attracting the attention and professional interest of young scientists to solving urgent problems in modern energy and the search for non-traditional energy carriers and alternative energy sources.

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Annex P-1

Dynamic viscosity coefficients of gases (μ, Pa·s) depending on temperature (t, °C) at pressure P0=1 atm and values of Sutherland constant (C) in the equation $\boldsymbol{\mu} = \boldsymbol{\mu_0} \frac{\boldsymbol{T_0} + \boldsymbol{C}}{\boldsymbol{T} + \boldsymbol{C}} \left(\frac{\boldsymbol{T}}{\boldsymbol{T_0}} \right)^{3/2}$, where: T_0 = 273,15 K; μ_0– dynamic coefficient viscosity at T_0 = 273,15 K and P_0=1 atm.

Name	**Formula**	**Dynamic viscosity coefficient, μ·10⁶, Pa·s**											***C*, K**	**Temperature range for C, °C**
		Temperature, *t*, °C												
		0	**20**	**50**	**100**	**150**	**200**	**250**	**300**	**400**	**600**	**800**		
Nitrogen	N_2	16,5	17,5	18,8	20,8	22,8	24,6	26,3	28	31,1	36,6	41,3	104	25-280
Ammonia	NH_3	9,3	10	11,1	12,8	14,6	16,5	18,1	-	-	-	-	503	20-300
Hydrogen	H_2	8,5	8,8	9,4	10,3	11,3	12,1	13	13,8	15,4	18,3	21	73	20-100
													86	100-200
													105	200-250
													234	713-892
Oxygen	O_2	19,2	20,2	21,8	24,4	26,7	29	31	33	36,9	43,5	49,3	125	15-830
Sulphur dioxide	SO_2	11,6	12,5	14	16,3	18,6	20,7	22,7	24,6	28,2	34,6	40,4	306	300-825
Carbon dioxide	CO_2	13,7	14,6	16	18,3	20,4	22,5	24,5	26,5	29,9	36	41,4	254	25-280
													213	300-824
Ethane	C_2H_6	8,6	9,1	10	11,4	12,8	14,1	15,3	-	-	-	-	252	20-250
Propane	C_3H_8	7,5	8	9	10,3	11,3	12,5	13,6	14,7	-	-	-	278	20-250
Methane	CH_4	10,4	10,9	11,8	13,3	14,7	16	17,3	18,5	-	-	-	162	20-500
Carbon tetrachloride	CCl_4	9,0	-	10,8	12	13,7	15,2	17	-	-	-	-	335	50-250
Chloroform	$CHCl_3$	9,5	10,0	11,0	12,5	14,2	15,8	17,5	19,1	-	-	-	462	14-345

Annex P-2

Dynamic coefficients of viscosity of liquid substances and aqueous solutions (μ, mPa·s) depending on temperature (*t*, °C)

Name	**Dynamic viscosity coefficient μ, mPa·s (sP)**									
	Temperature t, °C									
	0	**10**	**20**	**30**	**40**	**50**	**60**	**80**	**100**	**120**
Ethanol, 100%	1,78	1,46	1,19	1,00	0,825	0,701	0,591	0,435	0,326	0,248
Ethanol, 40%	7,14	4,39	2,91	2,02	1,48	1,13	0,89	0,60	0,44	0,34
Liquid ammonia	0,244	0,225	0,226	0,217	0,208	0,199	0,190	-	-	-
Ammonia water, 25%	-	1,72	1,30	1,05	0,855	0,71	0,60	0,42	0,32	0,23
Water	1,79	1,31	1,00	0,801	0,656	0,549	0,469	0,357	0,284	0,232
Glycerin, 50%	12	8,5	6,05	4,25	3,5	2,6	2,0	1,2	0,73	0,45
Glycerin, 100%	12100	3950	1480	600	330	180	102	35	13	5,2
Sodium chloride, 20%	2,67	1,99	1,56	1,24	1,03	0,87	0,74	0,57	0,46	0,38
Carbon tetrachloride	1,35	1,13	0,97	0,84	0,74	0,65	0,59	0,472	0,387	0,323

Annex P-3

Critical parameters of gases

Name	**Formula**	$M, \frac{\text{kg}}{\text{kmol}}$	T_c, **K**	P_c, **atm**	**ω** **Pitzer acentric factor**
Ammonia	NH_3	17,031	405,6	111,3	0,25
Nitrogen	N_2	28,013	126,2	33,5	0,04
Nitrogen dioxide	NO_2	46,006	431,4	100	0,86
Sulphur dioxide	SO_2	64,063	430,8	77,8	0,251
Oxygen	O_2	31,999	154,6	49,8	0,021
Freon-22	CF_2ClH	86,469	369,2	49,1	0,215
Freon-12	CF_2Cl_2	120,914	385	40,7	0,176
Freon-11	CCl_3F	137,368	471,2	43,5	0,188
Carbon dioxide	CO_2	44,01	304,2	72,8	0,225
Nitrogen monoxide	NO	30,006	180	64	0,607
Chlorine	CL_2	70,906	417	76	0,073
Nitrous oxide	N_2O	44,013	309,6	71,5	0,16
Hydrogen sulfide	H_2S	34,8	373,2	88,2	0,1
Sulfur trioxide	SO_3	80,058	491	81	0,41
Fluorine	F_2	37,997	143,3	51,5	0,048

Annex P-4

Critical parameters of hydrocarbons

Name	**Formula**	$M, \frac{\text{kg}}{\text{kmol}}$	T_c, **K**	P_c, **atm**	**ω** **Pitzer acentric factor**
Methane	CH_4	16,043	190,6	45,4	0,008
Acetylene	C_2H_2	26,038	308,3	60,6	0,184
Ethylene	C_2H_4	28,054	282,4	49,7	0,065
Ethane	C_2H_6	30,07	305,4	43,2	0,098
Propylene	C_3H_6	42,081	365	45,6	0,148
Propane	C_3H_8	44,097	369,8	41,9	0,152

1,3-Butadiene	C_4H_6	54,092	425	42,7	0,195
1-Butene	C_4H_8	56,108	419,6	39,7	0,187
n-butane	C_4H_{10}	58,124	425,2	37,5	0,193
Isobutane	C_4H_{10}	58,124	408,1	36	0,176
2-Butin	C_4H_6	54,092	488,6	50,2	0,124
1-Butin	C_4H_6	54,092	463,7	46,5	0,27
Cyclopentene	C_5H_8	68,119	506	–	–
Cyclobutane	C_4H_8	56,108	459,9	49,2	0,209
Isobutene	C_4H_8	56,108	417,9	39,5	0,190
1-Chlorobutane	C_4H_9CL	92,569	542	36,4	0,218

Annex P-5

Thermophysical characteristics of hydrocarbons

Name	Δh_V (T_{HB}), $\frac{\text{kkal}}{\text{kmol}}$	$\widetilde{C_P}$, $\frac{\text{kJ}}{\text{kmol} \cdot \text{K}}$				$\ln P_V(T) = A - \frac{B}{T+C}$, mm Hg		
		d_0	$d_1 \cdot 10^2$	$d_2 \cdot 10^5$	$d_3 \cdot 10^9$	**A**	**B**	**C**
Methane	1955	19,268	5,217	1,198	–11,326	15,2243	897,84	–7,16
Acetylene	4050	26,841	7,584	–5,011	14,133	16,3481	1637,14	–19,77
Ethylene	3237	3.809	15,671	–8,355	17,565	15,5368	1347,01	–18,15
Ethane	3515	5,414	17,824	–6,943	8,719	15,6637	1511,42	–17,16
Propylene	4400	3,712	23,472	–11,611	22,065	15,7027	1807,53	–26,15
Propane	4487	–4,228	30,649	–15,876	32,171	15,7260	1872,46	–25,16
1,3 - Butadiene	5370	–1,689	34,211	–23,418	63,395	15,7727	2142,66	–34,3
1-Butene	5238	–2,996	35,347	–19,919	44,665	15,7564	2132,42	–33,15
n-butane	5252	9,495	33,155	–11,091	–28,241	15,6782	2154,90	–34,42

Heat capacity constants of gases:

$$\widetilde{C_P} = \widetilde{d_0} + \widetilde{d_1}T + \widetilde{d_2}T^2 + \widetilde{d_3}T^3, \frac{\text{kJ}}{\text{kmol·K}};$$

Antoine's equation constants: A, B, C;

Heat of vaporization at normal barometric temperature T_{HB}, K: $\Delta h_V(T_{HB})$, $\frac{\text{kkal}}{\text{kmol}}$.

Annex P-6

Thermophysical characteristics of gases

Name	Δh_{V} (T_{HB}), $\frac{\mathrm{kkal}}{\mathrm{kmol}}$	$\widetilde{C_P}$, $\frac{\mathrm{kJ}}{\mathrm{kmol}\cdot\mathrm{K}}$				$\ln P_{\mathrm{V}}(T) = A - \frac{B}{T+C}$, mm Hg		
		d_0	**$d_1 \cdot 10^2$**	**$d_2 \cdot 10^5$**	**$d_3 \cdot 10^9$**	**A**	**B**	**C**
Ammonia	5580	27,3366	2,385	1,708	–11,858	16,9481	2132,5	–32,98
Nitrogen	1333	3,117	–1,358	2,682	–11,690	14,9542	588,72	–6,6
Nitrogen dioxide	4555	24,252	4,839	–2,082	0,293	20,5324	4141, 29	3,65
Sulphur dioxide	5955	23, 870	6,704	–4,965	13,291	16, 768	2302, 35	–35,97
Oxygen	1630	28,127	–0,0036	1,747	–10,659	15,4075	734,55	–6,45
Freon-22	4826	17, 313	16,194	–9,612	30, 608	15,5602	1704,8	–41,3
Freon-11	5920	41, 016	16, 312	–14,091	41, 494	15,8516	2401,61	–36,3
Carbon monoxide	1444	30, 893	–1,286	2,791	–12,725	14,3686	530,22	–13,15
Carbon dioxide	4100	19,81	7,349	–5,606	17, 166	22,5898	3103, 39	–0,16
Nitrogen monoxide	3300	29, 368	–0,094	0, 9754	–4,19	20,1314	1572, 52	–4,88

Heat capacity constants of gases:

$$\widetilde{C_P} = \widetilde{d_0} + \widetilde{d_1}T + \widetilde{d_2}T^2 + \widetilde{d_3}T^3, \frac{\mathrm{kJ}}{\mathrm{kmol}\cdot\mathrm{K}};$$

Antoine’s equation constants: A, B, C;

Heat of vaporization at normal barometric temperature T_{HB}, K: $\Delta h_{\mathrm{V}}(T_{\mathrm{HB}})$, $\frac{\mathrm{kkal}}{\mathrm{kmol}}$.

Khabibova Natalya Zamilovna

DISSIPATIVE FUNCTION IN ENGINEERING CALCULATIONS. FUNDAMENTAL PRINCIPLES AND PRACTICAL APPLICATIONS

Monograph

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Alma Mater ensured the formation of Khabibova N.Z. as a teacher and a scientist, and also instated her professional and personal maturity. In turn, the university received a dedicated employee - the scientific interests of Khabibova N.Z. – technical thermodynamics and energy-saving technologies, have been inextricably linked for 30 years with Mendeleev University of Chemical Technology - one of the leading classical universities and research centers in Russia.

This book presents the results of a study of the digital economy as a new paradigm of economic development, a system of economic relations implemented through the use of digital information computer technologies.

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