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FOREIGN TECHNOLOGY DIVISION

AIR FORCE SYSTEMS COMMAND

VAPOR-JET VACUUM PUMPS
(SELECTED CHAPTERS)

FOREIGN TECHNOLOGY DIVISION

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EDITED MACHINE TRANSLATION

VAPOR-JET VACUUM PUMPS (SELECTED CHAPTERS)

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U. S. BOARD ON GEOGRAPHIC NAMES TRANSLITERATION SYSTEM

Block	Italic	Transliteration	Block	Italic	Transliteration
А а	*А а*	A, a	Р р	*Р р*	R, r
Б б	*Б б*	B, b	С с	*С с*	S, s
В в	*В в*	V, v	Т т	*Т т*	T, t
Г г	*Г г*	G, g	У у	*У у*	U, u
Д д	*Д д*	D, d	Ф ф	*Ф ф*	F, f
Е е	*Е е*	Ye, ye; E, e*	Х х	*Х х*	Kh, kh
Ж ж	*Ж ж*	Zh, zh	Ц ц	*Ц ц*	Ts, ts
З з	*З з*	Z, z	Ч ч	*Ч ч*	Ch, ch
И и	*И и*	I, i	Ш ш	*Ш ш*	Sh, sh
Й й	*Й й*	Y, y	Щ щ	*Щ щ*	Shch, shch
К к	*К к*	K, k	Ъ ъ	*Ъ ъ*	"
Л л	*Л л*	L, l	Ы ы	*Ы ы*	Y, y
М м	*М м*	M, m	Ь ь	*Ь ь*	'
Н н	*Н н*	N, n	Э э	*Э э*	E, e
О о	*О о*	O, o	Ю ю	*Ю ю*	Yu, yu
П п	*П п*	P, p	Я я	*Я я*	Ya, ya

* ye initially, after vowels, and after ъ, ь; e elsewhere. When written as ё in Russian, transliterate as yё or ё. The use of diacritical marks is preferred, but such marks may be omitted when expediency dictates.

FOLLOWING ARE THE CORRESPONDING RUSSIAN AND ENGLISH

DESIGNATIONS OF THE TRIGONOMETRIC FUNCTIONS

Russian	English
sin	sin
cos	cos
tg	tan
ctg	cot
sec	sec
cosec	csc
sh	sinh
ch	cosh
th	tanh
cth	coth
sch	sech
csch	csch
arc sin	$\sin^{-1}$
arc cos	$\cos^{-1}$
arc tg	$\tan^{-1}$
arc ctg	$\cot^{-1}$
arc sec	$\sec^{-1}$
arc cosec	$\csc^{-1}$
arc sh	$\sinh^{-1}$
arc ch	$\cosh^{-1}$
arc th	$\tanh^{-1}$
arc cth	$\coth^{-1}$
arc sch	sech^{-1}
arc csch	csch^{-1}
rot	curl
lg	log

CHAPTER IV

JET VACUUM PUMPS

4-1. Working Principle

Jet vacuum pumps constitute a variety of jet apparatuses and serve to pump out large quantities of gas from various installations operating at pressures from 760 to 10^{-2} mm Hg.

The working principle of a jet pump consists of the following (Fig. 4-1). The vapor at an increased pressure passes through the expanding nozzle 1 at supersonic speed in the form of a turbulent or laminar stream. Through the turbulent mixing (by the turbulent stream) of the vortex masses of the vapor stream with the particles of the surrounding stream of gas capture and entrainment of the gas occurs in constricting heads 2, called the mixing chamber, serving to ensure, possibly, a more complete mixing of the vapor with the gas which is characterized by an even distribution of all the parameters of the mixture (pressure, density, speed, temperature) according to section of the flow. If stream is laminar, then as was already indicated earlier, the capture mechanism of the gas by the stream will be determined by the viscous friction between the boundary layers of the stream and the adjacent layers of the pumped gas. In other respects the operating mechanism of the jet pump will be the same as in the case of a turbulent stream.

Due to exchange of energies during mixing, the pressure of the pumped gas will increase, becoming equal to the static pressure of

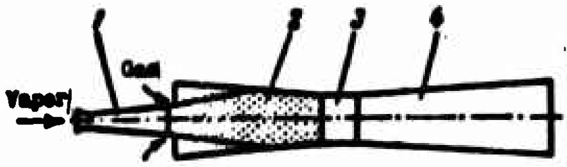


Fig. 4-1. Schematic diagram of a jet pump. 1 – nozzle; 2 – mixing chamber; 3 – throat; 4 – diffuser.

the mixture, but the flow rate will decrease (Fig. 4-2). Deceleration in this case is caused by the loss of energy during mixing. If one were to assume that the mixing of the flows occurs at a constant pressure, then the momentum of the mixture should be equal to sum of momenta of the flows:

$$\frac{G_c}{g} w_c = \frac{G_n}{g} w_{n2} + \frac{G_r}{g} w_{r2}, \qquad (4\text{-}1)$$

where G_c – quantity of the mixture, equal to $G_n + G_r$; G_n and G_r – quantity of vapor and pumped gas, respectively; w_{n2} and w_{r2} – velocity of the vapor and gas upon entry in the mixing chamber; w_c – velocity of mixture; hence

$$w_c = \frac{G_n w_{n2} + G_r w_{r2}}{G_n + G_r}. \qquad (4\text{-}2)$$

The kinetic energy of the mixture

$$E_c = \frac{G_n + G_r}{2g} w_c^2 = \frac{1}{2g} \frac{(G_n w_{n2} + G_r w_{r2})^2}{G_n + G_r} \qquad (4\text{-}3)$$

less the sums of kinetic energies of the flows

$$E_n + E_r = \frac{1}{2g} (G_n w_{n2}^2 + G_r w_{r2}^2)$$

by a value of

$$\Delta E = \frac{G_n G_r}{G_n + G_r} \frac{(w_{n2} - w_{r2})}{2g}. \qquad (4\text{-}4)$$

The reduction of the flow rate is also caused by friction loss and the appearance of shock waves. Due to the interaction of supersonic flow with walls of the mixing chamber weak oblique shock waves appear, which lead to a certain deceleration of flow. However, if the static pressure at the end of the mixing chamber is held

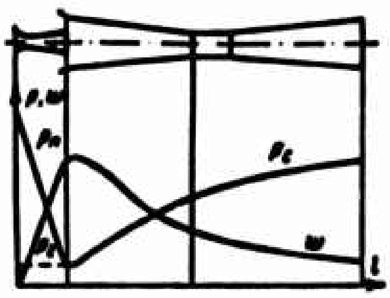


Fig. 4-2. Change in the static pressure and flow rate in a jet pump ($p_п$, $p_г$, $p_с$ – pressure of the vapor, gas, mixture).

sufficiently low, then the deceleration with respect to the shown causes can be such that the flow rate after mixing will remain supersonic and will smoothly decrease in the constricting head to a critical value. If, however, one were to increase the pressure behind the flows at the end of the mixing chamber, then under certain conditions, corresponding to those usually calculated, a normal shock wave in the mixing chamber will appear, as a result of which the flow rate transcend to the subsonic in jumps. The static pressure and flux density in this case will increase. However, in spite of the fact that the flow rate becomes subsonic, the total pressure of the mixture at the end of the mixing chamber, proportional to the energy of the flow and equal to the sum of static and dynamic pressures

$$p = p_{стат} + \frac{\rho w^2}{2}, \qquad (4\text{-}5)$$

will still be determined to a considerable degree by the value of the dynamic pressure.

At the same time, to obtain a high value of the maximum outlet pressure is important, so that the static pressure of the flow at the outlet of the jet pump will be possibly larger. This can be attained at the same value of total pressure only at the expense of a shift of kinetic energy of flow at the potential, i.e., by means of increasing the static component of pressure at the expense of decreasing the dynamic component. Expanding heads 4 (Fig. 4-1), called a diffuser serve this purpose. Its action is based on the property of subsonic flow being delayed in the expanding head with increase in static pressure.

Along with a conical mixing chamber a cylindrical mixing chamber is sometimes used, mainly in jet pumps operating at small coefficients of jet pumping and at small compression ratios.

The pressure, produced by the jet pump, essentially depends on the value of its maximum outlet pressure. Usually, a single stage of a jet vacuum compresses gas by 5-10 times. At compression ratios, larger than 10, jet pumps, as a rule, are not designed for economical considerations, connected with the increased vapor consumption at high compressions. Consequently, a single stage of a jet pump, operating with an exhaust in the atmosphere, can create a pressure of 100-150 mm Hg.

In order to produce lower pressures it is necessary to set up a series of stages. In this case a condenser is usually installed behind the jet pump stage, in which the vapor is condensed, and the gas is pumped out by the following stage. A diagram of a typical vapor-water jet pump with an intermediate condenser is shown in Fig. 4-3. In order to produce even a small injected pressure with single stage of jet pump it is necessary to maintain low pressure behind the stage, which usually is attained by installing an auxiliary vacuum pump.

Water, mercury and vacuum oils are used as working fluids in jet vacuum pumps. The kind of working fluid determines to a considerable degree the operating conditions, characteristics, and design of the pump.

In vacuum technology vapor jet pumps are used as independent vacuum pumps, as well as outlet stages of multistage booster and high-vacuum pumps.

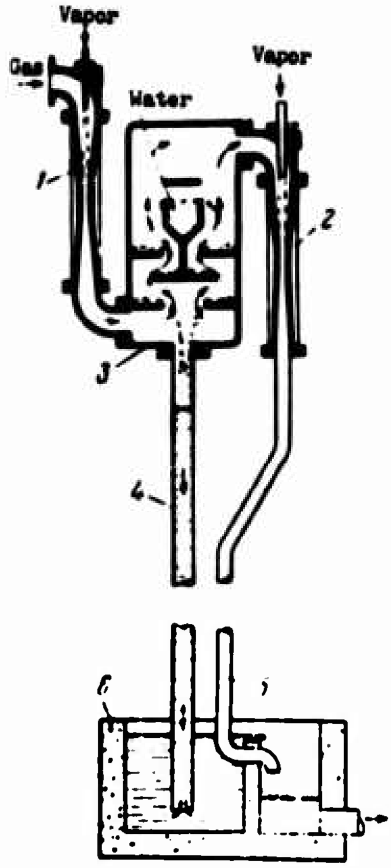


Fig. 4-3. Diagram of a two-stage vapor-water jet pump. 1 – first stage; 2 – second stage; 3 – mixing condenser; 4 – overflow pipe; 5 – exhaust pipe; 6 – barometric condensate cistern.

4-2. Characteristics

The basic characteristic of vapor jet pumps is dependence of productivity on the inlet pressure. Usually, for vapor jet pumps this characteristic is depicted as inverted coordinates, i.e., as a dependence of the inlet pressure on the productivity or external load (Fig. 4-4). The characteristic has two sections, inclined at various angles to the axis of the abscissas. The slightly inclined one – the working section, and the steeply inclined one – the overload section. Of the characteristics in the working section considerable changes in the load changes the inlet pressure slightly. In the transfer section small changes in load result in a sharp increase of the inlet pressure.

A pump is always calculated for performance characteristics in the working section. Transition to the overloaded section is accompanied by the unstable work of the pump and by the increased migration of vapors of the working fluid in the pumped out volume.

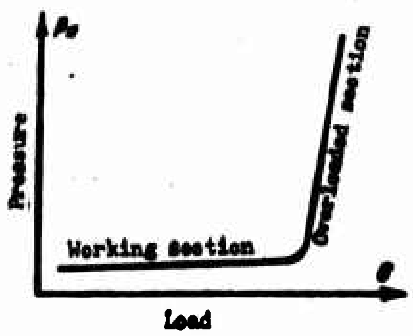


Fig. 4-4. Characteristics of vapor jet pump.

Therefore, the characteristic of the prolonged work of the pump in the overloaded section is not recommended.

The inflection point of characteristic, corresponding to the peak output of the pump in the working section, is the design operating point, for which one usually makes the calculations and design for a vapor jet pump. The output of vapor jet pumps is usually measured in kilograms per hour. Volume output or operating speed of the pump, depending upon the inlet pressure, has the form of curve with a maximum (Fig. 4-5), where the maximum corresponds to the operating point.

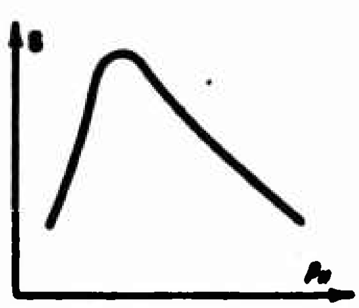


Fig. 4-5. Dependence of the operating speed of a vapor jet pump on the inlet pressure.

The maximum vacuum for vapor jet pumps is usually a secondary characteristic, since the basic assignment of pumps is for the pumping out of possibly large quantities of gas, i.e., operation in the region of peak output.

For vacuum vapor jet pumps along with the ordinary parameters (output, operating speed, maximum vacuum and the maximum outlet pressure), they are characterized by one more important parameter — the coefficient of ejection. Coefficient of ejection is equal to ratio of the weight of pumped gas (kg/h) to the weight of the working vapor, passing through the nozzle:

$$\nu = \frac{G_r}{G_n}. \qquad (4\text{-}6)$$

For vacuum vapor jet pumps the coefficient of ejection is usually less than unity. The graphic dependence of the coefficient of ejection on the inlet pressure has the form of a curve, similar to the characteristics of the pump in Fig. 4-4.

4-3. Theory and Design of Vapor Jet Pumps

a) *State of the theory and design.* In spite of the exceptional simplicity of the makeup of jet pumps the physical processes going on inside them are very complicated and varied. A study of these processes became the object of numerous theoretical and experimental investigations, starting from the second half of the past century, and up to the present [37-159]. The main aim of these investigations was to devise methods of calculating and designing jet pumps. At present much is published on the theoretical as well as empirical methods of design. As for the theoretical works devoted to the study of jet pumps with a large compression ratio of the ejected gas and large expansion ratio of the ejecting vapor, including the vacuum jet pumps, the most significant are the works, carried out at the Central Institute of Aerohydrodynamics [TsAGI] under the leadership of S. A. Khristianovich [38, 156, 39], at the Moscow Power Engineering Institute [MEI] under the leadership of M. Ye. Deych [40-44] and at the All-Union Institute of Heat Engineering [VTI] under the leadership of Ye. Ya. Sokolov, N. M. Zinger and L. D. Berman [45-56]. The empirical method of design based on the results of the generalization of an experiment in designing and investigating jet pumps, was most fully developed by Wiegand [37, 111, 112].

An analysis shows that the majority of theoretical methods is based mainly on some original theoretical premises and they differ from one another only in the absoluteness of the treatment of the problem. The basis of these methods is the consideration of the ejecting and ejected flows as two separate flows upon entry in mixing chamber. The determination of the parameters of the jet pump is made in this case with the help of fundamental equations of gas dynamics, recorded for the initial and final sections of the mixing chamber. Such a principle of calculation of the mixing chamber permits one to obtain the basic calculated equations without consideration of the processes going on. At the same time, while setting up the necessary calculated relationships of the jet pump along with the theoretical dependences a whole series of empirical coefficients is used, in particular, the loss factors, the relationships for axial dimensions, angles and others. For this reason such methods turn out to be applicable for a calculation of only those conditions and designs of jet pumps, for which the necessary empirical values are well-known. This circumstance imposes essentially the same limitations on the theoretical methods of calculation, which also exist for purely empirical methods of calculation. However, compared to the empirical methods of calculation, the theoretical methods permit one to not only determine the basic parameters and dimensions of the jet pump, but also to construct its characteristics as well, and to follow the change in the different parameters depending upon the operating conditions of the jet pump.

Taking into account these remarks, it is most expedient to use jointly both the theoretical and the empirical methods in the engineering calculations of the jet pumps. When determining the basic dimensions of a jet pump for assigned parameters at the operating point, it is more convenient to use the simple empirical method. The construction of the characteristics of the pump and the analysis of change of its parameters depending upon various factors, can be performed with the help of the necessary theoretical relationships.

b) *The calculated diagram of a jet pump.* Let us assume that nozzle outlet section 1-1 is made coincident with inlet section of the mixing chamber, 2-2 (Fig. 4-6a). Mixing chamber consists of a conical section between sections 2-2 and 3'-3' and a cylindrical stabilizing section between sections 3'-3' and 3-3. Let us

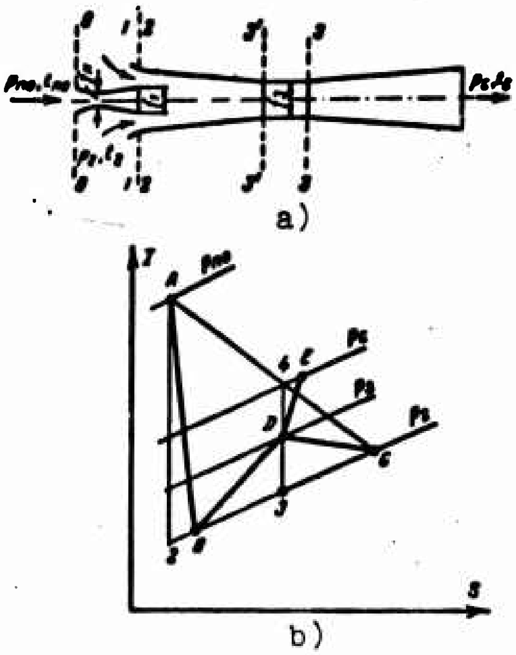


Fig. 4-6. Calculated diagram of a jet pump.

introduce the designations:

parameters of the working vapor $p_{\text{п}}$, $t_{\text{п}}$, etc.;

parameters of the ejected gas $p_{\text{г}}$, $t_{\text{г}}$, etc.;

parameters of mixture behind the diffuser $p_{\text{с}}$, $t_{\text{с}}$, etc.;

critical parameters $p_{\text{п.к}}$, $p_{\text{с.к}}$, etc.;

parameters of the vapor upon entry in nozzle $p_{\text{п}0}$, $t_{\text{п}0}$, $v_{\text{п}0}$, etc.;

parameters of the vapor in the critical section of the nozzle $f_{\text{к}}$: $p_{\text{п.к}}$, $w_{\text{п.к}}$, $t_{\text{п.к}}$, $v_{\text{п.к}}$, etc.;

parameters of the vapor in the nozzle outlet section f_1: $p_{\text{п}1}$, $t_{\text{п}1}$, $v_{\text{п}1}$, $w_{\text{п}1}$, etc.;

parameters of the gas upon entry in jet pump $p_{\text{г}}$, $t_{\text{г}}$, $w_{\text{г}}$, etc.;

parameters of the gas in the inlet section of the mixing chamber $f_{\text{г}2}$ (annular section between the nozzle and walls of the chamber) $p_{\text{г}2}$, $t_{\text{г}2}$, $w_{\text{г}2}$, etc.;

parameters of the mixture in Section 3-3 (f_3): p_3; t_3; w_3, etc.;

adiabatic indices: vapor – $\kappa_п$; gas – $\kappa_г$; mixture – $\kappa_с$.

Equality is assumed for the static pressures of the vapor and gas in section 2-2.

The processes of flow in the nozzle and diffuser, and also the process of mixing proceed with losses.

In the diagram the *I-S* process in the operation of jet pump can be approximately depicted as shown in Fig. 4-6b. In an ideal jet pump without losses the working vapor from point *A*, characterizing its initial state, is expanded in the nozzle adiabatically to pressure p_2 at point 2. The state of the ejected vapor upon entry in mixing chamber is characterized by point *C*. During isobaric mixing the state of mixture in front of the diffuser is characterized by point 3. In the diffuser the mixture is compressed adiabatically from pressure p_2 (point 3) up to pressure $p_с$ (point 4). In an actual jet pump the expansion of vapor in the nozzle occurs with losses; therefore, the final state of the vapor after expansion corresponds to a certain point *B*. The process of mixing is also accompanied by losses and does not flow isobarically, so that the state of mixture at the end of mixing chamber will be characterized by point *D*. Compression of mixture in the diffuser is also accompanied by losses, and the state of mixture at the end of compression will be characterized by point *E*.

c) *Determination of the basic parameters and geometric dimensions of the jet pump.* Usually, when designing a jet pump the initial parameters for the calculation are:

1. Output at the assigned inlet pressure (operating point).

2. Parameters of the ejected gas at the operating point (composition, pressure, temperature).

3. Parameters of the working vapor (pressure, temperature).

4. Temperature of the cooling agent in the condenser.

5. Maximum outlet pressure.

Furthermore, the known basic thermodynamic characteristics of the working and ejected media are assumed.

It is necessary to determine the dimensions of the jet pump, thereby ensuring the obtaining of the assigned characteristics.

The greatest difficulties in calculation appear during the determination of the coefficient of ejection, i.e., the quantity of working vapor, necessary to ensure the prescribed output at the assigned compression ratio. The value of the coefficient of ejection depends on many factors, such as structure of the stream, operating conditions of the jet pump, geometric dimensions of the nozzle and mixing chamber, and others. At the same time, during the previous calculation of a jet pump many of these factors turn out to be unknowns; for example, the geometric dimensions of the nozzle and mixing chamber. In connection with this, when concluding the calculated equation for the coefficient of ejection, one usually makes a whole series of simplifying assumptions, thereby allowing one to obtain an acceptable relationship for the preliminary designs.

Sokolov and Zinger [45] in deriving an approximate equation for the coefficient of ejection, assume that the working vapor and ejected gas enter mixing chamber as two separate flows, and that the process of mixing in the conical part of the mixing chamber flows isobarically and the velocity of the ejected flow upon entry in mixing chamber is equal to zero ($w_{r2} = 0$). Taking into account these assumptions they can regard the entire mixing chamber as being cylindrical with a diameter, d_3, disregarding its expansion in the inlet section. For the inlet and outlet sections of the cylindrical mixing chamber the equation of momentum can be recorded

$$\varphi_2\left[\frac{G_n}{g}w_{n2}+\frac{G_r}{g}w_{r2}\right]-\frac{G_n+G_r}{g}w_3=(p_3-p_2)f_3, \qquad (4\text{-}7)$$

where p_2 and p_3 - static pressures in the inlet and outlet sections of the cylindrical mixing chamber; ϕ_2 - coefficient of the velocity of mixing chamber, considering the loss of momentum due to friction.

From the equation of the conservation of mass

$$G_n+G_r=G_3 \qquad (4\text{-}8)$$

taking into account that $G=\frac{fw}{v}$ and $\frac{G_r}{G_n}=\nu$, we will obtain:

$$f_3=\frac{G_n(1+\nu)v_3}{w_3}, \qquad (4\text{-}9)$$

where v_3 - specific volume of the mixed flow in the outlet section of the mixing chamber.

Solving (4-7) and (4-9) jointly taking into account $w_{r2} = 0$, we will find:

$$\nu=\frac{\varphi_2 w_{n2}-w_3\left[1+\frac{(p_3-p_2)v_3 g}{w_3^2}\right]}{w_3\left[1+\frac{(p_3-p_2)v_3 g}{w_3^2}\right]}. \qquad (4\text{-}10)$$

Considering $w/w_к=\lambda$, we can record:

$$w_{n2}=\varphi_1 w_{n.к}\lambda_{n2}, \qquad (4\text{-}11)$$

where ϕ_1 - coefficient of the velocity of the nozzle.

$$w_3=\frac{w_{c.к}}{\varphi_3}\lambda_{c3}; \qquad (4\text{-}12)$$

here ϕ_3 - coefficient of the velocity of the diffuser.

Inserting the value of the velocity in (4-10) from (4-11) and (4-12) and

$$v_2 = v_c\left(\frac{p_c}{p_2}\right)^{\frac{1}{\kappa_c}},$$

after corresponding conversions, we will obtain:

$$\nu = \frac{K_1 w_{н.н}\lambda_{н2}}{K_2 w_{c.н}\lambda_{c2}} - 1, \tag{4-13}$$

where

$$K_1 = \varphi_1\varphi_2\varphi_3; \tag{4-14}$$

$$K_2 = \left(\sqrt{\frac{1-\left(\frac{p_2}{p_c}\right)^{\frac{\kappa_c-1}{\kappa_c}}}{1-\left(\frac{p_1}{p_c}\right)^{\frac{\kappa_c-1}{\kappa_c}}}} + \varphi_3^2\frac{\kappa_c-1}{2\kappa_c}\times\right.$$

$$\left.\times\frac{\frac{p_2}{p_c}-\frac{p_1}{p_c}}{\left(\frac{p_1}{p_c}\right)^{\frac{1}{\kappa_c}}\sqrt{1-\left(\frac{p_1}{p_c}\right)^{\frac{\kappa_c-1}{\kappa_c}}}\sqrt{1-\left(\frac{p_2}{p_c}\right)^{\frac{\kappa_c-1}{\kappa_c}}}}\right)^{\frac{1}{\varphi_2}}. \tag{4-15}$$

By expressing λ from equation (3-27) and inserting its value in (4-13), we will have:

$$\nu = \frac{K_1}{K_2}\frac{w_{н.н}}{w_{c.н}}\sqrt{\frac{\left[1-\left(\frac{p_2}{p_{н0}}\right)^{\frac{\kappa_н-1}{\kappa_н}}\right]\frac{\kappa_н+1}{\kappa_н-1}}{\left[1-\left(\frac{p_2}{p_c}\right)^{\frac{\kappa_c-1}{\kappa_c}}\right]\frac{\kappa_c+1}{\kappa_c-1}}} - 1. \tag{4-16}$$

At this point, according to condition $w_{r2}=0,\ p_2=p_r$.

On the basis of the experimental investigations of vapor-water jet pumps, Sokolov and Zinger recommend using:

$\varphi_1=0{,}95$; $\varphi_2=0{,}975$; $\varphi_3=0{,}9$; $K_1=0{,}834$ and $K_2=1$.

In those cases where the ejected medium is a vapor, one can use the *I-S*-diagram to determine the coefficient of ejection. In this case equation (4-16) can be recorded in the form

$$v = 0{,}834\sqrt{\frac{H_п}{H_с}} - 1, \qquad (4-17)$$

where $H_п$ – adiabatic drop in heat during the expansion of the working vapor from pressure $p_{п0}$ to pressure p_2; $H_с$ – adiabatic drop in heat during the compression of the mixed flow from pressure p_2 to pressure $p_с$.

By virtue of the accepted assumptions during the derivations, equation (4-16) can be regarded only as very approximate and is useful only for the tentative preliminary calculation.

It is necessary to note that during the calculation based on equation (4-16) difficulties appear in the determination of the adiabatic index $\kappa_с$ and critical velocity $w_{с.к}$ of the mixture, since these values, in turn, depend on the coefficient of ejection.

The equation for the calculation of $\kappa_с$ can be obtained from simple relationships for a mixture of gases. Taking into account (1-71) we can record:

$$\kappa_с = \frac{C_{pс}}{C_{vс}} = \frac{g_п C_{pп} + g_г C_{pг}}{g_п C_{vп} + g_г C_{vг}},$$

where $g_п$ and $g_г$ – fraction of weight of the working vapor and ejected gas in the mixture.

Considering that $\frac{g_г}{g_п} = v$, after corresponding conversions we will obtain:

$$\kappa_с = \frac{\kappa_п + \kappa_г v a}{1 + v a}, \qquad (4-18)$$

where

$$a = \frac{C_{vг}}{C_{vп}}.$$

The equation for determining $w_{с.к}$ can be obtained from the law of the conservation of energy, recorded for initial and final sections of mixing chamber:

$$Q_з = Q_{п2} + Q_{г2}. \tag{4-19}$$

Taking into account the first law of thermodynamics

$$dQ = dU + A\frac{dw^2}{2g} + Ad(pv)$$

law of the conservation of energy can be recorded:

$$\left(\frac{w_3^2}{2g} + \frac{1}{A}C_{vc}T_{c3}\right)G_c + F_3p_3w_3 =$$
$$= \left(\frac{w_{п2}^2}{2g} + \frac{1}{A}C_{vп}T_{п2}\right)G_п + F_{п2}p_{п2}w_{п2} +$$
$$+ \left(\frac{w_{г2}^2}{2g} + \frac{1}{A}C_{vг}T_{г2}\right)G_г + F_{г2}p_{г2}w_{г2},$$

where $1/A$ – mechanical heat equivalent.

Using the equation: the state $p = g\rho RT$, the speed of sound $w_{зв}^2 = \kappa\frac{p}{\rho}$ and Mayer's formula $R = \frac{1}{A}(C_p - C_v)$, we will obtain

$$\frac{1}{A}C_vT = \frac{w_{зв}^2}{g\kappa(\kappa - 1)}.$$

Inserting this expression, and also $w = \frac{G}{F\rho g}$ and $\frac{p}{\rho} = \frac{w_{зв}^2}{\kappa}$ in the equation of the conservation of energy, we will have:

$$\left(\frac{w_3^2}{2} + \frac{w_{зв3}^2}{\kappa_c - 1}\right)G_c = \left(\frac{w_{п2}^2}{2} + \frac{w_{зв.п2}^2}{\kappa_п - 1}\right)G_п +$$
$$+ \left(\frac{w_{г2}^2}{2} + \frac{w_{зв.г2}^2}{\kappa_г - 1}\right)G_г. \tag{4-20}$$

By converting this expression, taking into account equations (3-21), (3-34) and $w_{зв}^2 = \kappa gRT$, we will obtain finally:

$$\frac{\kappa_г + 1}{\kappa_г - 1}\nu w_{г.к}^2 + \frac{\kappa_п + 1}{\kappa_п - 1}w_{п.к}^2 = \frac{\kappa_c + 1}{\kappa_c - 1}(1 + \nu)w_{c.к}^2. \tag{4-21}$$

If the adiabatic indices of miscible flows are not changed in the process of mixing, then for the determination of κ_c and $w_{c.к}$

the equations (4-18) and (4-21) can be used. To do this, preliminarily one can assume

$$w_{с.н} = \frac{w_{р.н} + w_{н.н}}{2}$$

and

$$\kappa_с = \kappa_р.$$

According to these values one can determine ν and then, recalculate $\kappa_с$ and $w_{с.н}$ according to equations (4-18) and (4-21); based on the obtained values again calculate ν and again check the values $\kappa_с$ and $w_{с.н}$. By such sequential approximations the value of the coefficient of ejection can be determined.

However, the calculation becomes indefinite, if the adiabatic indices of flow are changed in the process of mixing, since in this case one is no longer allowed to put values $\kappa_с$ and $w_{с.н}$ in equation (4-21), which the flows had at the front end of mixing chamber. Actually in many cases the adiabatic index of the working vapor can be changed in the process of mixing.

As was already indicated, the processes, going on in mixing chamber, are connected with losses of energy: upon impulse according to equation (4-4) with a friction loss, and also with losses, caused by shock waves. With heat-insulated flow (without the transmission of heat into the environment) one may assume that the heat losses in the chamber goes on wholly through the increase in heat content of the mixture. In this case, if working vapor at the nozzle outlet is humid, then its drying occurs, and if the vapor is dryly saturated, then its overheating occurs. In both cases the adiabatic index of the vapor in the process of mixing is changed [see equation (1-111) for water vapor]. Only in case of superheated steam upon entry in mixing chamber does the adiabatic index $\kappa_п$ remain constant in the process of mixing.

For a case when the adiabatic index of the working vapor is changed in process of mixing, an approximate thermodynamic calculation of value $\kappa_с$ perhaps can be made under the assumption that the process of mixing proceeds isobarically, i.e., $p_2 = p_3$ (shock waves are absent).

Let us determine, firstly, the loss of energy during mixing. With this aim equation (4-4) can be converted, taking into account the loss factors and going under the assumption that $w_{r2} = 0$, into the following calculated form [78]:

$$q = \frac{\Delta E}{G_n} = \frac{1}{8\,380}\left[1 - \frac{\varphi_2^2}{\Psi_2^2(1+\nu)}\right] w_{n2}^2 \text{ [kcal/kg]}, \tag{4-22}$$

where Ψ_2 – coefficient, accounting for the irregularity of velocity distribution through the section of mixing chamber. According to [78] $\Psi_2 \approx 0{,}94$.

Inserting the values $\phi_2 = 0.975$ and $\Psi_2 = 0.94$ in (4-22), we will have:

$$q = \frac{1}{8\,380}\left[1 - \frac{1{,}07}{1+\nu}\right] w_{n2}^2 \text{ [kcal/kg]}. \tag{4-23}$$

Assuming that heat q is directed wholly to drying the working vapor, we will obtain the heat content of the vapor at the end of mixing

$$I_{n3} = I_{n2} + q. \tag{4-24}$$

According to the thermodynamic tables we can determine temperature t''_{n3}, the heat content I_{n3} and the heat of vaporization Λ for a dryly saturated vapor at a pressure of $p_3 = p_2$. Further, we can determine the degree of dryness vapor at the end of the mixing chamber according to equation

$$x_3 = 1 - \frac{I''_{n3} - I_{n3}}{\Lambda_3}. \tag{4-25}$$

Now for given x_3 and t''_{n3} the adiabatic index of the working vapor at the end of mixing can be determined. Thus, for water vapor the equation (1-111) can be used. The calculated adiabatic index κ_n thusly is put in equation (4-18) and κ_c is determined.

If the ejected gas consists of a vapor-air mixture, then the heat content of the mixture of working and ejected vapor at the end of the mixing chamber is equal to:

$$I_c = \frac{G_p I_{p2} + G'_n I'_{n2}}{G'_n + G_p}, \qquad (4\text{-}26)$$

where G'_n – quantity of vapor in the drawn-off vapor-air mixture $(G_r = G'_в + G'_n)$; I'_n – heat content of the dryly saturated vapor, determined according to the temperature of incoming vapor-air mixture t_r in jet pump.

Inserting $g_n = \frac{G'_n}{G_r}$, in (4-26) we will obtain:

$$I_c = \frac{I_{p2} + g_n u I'_{n2}}{1 + g_n u}. \qquad (4\text{-}27)$$

In accordance with equations (1-31) and (1-33) the partial pressure of vapor in the vapor-air mixture at the end of mixing chamber can be recorded as

$$p_{n3} = \frac{p_3}{1 + \frac{R_в}{R_n}\frac{G'_в}{G_p + G'_n}} = \frac{p_3}{1 + \frac{R_в}{R_n}\frac{(1-g_n)u}{1+g_n u}}, \qquad (4\text{-}28)$$

where $R_в$ and R_n – gas constants of air and vapor. According to the thermodynamic tables we can determine the temperature, heat content and heat of vaporization for dryly saturated vapor at pressure p_{n3}. Then, according to equation (4-25) we can determine x, according to equation (1-111) – κ_n and, finally, we can determine κ_c according to equation (4-18), which for a case of pumping out the vapor-air mixture, acquires the form:

$$\kappa_c = \frac{\kappa_n(1+g_n u) + \kappa_в(1-g_n)ua}{1 + g_n u + (1-g_n)ua}. \qquad (4\text{-}29)$$

For the examined case of mixing with the loss of energy, the equation of the conservation of energy (4-21) is no longer useful for the determination of $w_{с.к}$. In this case we can determine $w_{с.к}$ by parameters of flow behind the diffuser in accordance with (3-19), i.e.,

$$w_{с.к}=\sqrt{2g\frac{\kappa_с}{\kappa_с+1}R_сT_с}. \tag{4-30}$$

We can determine the gas constant of mixture $R_с$ according to (1-31) from the equation

$$R_с=\frac{R_в(1+g_п\nu)+\nu(1-g_п)R_п}{1+\nu}. \tag{4-31}$$

We can determine temperature $t_с$ by the partial pressure of the vapor $p_{п.с}$ in the mixture behind the diffuser. Partial pressure of vapor $p_{п.с}$ can be determined from the equation (4-28), by inserting pressure $p_с$ in the equation instead of p_3, equal to the given value of the maximum outlet pressure of jet pump. According to the thermodynamic tables we can find temperature $t_с$ for dryly saturated vapor at pressure $p_{п.с}$.

By inserting $g_п = 0$ in (4-28), (4-29) and (4-30), we will obtain the equation for a case of pumping out of pure air.

The given reasonings, definitizing the calculation of the coefficient of ejection during the operation of jet pump on saturated vapor, are accurate not only with respect to equation (4-16), but also in the case of the calculation of the characteristics of the jet pump.

In addition to that said above it is necessary to note that along with losses in mixing chamber, leading to a change in the adiabatic index of the vapor, losses also take place in the diffuser which can lead to a change in the adiabatic index of vapor during flow of the vapor-gas mixture in the diffuser. This circumstance, not considered in the calculation, along with the absence of heat insulation the flow in a real case can lead to a certain deviation of the real characteristics of the jet pump from that calculated.

Thus, as was already noted above, equation (4-16) permits one to determine the approximate value of the coefficient of ejection, necessary for the preliminary, tentative design of the dimensions of the jet pump. In this case, the accuracy of the design according to equation (4-16) is less the less value ν is. It is not difficult to see that when the value $\nu \approx 0.1$, inaccuracies in the calculation of the first member of the equation leads to an error of 10% in the determination of ν based on 100%, and when values of $\nu \approx 0.01$ to such an error an inaccuracy of 1% in the calculation of the first member is already made (!). In connection with this, the use of equation (4-16) for the calculation of $\nu < 0.1$ becomes practically impossible.

The basic transverse dimensions of the jet pump are determined by two basic geometric parameters $f_{п1}/f_{п.к}$ and $f_3/f_{п.к}$. The expansion of the nozzle $f_{п1}/f_{п.к}$ is calculated according to the relationship (3-30), in which λ is determined from equation (3-27) based on the assigned $p_{п0}$ and p_2. Parameter $f_3/f_{п.к}$ can be calculated according to the relationship, obtained from the equation of consumption:

$$f_3 w_3 \rho_3 = (1+\nu) f_{п.к} w_{п.к} \rho_{п.к}. \qquad (4\text{-}32)$$

Hence, taking into account (3-18) and (3-25) after simple conversions we will obtain:

$$\frac{f_3}{f_{п.к}} = \frac{\kappa_п}{\kappa_с}\frac{\kappa_с+1}{\kappa_п+1}\frac{\left(\frac{2}{\kappa_п+1}\right)^{\frac{1}{\kappa_п-1}}}{\left(\frac{2}{\kappa_с+1}\right)^{\frac{1}{\kappa_с-1}}}\frac{w_{с.к}}{w_{п.к}}\frac{p_п}{p_с}\frac{1}{q_{с3}}(1+\nu), \qquad (4\text{-}33)$$

where

$$q_{с3} = \left(\frac{\kappa_с+1}{2}\right)^{\frac{1}{\kappa_с-1}}\lambda_{с3}\left(1-\frac{\kappa_с-1}{\kappa_с+1}\lambda_{с3}^2\right)^{\frac{1}{\kappa_с-1}}. \qquad (4\text{-}33a)$$

Considering the large compression ratio of the gas in the jet pump, one can assume that the compression in the diffuser is critical, which in many cases, turns out to be accurate. In this case $q_{c3} = 1$, and the equation (4-33) acquires the form:

$$\frac{f_3}{f_{\text{н.к}}} = z\,\frac{w_{\text{с.к}}}{w_{\text{н.к}}}\,\frac{p_{\text{н}}}{p_{\text{с}}}\,(1+\nu), \tag{4-34}$$

where

$$z = \frac{\kappa_{\text{н}}}{\kappa_{\text{с}}}\,\frac{\kappa_{\text{с}}+1}{\kappa_{\text{н}}+1}\,\frac{\left(\dfrac{2}{\kappa_{\text{н}}+1}\right)^{\frac{1}{\kappa_{\text{н}}-1}}}{\left(\dfrac{2}{\kappa_{\text{с}}+1}\right)^{\frac{1}{\kappa_{\text{с}}-1}}}.$$

Values determined by equations (4-16) and (4-34) are preliminary and should be checked in terms of the equation of characteristics.

The equation of characteristics can be obtained from the equation of momentum, recorded for initial and final sections of mixing chamber, taking into account the reaction of the wall of the conical section (effuser) of the mixing chamber

$$p_{\text{п}2} f_{\text{п}2} + p_{\text{р}2} f_{\text{р}2} - \int_{f_2}^{f_3} p\,df - p_3 f_3 + \varphi_2 \frac{G_{\text{н}}}{g} w_{\text{п}2} + \\ + \varphi_2 \frac{G_{\text{р}}}{g} w_{\text{р}2} - \frac{G_{\text{н}} + G_{\text{р}}}{g} w_3 = 0. \tag{4-35}$$

Assuming $p_{\text{п}2} = p_{\text{п}1}$; $f_{\text{п}2} = f_{\text{п}1}$; $w_{\text{п}2} = w_{\text{п}1}$; $f_{\text{р}2} = f_2 - f_{\text{п}1}$ and linear increase in pressure in the effuser, Zinger [45, 46] derives the equation of the characteristics of vapor-jet pump with mixing chamber, consisting of conical and cylindrical sections:

$$\frac{p_{\text{с}}}{p_{\text{р}}} = \frac{1}{\Pi_{\text{с}3}} \left\{ \Pi_{\text{н}3} \frac{p_{\text{н}}}{p_{\text{р}}} \frac{f_{\text{н}3}}{f_3} \frac{1}{1 + \frac{a}{3}\Phi_3} + \Pi_{\text{р}3} \frac{f_{\text{р}3}}{f_3} \frac{1 - \frac{1}{3}\frac{f_3}{f_{\text{р}3}}\Phi_1}{1 + \frac{a}{3}\Phi_3} + \right. \\ + \frac{\kappa_{\text{н}} \Pi_{\text{н.к}}}{\varphi_3} \frac{f_{\text{н.к}}}{f_3} \frac{p_{\text{н}}}{p_{\text{р}}} \frac{1}{1 + \frac{a}{3}\Phi_3} \times \\ \left. \times \left[K_1 \lambda_{\text{н}3} + K_2 \nu \frac{w_{\text{р.к}}}{w_{\text{н.к}}} \lambda_{\text{р}3} - (1+\nu) \frac{w_{\text{с.к}}}{w_{\text{н.к}}} \lambda_{\text{с}3} \right] \right\}; \tag{4-36}$$

$$\Phi_1 = 2\frac{f_2}{f_3} - \sqrt{\frac{f_2}{f_3}} - 1;$$
$$\Phi_2 = \frac{f_2}{f_3} + \sqrt{\frac{f_2}{f_3}} - 2; \qquad \text{(4-36 cont'd)}$$

$a = p'_3/p_3$ — pressure ratio at the beginning and end of the cylindrical section of the mixing chamber. According to Zinger [46], a = 0.5-0.75, where the larger values correspond to the short cylindrical section, the smaller values – to the long; $\Pi_{c3} = p_3/p_c$; $\Pi_{г3} = p_3/p_г$; $\Pi_{п3} = p_3/p_{п0}$ are calculated with respect to (3-27) and $\Pi_{п.к} = p_{п.к}/p_{п0}$ — with respect to (3-19); $w_{п.к}$ and $w_{г.к}$ can be determined according to the equations (3-21); $\lambda_{п2}$ – according to the equation (3-27)

Value $w_{c.к}$ is calculated according to method, described above in the calculation of the preliminary value, ν.

Value $\lambda_{г2}$ is determined according to the function

$$q_{г2} = \frac{f_{г.с}}{f_{г2}} = \left(\frac{\kappa_г + 1}{2}\right)^{\frac{1}{\kappa_г - 1}} \lambda_{г2}\left(1 - \frac{\kappa_г - 1}{\kappa_г + 1}\lambda_{г2}^2\right)^{\frac{1}{\kappa_г - 1}} = \frac{\kappa_п}{\kappa_г}\,\frac{w_{г.к}}{w_{п.к}}\,\frac{\Pi_{п.к}}{\Pi_{г.к}}\,\frac{p_п}{p_г}\,\frac{f_{п.к}}{f_{г2}}\,\nu, \qquad \text{(4-37)}$$

where $f_{г2} = f_2 - f_{п2}$.

Value λ_{c3} can be determined according to function q_{c3} from the equation (4-33a). $\lambda_{г2}$ and λ_{c3} have a value less than unity.

The ratio of the areas of sections are defined as:

$$\frac{f_{п2}}{f_3} = \frac{f_{п1}}{f_3} = \frac{f_{п1}}{f_{г.с}} : \frac{f_3}{f_{г.с}}; \qquad \text{(4-38)}$$

$$\frac{f_{г2}}{f_3} = \frac{f_2}{f_3} - \frac{f_{п2}}{f_3}. \qquad \text{(4-39)}$$

The ratio of the inlet section effuser to the section of the cylindrical part of the mixing chamber, f_2/f_3, according to [45] should lie within the limits, 2-2.8.

The coefficients $K_1=\varphi_1\varphi_2\varphi_3$ and $K_2=\varphi_2\varphi_3\varphi_4$ according to [45] have values of $K_1 = 0.834$ and $K_2 = 0.812$. Here, ϕ_4 – coefficient of velocity, accounting for the loss of ejected flow; $\phi_4 = 0.925$.

As can be seen from equation (4-36), the characteristics of vapor-jet pump depends not on its absolute geometric dimensions, but on the relative geometric parameters $f_3/f_{п.к}$ and $f_{п1}/f_{п.к}$, which are the main parameters of similarity of the jet pumps.

In order to construct the characteristics of a jet pump the desired values $p_г$ and $p_с$ are prescribed. Usually, one constructs characteristics for variable $p_г$ at a constant value, $p_с$. For the assumed $p_г$ and $p_с$ based on the equation (4-16) one can determine the preliminary value of the coefficient of ejection ν. Then, all the values, entering into equation (4-36) are determined, and compression ratio $p_с/p_г$ is calculated. If the obtained value does not coincide with the preliminarily assumed value, then new values $p_с$ and $p_г$ are assigned and recalculations made, seeking results by a method of gradual approximations of coincidence.

Typical characteristics of a jet pump during various values of a basic geometric parameter $f_3/f_{п.к}$ are shown in Fig. 4-7 [45]. As can be seen, the characteristics have two sections: *AB*, on which the change of the compression ratio does not result in a change of the coefficient of ejection, and *BC*, on which small changes of the compression ratio result in considerable changes in the coefficient of ejection. Sections *BC* and *AB* correspond to the working and transfer sections of the characteristics in Fig. 4-4. At point *B* the coefficient of ejection has a maximum value, corresponding to the so-called maximum rating of the operation of a jet pump. Accordingly, under these conditions we have a maximum coefficient of ejection and a maximum counterpressure, equal to the peak output pressure of the jet pump.

As the experiment shows [42, 43, 47, 48], the critical speed of the mixed flow in the throat of the diffuser corresponds to the maximum rating of operation of jet pump.

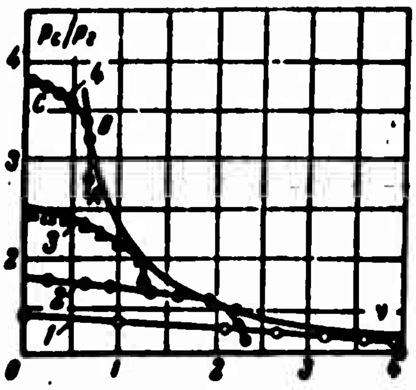


Fig. 4-7. Characteristics of $p_c/p_г = f(\nu)$ of a vapor-jet pump at various values of $f_3/f_{п.к}$. Curve 1 – $f_3/f_{п.к}$ = 1,800; curve 2 – $f_3/f_{п.к}$ = 970; curve 3 – $f_3/f_{п.к}$ = 500; curve 4 – $f_3/f_{п.к}$ = 260.

For an explanation of the mechanism of the onset of the maximum rating let us examine the operation of jet pump during variable counterpressure (Fig. 4-8). Let us assume that the counterpressure behind a jet pump has a certain value $p_{c(0)}$, whereby the coefficient of ejection is $\nu = 0$. Under this condition, as already noted above, in mixing chamber a compression shock appears, after which the flow rate becomes subsonic and then systematically decreases in the diffuser with a simultaneous increase in pressure. In this case all the energy of the working flow is expended in overcoming the resistance of the apparatus and restoration of the initial pressure. In spite of the fact that pressure ratio, $p_г/p_c$ in this case is less than critical, in the throat of the diffuser the pressure is set higher than the critical. If one now were to start lowering counterpressure p_c, then a part of the energy of the working flow expended in overcoming the pressure, would be freed, thanks to that which becomes available for the ejection of gas and its compression up to the outlet pressure. According to the lowering of pressure p_c, the rarefaction wave spreading upwards along the flow with the speed of sound, will cause the reconstruction of the flow to the front of compression shock; in this case, pressure p_3 in the throat does not decrease, and the coefficient of ejection of the apparatus does increase. This operating condition corresponds to section *BC* of the characteristics in Fig. 4-7. Thus, it will continue as long as the counterpressure does not drop to a certain limit of value $p_{c(пр)}$, where pressure p_3 and speed w_3 will attain their critical values. With further lowering of counterpressure, the rarefaction wave cannot even penetrate in the throat of the diffuser, and

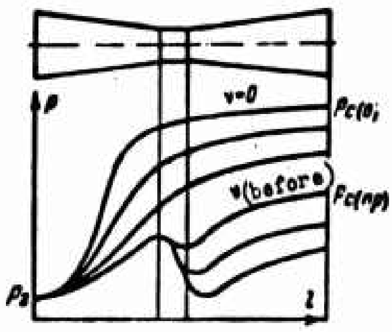


Fig. 4-8. Distribution of pressure through the jet pump under variable operating conditions.

consequently, the state of flow behind the throat up to the diffuser, will remain constant. Accordingly, the coefficient of ejection, by attaining its maximum limit of value ν_{np}, will not even be changed (section *AB* in Fig. 4-7). When $p_c < p_{c(np)}$, the flow rate will already increase in the diffuser up to supersonic, and the pressure will drop as long as a compression shock will not occur in a certain section, and the speed again will not become subsonic. According to the lowering of the counterpressure, the position of front of the shock is displaced towards the outlet section of the diffuser (Fig. 4-8).

The design of the characteristics of the jet pump based on the maximum rating can be done according to the equation (4-36) with the permutation of parameters in it, corresponding to the critical flow in the throat of the diffuser.

With help of the equation of constancy of flow rate one can obtain a relationship for the determination of the maximum counter-pressure $p_{c(np)}$:

$$G_{c(np)} = G_n [1 + \nu_{np}]. \qquad (4\text{-}40)$$

The equation of the flow rate of vapor through the nozzle is $G_n = f_{n.k} w_{n.k} g \rho_{n.k}$ taking into account that (3-19), (3-20) and (3-21) can be converted into the form:

$$G_n = \frac{\kappa_n g \Pi_{n.k} p_{n0} f_{n.k}}{w_{n.k}}. \qquad (4\text{-}41)$$

Analogously

$$G_{c(\text{пр})} = \frac{\kappa_c g \Pi_{c.\text{к}} p_{c(\text{пр})} f_c}{w_{c.\text{к}}}. \qquad (4\text{-}42)$$

By solving jointly equations (4-40), (4-41), and (4-42), we obtain:

$$p_{c(\text{пр})} = \frac{\kappa_\text{п}}{\kappa_c}\frac{\Pi_{\text{п.к}}}{\Pi_{c.\text{к}}}\frac{w_{c.\text{к}}}{w_{\text{п.к}}}\frac{f_{\text{п.к}}}{f_c}\, p_{\text{н0}}(1+\nu_{\text{пр}}), \qquad (4\text{-}43)$$

where $\Pi_{\text{п.к}}$ and $\Pi_{\text{с.к}}$ – critical pressure ratios in the nozzle and throat of the diffuser determined by equation (3-19).

Given in Fig. 4-9 is a family of curves $p_\text{с} = f(\nu)$ for various inlet pressures $p_\text{г}$. In section 1-2 characteristics with a decrease of $p_\text{с}$, the coefficient of ejection increases, and at point 2 at a maximum counterpressure $p_{\text{с(пр)}}$ the maximum coefficient of ejection is attained. A further decrease of $p_\text{с}$ does not change ν. Dot-dash line *a*-*b* is the line of maximum counterpressures. The intersection of this line with line *c*-*d* of the actual counterpressures determines the point of overload of the characteristics of a jet pump $p_\text{г} = f(\nu)$ at point g. At coefficients of ejection smaller than that at point *e* of the intersection of lines *a*-*b* and *c*-*d*, the jet pump operates at a maximum rating (branch *f*-g characteristics), and at coefficients of ejection larger than that at point *e*, – at a near-maximum rating (overloaded branch g-*h*).

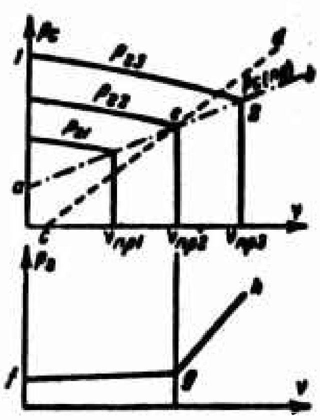


Fig. 4-9. Dependence of the characteristics of the jet pump on the counterpressure at various inlet pressures ($p_{\text{г}3} > p_{\text{г}2} > p_{\text{г}1}$).

Thus, by formulating the line of maximum counterpressures according to equation (4-43) according to its intersection with the line of actual counterpressures, we find the point of overload of the characteristics of the jet pump.

As can be seen from (4-43) and Fig. 4-9, the value of maximum counterpressure depends on the coefficient of ejection, since there is a minimum value when $\nu = 0$.

The value of the maximum coefficient of ejection determines the maximum speed of the ejected flow in the entrance section of mixing chamber with given dimensions of the jet pump. This speed in all cases cannot be more than the critical. Therefore, the value of the maximum coefficient of ejection cannot be larger than that, which is set when achieving the critical speed of the ejected flow. In connection with this Sokolov and Zinger [45] introduced the idea of a so-called second maximum rating, which is characterized by the fact that in a given section of the effuser, f_8, a critical speed of the ejected flow is attained. The maximum coefficient of ejection can be determined in this case by the equation

$$\nu_{\text{кр}} = \left(\frac{f_8}{f_{\text{п.к}}} - \frac{1}{q_{\text{н.р}}}\right)\frac{K_{\text{р}}}{K_{\text{н}}}\frac{\Pi_{\text{р.у}}}{\Pi_{\text{н.к}}}\frac{w_{\text{н.к}}}{w_{\text{р.к}}}\frac{p_{\text{р}}}{p_{\text{н0}}}. \qquad (4\text{-}44)$$

From (4-44) we obtain, accordingly

$$\frac{f_8}{f_{\text{п.к}}} = \frac{1}{q_{\text{н.р}}} + \frac{w_{\text{р.к}}}{w_{\text{н.к}}}\frac{K_{\text{н}}}{K_{\text{р}}}\frac{\Pi_{\text{н.к}}}{\Pi_{\text{р.к}}}\frac{p_{\text{н0}}}{p_{\text{р}}}\nu_{\text{кр}}, \qquad (4\text{-}45)$$

where $\nu_{\text{кр}}$ – maximum coefficient of ejection, determined by equation (4-16).

One can also obtain the relationship

$$\frac{f_8}{f_3} = \frac{w_{\text{н.к}}}{w_{\text{с.к}}}\frac{p_{\text{с}}}{p_{\text{н0}}}\frac{1}{1+\nu_{\text{кр}}} - \frac{1}{q_{\text{н.р}}} + \frac{w_{\text{р.к}}}{w_{\text{с.к}}}\frac{K_{\text{н}}}{K_{\text{р}}}\frac{\Pi_{\text{н.к}}}{\Pi_{\text{р.к}}}\frac{p_{\text{с}}}{p_{\text{р}}}\frac{\nu_{\text{кр}}}{1+\nu_{\text{кр}}}. \qquad (4\text{-}46)$$

According to Sokolov and Zinger, the ratio f_8/f_3, determined by equation (4-46), should not exceed 1.5-2.0. Otherwise, the coefficient of ejection ν, entering in this equation, cannot be realized.

The calculation according to equation (4-44) carries a conditional character, since the ratio $f_8/f_{\text{п.к}}$ cannot be calculated analytically and should be determined from the experiment. Therefore, in the construction of the characteristics of a newly designed jet pump with a value $f_{s}/f_{п.к}$ there are necessary steps, taking into account that $f_{s}/f_{3} \leq 1.5$-2.0. Sokolov and Zinger affirm that the characteristics built for real jet pumps according to equation (4-44) with use of the experimental value f_{s}/f_{3}, agree well with the empirical curves. Meanwhile, the experimental value, f_{s}/f_{3} (or $f_{s}/f_{п.к}$) was determined namely by means of comparison of the experimental data with the equation (4-44). Therefore, equation (4-44) can be more readily regarded as a certain semi-empirical dependence, describing the character of change of the parameters of the jet pump.

The question whether a second maximum rating in real jet pumps is attained in reality and whether it can be regulated as compared to the maximum rating at a critical speed in the throat of the diffuser, now remains indefinite. At the same time, in the above mentioned works on the study of the maximum rating at a critical speed in the throat of the diffuser, good agreement is indicated between the calculated characteristics and the experimental.

Let us go back to the calculation of the dimensions of the jet pump. Thus, the calculation is conducted in two stages: the preliminary one according to the equation (4-16) and the final one according to the equation of characteristics (4-36) for the maximum rating. Based on the calculated value $\nu_{пр}$ the vapor flow rate is determined through the nozzle from equation (4-6). Then according to equation (3-22) the area of the critical section of the nozzle, $f_{п.к}$ is determined and according to equation (3-30), also the geometric parameter of the nozzle, $f_{п1}/f_{п.к}$. In this case λ is calculated for the maximum operating pressure $p_{г}$. The expansion angle of nozzle, based on experimental data is taken to be equal to 10-20° (see Chapter III, Section 3-3b). With large expansions of the nozzles, optimum results are sometimes obtained at angles of 25-30°. According to equation (4-34) the basic geometric parameter $f_{3}/f_{п.к}$ is determined and diameter of the throat d_{3} is calculated. The diameter of the entrance section of the effuser d_{2}, is taken based on experimental data. Sokolov and Zinger [45] recommend selecting it, proceeding from relationship $f_2/f_3 = 2-2.8$. Vigand [37] in the calculation of vapor-water jet pumps determines it, proceeding from the speed of the ejected flow in an annular section between the nozz'? and the effuser $w_{r2} \leq 90$ m/s. The understanding of diameter d_2 results in a considerable decrease of the coefficient of ejection (Fig. 4-10). It is recommended to take 5-7° as the cone angle of the effuser according to many investigations.

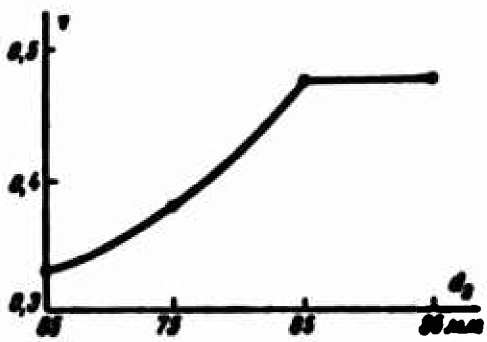


Fig. 4-10. Change in the coefficient of ejection with a change in the diameter of the entrance section effuser $(L=8d_3,\ \xi=p_c/p_r=5)$.

The length of the mixing chamber, as already indicated, plays an essential role in safeguarding the required characteristics of the jet pump. With a length of mixing chamber less than optimum, the uniform distribution of the parameters of mixture before input in the diffuser is not ensured; this leads to considerable losses in the diffuser and a lowering of the efficiency of the jet pump. Optimum value of length L of the mixing chamber depends on the compression ratio $\xi=p_c/p_r$ of the ejected gas and the expansion ratio $E=p_a/p_r$ of the working vapor. Usually it is recommended to take

$$L=(6-10)\, d_3. \qquad (4\text{-}47)$$

where L - distance from nozzle outlet section up to the throat of the diffuser. Vigand, on the basis of experimental data for vapor-water jet pumps, recommends to select L/d_3, depending upon value ξ according to diagram (Fig. 4-11). The diagram was checked in the range $E = 300-2{,}000$. During construction of a jet pump one should select the coning of the effuser and inlet diameter d_2 in such a way as to ensure an optimum value L.

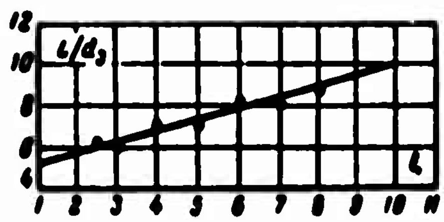


Fig. 4-11. Diagram for the determination of the optimum ratio L/d_3.

The selection of the outlet diameter of the diffuser determines the velocity at the outlet section and accordingly, the degree of restoration of pressure. The less the speed at the outlet, the bigger the value of the static pressure of flow. Consequently, to produce a possibly greater static pressure behind the diffuser it is necessary to have a greater expansion of the diffuser. However, in this case, the coning angle of the diffuser should not exceed, accordingly, the numerous experimental data by 5-8°. Therefore, the large expansion of the diffuser will result in its greater length and accordingly, to increased losses. Optimum results are obtained from the data of Vigand when $d_c = (1{,}6-2)\ d_3$.

d) *Simplified calculation*. Above described theoretical method of calculating the coefficient of ejection ν and the basic geometric parameter of the jet pump $f_3/f_{п.к}$ is, as one may see, very cumbersome and complex, and it not always ensures good agreement with the experiment. Naturally therefore, there is a tendency to use simplified methods of calculation in the engineering calculations, based on the generalization of the experimental data. Such methods, of course, cannot be universal. They are based on the similarity of jet pumps of a specified construction, operating under specified conditions and with specified working media, and therefore, are applicable only in the calculation of such kinds of jet pumps.

There are an adequate number of experimental methods for calculating vapor-water jet pumps; however, the most tested and widely accepted is the Vigand method [37]. To calculate the coefficient of ejection, Vigand recommends a diagram, built on experimental data and repeatedly tested (Fig. 4-12). According to this diagram the coefficinet of ejection is determined depending upon expansion ratio E of the working vapor in the nozzle and the compression ratio ξ of the pumped gas. Vigand uses this diagram both in the case of pumping out air, and in the case of pumping out a vapor-air mixture of any composition. During the pumping out of vapor calculated according to the diagram, the obtained coefficient of ejection is underrated which provides approximately a 20-30% reserve based on the expended vapor.

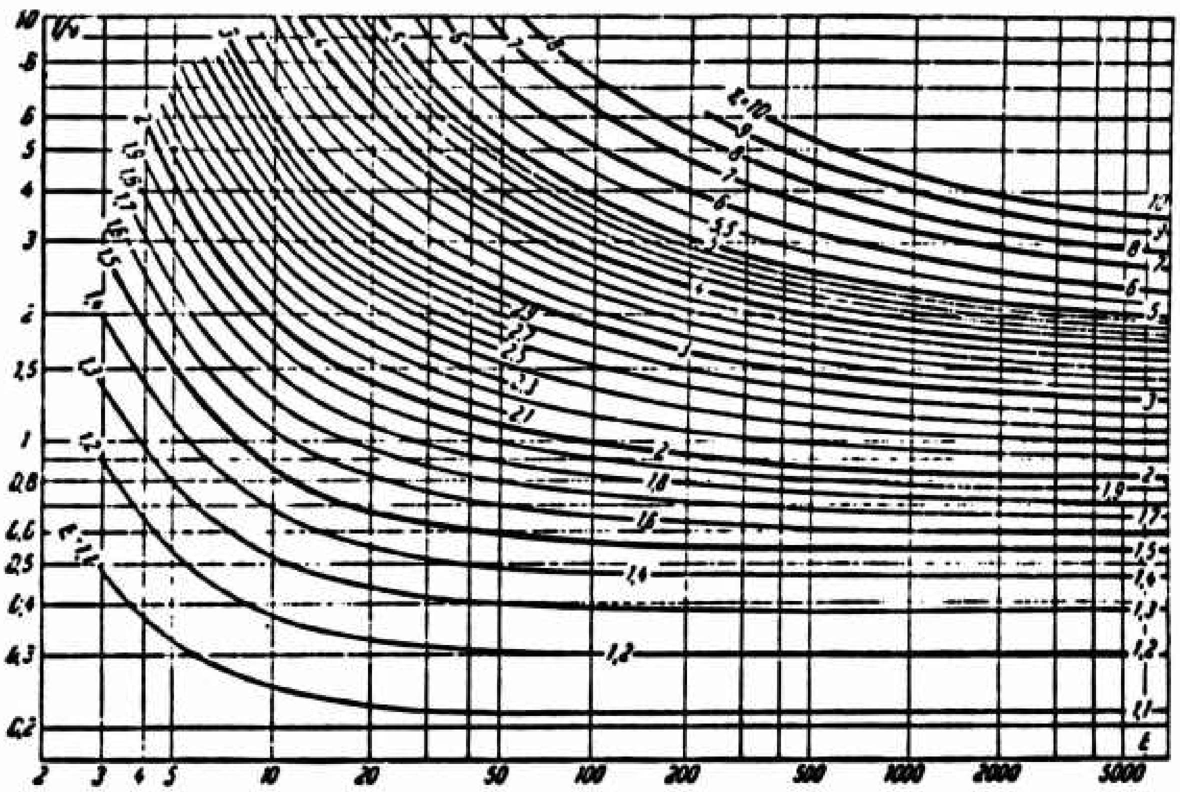


Fig. 4-12. Diagram for the determination of the coefficient of ejection.

Diagram is real for the calculation of jet pumps with a diameter of the throat of 10-100 mm. With diameters larger than 100 mm computed value ξ should increase by 5%. Diameters of the throat larger than 250 mm are not recommended for use. With a pressure of the working vapor higher than 15 atm(tech.) the diagram gives a somewhat overrated value ν.

Vigand determines the area of the section f_3 of the throat of the diffuser, proceeding from the assumption that a critical speed is not attained in the throat.

$$f_3 = \frac{G_c}{3600 \cdot 100 \sqrt{\frac{p_c}{v_c}}\,\phi}\ [m^2], \qquad (4\text{-}48)$$

where G_c – flow rate of the mixture, kg/h; p_c – counterpressure, kg/cm^2; v_c – specific volume of the mixture with p_c (taken just as for saturated vapor), m^3/kg; ϕ – correction factor, depending on E and ξ, and determined according to the diagram (Fig. 4-13).

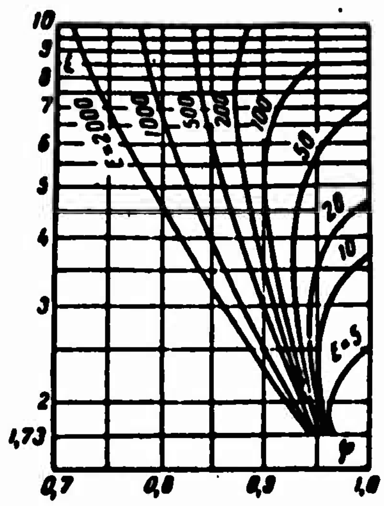


Fig. 4-13. Values of the correction factor for the equation (4-48).

e) *Effect of the geometric parameters and operating conditions on the characteristics of the jet pump.* The equation of the characteristics (4-36) and experimental investigations permit one to estimate effect of the different geometric and conditioning factors on the operation of the jet pump.

The basic geometric parameter. Effect of the basic geometric parameter $f_3/f_{п.к}$ on the characteristics of jet pump can be seen from Fig. 4-7. A decrease in $f_3/f_{п.к}$ leads to an increase of the compression ratio and a decrease in the maximum coefficient of ejection.

The length of the cylindrical part of mixing chamber. In a vapor-jet jet pump with a conical mixing chamber the increase in pressure of the mixture occurs both in the effuser and in the throat. Therefore, jet pumps with a developed conical section of the mixing chamber are sometimes made with a short cylindrical section. Thus, Vigand recommends taking the length of the throat, equal to its diameter. However, according to a number of investigations [43, 45] the length of the throat substantially affects the characteristics of the ejector. Figure 4-14 gives characteristics of a jet pump with a different form and dimensions of the flow-through unit. As can be seen from the figure, the smallest value of maximum output is obtained with the diffuser and the short cylindrical section lacking. At the same time the lengthening of the cylindrical section from $l_3 = d_3$ to $l_3 = 6d_3$ in the presence of the diffuser increases the maximum output of the jet pump by almost 1.5 times. According to the data [43] the length of the throat of $l = 4d_3$ is optimum.

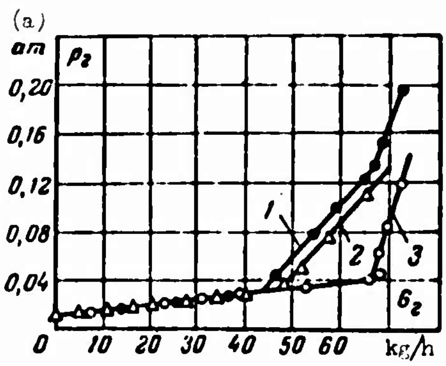


Fig. 4-14. Characteristics of the jet pump with different forms of the flow-through unit. 1 – conical mixing chamber with a throat $l = d_3$, diffuser is lacking; 2 – conical mixing chamber with a throat $l = d_3$ and a normal diffuser; 3 – conical mixing chamber with $l = 6d_3$ and a normal diffuser. KEY: (a) atm(tech.).

Distance between the nozzle and the mixing chamber. The distance between the nozzle and mixing chamber affects the characteristics of the jet pump. Figure 4-15 gives the dependence of the maximum coefficient of jet pump on the distance l_1 between the nozzle and mixing chamber at different values of a basic geometric parameter $f_3/f_{п.к}$. As can be seen, with a small $f_3/f_{п.к}$ the coefficient of ejection with an increase l_1 at first slightly increases to a certain maximum value, but then rapidly decreases. At more significant values of $f_3/f_{п.к}$, this dependence is more weakly expressed. Vigand recommends positioning the nozzle at a distance $l_1 = (0\text{-}1)\ d_3$.

Other geometric factors. Among the other geometric factors the coaxial alignment of the nozzle and diffuser are highly important in the production of the optimum characteristics of a jet pump. Experimentation shows that misalignment can substantially lower the maximum counterpressure and coefficient of ejection.

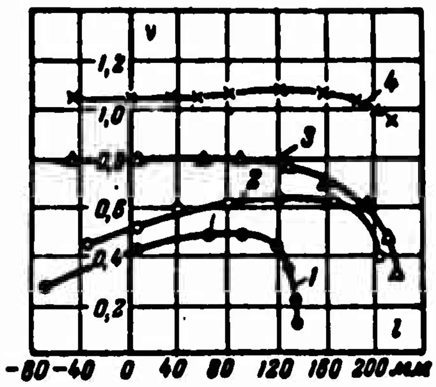


Fig. 4-15. Dependence of the maximum coefficient of ejection on the distance between the nozzle and mixing chamber at different values of $f_3/f_{п.к}$ [52]: $(f_3/f_{п.к})_1 < (f_3/f_{п.к})_2 < (f_3/f_{п.к})_3 < (f_3/f_{п.к})_4$.

Also the form of the inlet section of mixing chamber plays an important role. It is necessary to provide it with a smooth curvature formula (Fig. 4-16).

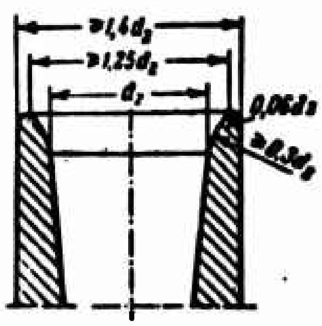


Fig. 4-16. Form of the inlet section of the mixing chamber [37].

The expansion of the nozzle $f_1/f_{п.к}$ also renders a certain effect on the characteristics of the jet pump, since this effect is even greater the less the coefficient of ejection is. With small values of parameter $f_1/f_{п.к}$ (less than the calculated), the expansion of working flow up to pressure p_2 upon entrance in mixing chamber occurs behind the nozzle with losses. In this case the compression ratio and the maximum coefficient of ejection drop. With an increase of $f_1/f_{п.к}$ the static pressures of the working flow from the nozzle and ejected flow upon entry in the mixing chamber can be compared, the losses from eddy effects disappear and the value of compression ratio and the coefficient of ejection increase. Therefore, a nozzle with an expansion close to that calculated is recommended for use. Vigand [37] recommends using an expansion of a nozzle equal to 97% of the calculated one.

It is necessary to note that with large expansions the losses in the nozzles can be considerable due to the great thickness of the boundary layer and the sporatic condensation usually appearing (see Section 3-3). Therefore, the characteristics of jet pumps with such nozzles, calculated taking into account the usually accepted values for losses, can substantially differ from the actual characteristics. At present any kind of information on values of losses in jet pumps with an expansion of vapor, E > 10,000, is lacking; this seriously hampers the design of such jet pumps.

Pressure of working vapor. The change in the pressure of working vapor substantially affects the characteristics of the jet pump, since in this case flow rate and available energy of the vapor are changed. Figure 4-17 gives the dependence of the suction pressure on the load, expressed in percent of the rated productivity, at different pressures of the working vapor based on [121]. If the pressure behind the stage is held constant, then, as can be seen from Fig. 4-17, with an increase in the pressure of the vapor, suction pressure decreases, and the productivity increases. This behavior is also reflected in the diagram (Fig. 4-12).

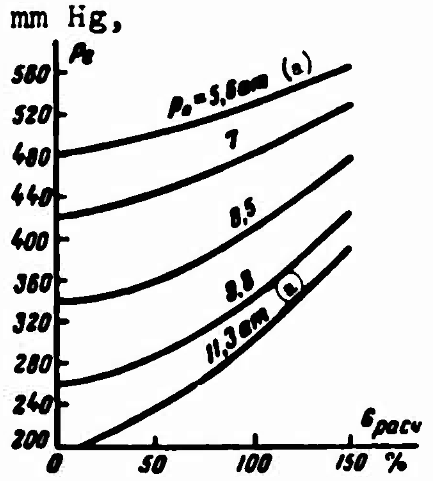


Fig. 4-17. Dependence of the suction pressure on the load (in percent of the rated productivity) at various pressures of the vapor. The calculated pressure of the vapor is 7 atm(tech.). KEY: (a) atm(tech.).

An increase in the pressure of the vapor leads, furthermore, to an increase in the maximum counterpressure and a corresponding compression ratio with a constant load.

Flow rate of the working vapor. A change in critical throat diameter of the nozzle, resulting on the one hand, in a change of vapor consumption, and on the other hand, in a change of the basic geometric parameter $f_3/f_{п.к}$, substantially affects the characteristics of the jet pump (Fig. 4-18). As can be seen from the figure, with an increase in the critical throat diameter of the nozzle, the maximum productivity of the ejector strongly increases, practically proportional to the increase in consumption of the working vapor. At the same time, in this case a certain increase in suction pressure during an identical load is observed.

(a)

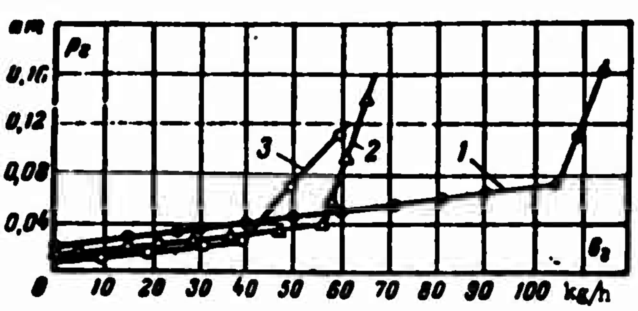


Fig. 4-18. Characteristics of first stage of a two-stage jet pump under operation at different critical throat diameters of the nozzle (d_3 = 50 mm, p_0 = 16 atm(tech.)) [45]. 1 – with a critical throat ciameter of the nozzle of $d_{п.к}$ = 7 mm; 2 – $d_{п.к}$ = 6 mm; 3 – $d_{п.к}$ = 4 mm. KEY: (a) atm(tech.).

Temperature of the pumped gas. With an increase in the temperature of the pumped gas the productivity of the jet pump decreases. The decrease in productivity according to [155] is approximately 5% for each 100°C, as compared to the value of productivity at a normal temperature (Fig. 4-19).

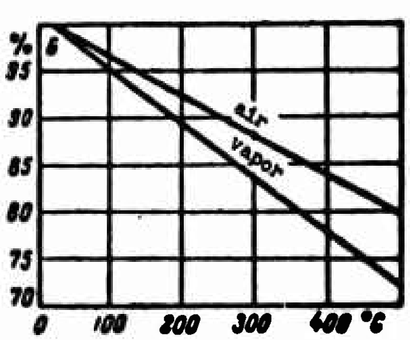


Fig. 4-19. Dependence of the productivity of a jet pump on the temperature of the ejected gas-air and vapor (in percent of the productivity at 20°C) [157, 158].

Molecular weight of the pumped gas. With an increase in the molecular weight of the pumped out gas, the productivity of the jet pump increases (Fig. 4-20). Such a dependence is caused by the viscous character of the flow of the pumped gas, and is analogous to the dependence on the molecular weight of the carrying capacity under viscous conditions (see Section 2-6).

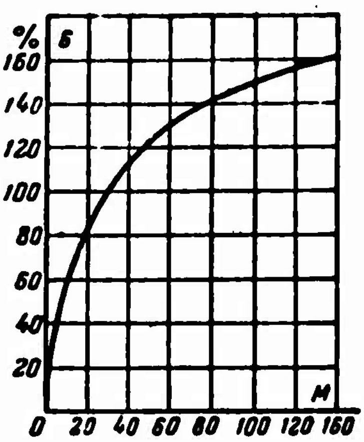


Fig. 4-20. Dependence of the productivity of the jet pump on the molecular weight of the ejected gas (in percent of the productivity based on air) [157, 159].

f) *Characteristics of multistage jet pumps.* As already indicated, when there is a necessity to carry out large compression ratios of pumped gas multistage jet pumps are used. In this case the selection of a compression ratio for a single stage is determined by the optimum relationship between the parameters of the stage and the vapor consumption, necessary for their realization, i.e., efficiency factor of the stage (efficiency). The efficiency factor η can be expressed as the ratio of the work of the compression of gas from pressure $p_г$ up to the pressure p_c, to the work accomplished by the vapor during an adiabatic expansion from pressure $p_{п0}$ to pressure $p_2 = p_г$. For simplicity by taking the process of compression as isothermal, we can record, taking into account the equation (3-6):

$$\eta = \frac{AG_г RT \ln \frac{p_c}{p_г}}{G_п \Delta I}, \qquad (4\text{-}49)$$

where ΔI — difference in heat content of the vapor upon entry and exit of the nozzle.

Using equation (4-43) for the maximum counterpressure and equation (4-6), we can record:

$$\eta = c\nu \ln (1+\nu_{пр})z, \qquad (4\text{-}50)$$

where $c = \frac{ART}{\Delta I}$ — constant value;

$z = \frac{K_н}{K_с}\frac{\Pi_{н.н}}{\Pi_{с.н}}\frac{w_{с.н}}{w_{н.н}}\frac{l_{н.н}}{l_с}\frac{\rho_н}{\rho_р}$ — function $\nu_{пр}$.

As can be seen, the efficiency of the jet pump is a composite function from the coefficient of ejection and has a maximum value at a certain optimum value $\nu_{опт}$, and consequently, (at a constant expansion ratio of vapor E), also at a specified compression ratio $\xi_{опт}$ (see Fig. 4-12). When $\xi > \xi_{опт}$ the efficiency of the jet pump decreases. To take ξ larger than 7-10 is hardly recommended.

During multistage compression most ideal is an equal distribution of compression ratios between the stages

$$\xi_{ст} = \sqrt[n]{\xi}, \qquad (4\text{-}51)$$

where $\xi_{ст}$ – compression ratio in one stage; ξ – full compression ratio, n – number of stages. However, in a concrete case with the distribution of the compression ratios between the stages it is necessary to consider the operation condition of the condensers. Therefore, in practice it is frequently necessary to refrain from an equal distribution of the compression ratios.

The design of a multistage jet pump amounts to a calculation of each stage for a selected compression ratio and prescribed productivity, and also amounts to a calculation of the intermediate condensers between stages. The design of the stages is made in accordance with the above-stated methods. In this case, every subsequent stage pumps not only air, compressed by the first stage, but also vapor, separating from the condensate in the condenser. The vapor content of vapor-air mixture discharging from the condenser can be determined from the equation

$$d = \frac{\frac{R_в}{R_п}\frac{p_п}{p - p_п}}{1 + \frac{R_в}{R_п}\frac{p_п}{p - p_п}} \text{ kg of vapor per 1 kg of the mixture,} \quad (4\text{-}52)$$

where $R_в$ and $R_п$ – gas constants of air and vapor; p – overall pressure of the mixture (pressure in the condenser); $p_п$ – elasticity of the saturated vapor at a temperature of the mixture discharging from the condenser.

The weight of the vapor-air mixture, pumped by the stage behind the condenser, accordingly, is equal to:

$$G = \frac{G_в}{1 - d}, \quad (4\text{-}53)$$

where $G_в$ – quantity of air, pumped by the stage.

For multistage vapor-water jet pumps, besides the quantity of air $G'_в$ removed from the installation, also the quantity of air $G''_в$ is considered, sucked through the looseness in the connections, and the quantity of air $G'''_в$, included with the cooling water in the condensers of mixing. Thus, for these pumps

$$G_в = G'_в + G''_в + G'''_в. \quad (4\text{-}54)$$

Vigand assumes $G''_в = 0{,}1–0{,}2$ kg/h per linear meter of linings in sectional vacuum connections and $G'''_в = 0{,}1–0{,}25$ kg per 1 m^3 of water.

The design of condensers is made based on usual methods, acceptable for these apparatuses [161-165].

In a multistage jet pump, the operation of separate stages is interconnected: the suction pressure of every subsequent stage is simultaneously the counterpressure for the preceding stage. Therefore, in the process of pumping out the stages operate at a variable counterpressure and variable suction pressure. Characteristics of every preceding stage depends on the operation of the subsequent one.

Figure 4-21 gives the characteristics of I and II stages of a two-stage jet pump during their combined work. The dot-dash curve is designated as the curve of maximum counterpressure of I stage. The point of intersection of this line with the characteristics of II stage determines the point Б overload of the first stage. To the left of A the suction pressure of II stage turns out to be all along the segments of the characteristics lower than the maximum counterpressure of I stage, and therefore, to the left of point Б the characteristics of I stage corresponds to the maximum rating. In this segment the characteristics of I stage do not depend on the operation of II stage. But then to the right point Б, the characteristics of II stage lies higher than the line of maximum counterpressure i.e., on this segment the suction pressure of II stage, and accordingly the counterpressure as well behind I stage turns out to be higher than the maximum counterpressure of I stage; the characteristics of I stage rises sharply upwards. On this segment $\nu < \nu_{np}$ and depends on operation of II stage.

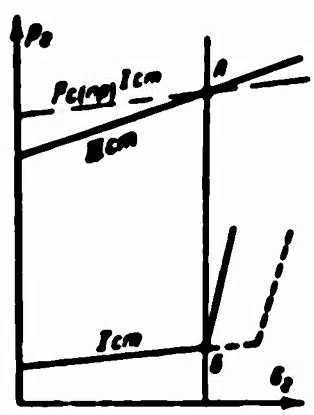


Fig. 4-21. Characteristics of I and II stages of a two-stage jet pump during their combined operation.

If I stage were to work independently of the second, then the point of overload would be located to the right (dotted line on the characteristics).

Interrelationship of the characteristics of stages also appears during a change in different conditioning factors. Thus, during a change of the conditions of cooling the condenser, with a temperature rise of the cooling water the suction pressure of II stage also increases. The characteristics of II stage is displaced upwards

(Fig. 4-22a). In this case the point of intersection of characteristics from the line of maximum counterpressure of I stage is shifted to the left, and the transfer conditions of I stage proceed at a smaller value of $\nu_{пр}$.

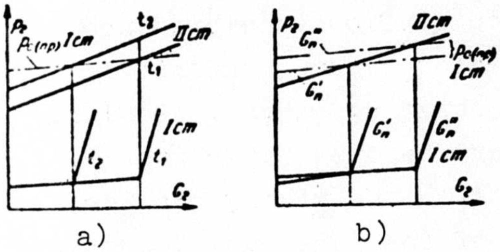


Fig. 4-22. Characteristics of a two-stage jet pump during a change of operating conditions. a) increase in the temperature of the cooling water; b) increase in the vapor consumption in the first stage.

With an increase in vapor consumption in I stage, the curve of maximum counterpressures is displaced upwards (Fig. 4-22b). The point of intersection of this curve with the characteristics of II stage is displaced to the right, and accordingly, the transfer conditions of I stage proceed at a larger value of $\nu_{пр}$.

4-4. Vapor-Oil Jet Pumps

a) *Working principle and design.* Special vacuum oils are used as working fluids in vapor-oil jet pumps. Oil vapor in these pumps forms in the boiler, designed for the pump. Behind the jet pump a condenser is installed, after which the condensate is recycled in the boiler by an overflow pipe. Thus, the continuous circulation of oil in the pump is ensured. Pumps usually have a high value for the largest outlet pressure (maximum counterpressure), equal to 2-3 mm Hg, and a peak output in the region of pressures of 10^{-1}-1 mm Hg.

In contrast to the above described diagram of a jet pump, vapor-oil pumps have only a mixing chamber and do not have a diffuser, in spite of its essential role in the operation of a jet pump. This is done for the sake of reducing the overall dimension of the pump.

Figure 4-23 gives a diagram of the design of an EN-50 vapor-oil jet pump with a conical mixing chamber. The pump layout is clear from the figure. The characteristic feature of construction is the incomplete cooling of mixing chamber: only for the inlets and outlets; the central part is heat-insulated. Cooling near the inlet hole is necessary for the condensation of the flow lines of vapor, directed to that side, the opposing direction of flow (the presence of these flow lines reduces operating speed of the pump, especially during operation on the lower limit of the operating range). Cooling near the outlet of mixing chamber is necessary because otherwise a vapor backwater is created upon discharge from the mixing chamber, leading to a lowering of the measured value of the peak outlet pressure by air. The vapor intake pipe in the lower part is heat-insulated, and in the upper part connected with a cooled pipe through an intermediate long pipe in order to reduce the thermal conduction from the hot vapor intake pipe to the cold pipe. The pocket forming in this case serves simultaneously for the collection of condensate flowing from it through the overflow pipe, into the boiler. Behind the mixing chamber a condenser cooled by water is installed; it consists of a cylinder with disks on the inside attached to a cooled rod.

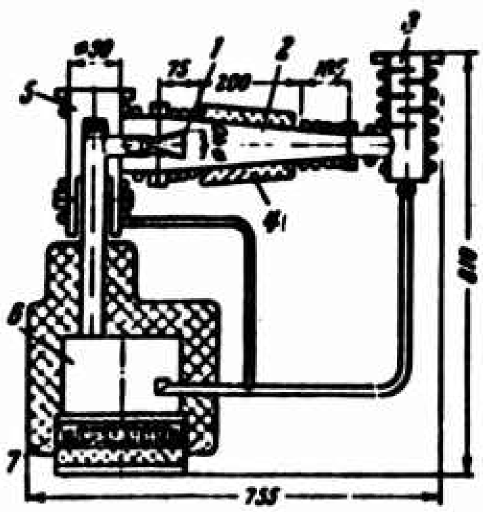


Fig. 4-23. Diagram of the construction of a EN-50 vapor-oil jet pump. 1 – nozzle; 2 – mixing chamber; 3 – condenser; 4 – thermal insulation; 5 – inlet pipe; 6 – boiler; 7 – electric heater.

The dependence of the response of the pump on the injection pressure in shown in Fig. 4-24. The pump consumes 1.5 kW of power.

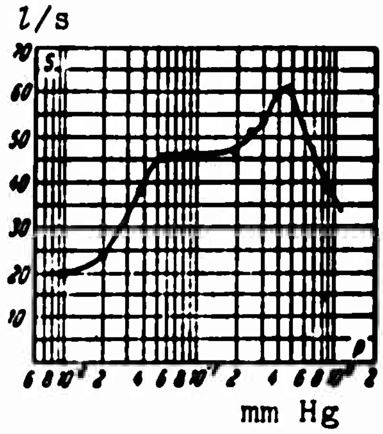


Fig. 4-24. Dependence of response of EN-50 pump on the injection pressure.

The peak outlet pressure is 2.0 mm Hg. The electric heater of the pump consists of a ceramic plate with concentric grooves, in which a nichrome coil is seated. The plate is set in a metallic housing and is heat-insulated. In external appearance and construction, the electric heater reminds one of an ordinary electric stove. Special vacuum oils are used as a working fluid in the pump: vaseline ("G" and VM-3) and a silicon organic liquid (PFMS-1).

Figure 4-25 gives a diagram of the two-stage vapor-oil pump, ODR-300, released by the Leybold firm in FRG. The mixing chambers of both stages of the pump are cooled by water. The boiler is made in the form of a pipe, is heated by a tubular heater inserted inside the boiler. The dependence of response of the pump on the injection pressure is shown in Fig. 4-26. L-50 oil (pentachlorodiphenyl) is used as a working fluid in the pump. The pump consumes 9 kW of power and has a peak outlet pressure of 3 mm Hg.

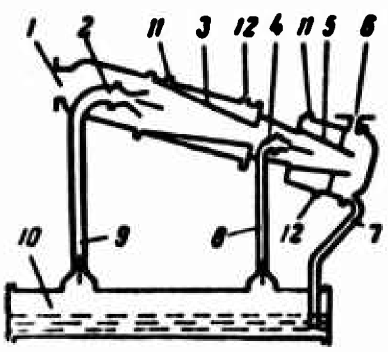


Fig. 4-25. Diagram of the construction of a two-stage vapor-oil ODR-300 pump. 1 - injection pipe; 2 - nozzle of the first stage; 3 - mixing chamber of the first stage; 4 - nozzle of second stage; 5 - mixing chamber of the second stage; 6 - outlet pipe; 7 - overflow pipe; 8, 9 - vapor-feed pipes; 10 - boiler; 11 - water intake; 12 - water outlet.

Along with their application as independent single and two-stage pumps, vapor-oil jet pumps are also used as outlet stages in booster

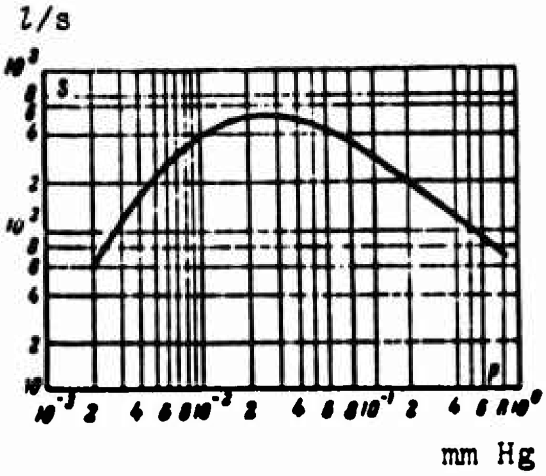


Fig. 4-26. Dependence of the response of ODR-300 pump on the injection pressure.

and high-vacuum multistage pumps for the purpose of obtaining a high value of the peak outlet pressure in these pumps (see Chapters V, VI).

Where the peak outlet pressure of vapor-oil jet pumps, $p_{c(\text{пр})}$, is less than atmospheric, auxiliary vacuum mechanical pumps, compressing the gas from pressure $p_{c(\text{пр})}$ up to atmospheric are used to ensure their operation.

b) *Characteristics*. Since vapor-oil jet pumps work in combination with auxiliary mechanical pumps, then the characteristics of the jet pump depends on the operation of the mechanical pump; similarly, the characteristics of I stage of a multistage jet pump depend on the operation of II stage (see Fig. 4-21). Thus, with an increase in the output of the auxiliary mechanical pump, the maximum output of jet pump will be increased, and accordingly, the maximum of characteristic $S = f(p_{\text{вп}})$ will be shifted in a region of higher pressures (Fig. 4-27) [120].

The value of the peak outlet pressure of the jet pump depends on the value of the injection pressure, since the character of dependence can be different for various pumps and can be determined mainly by the design of mixing chamber.

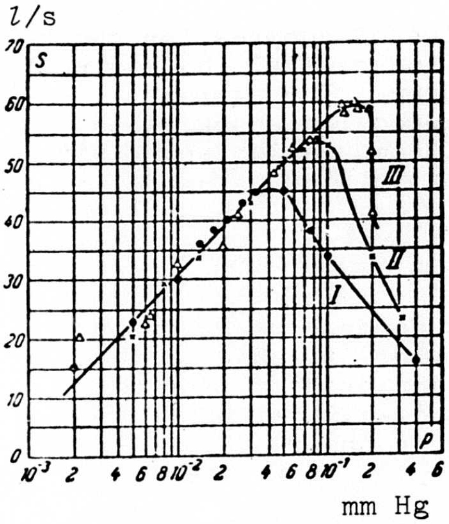


Fig. 4-27. Dependence of response of a vapor-oil jet pump on the injection pressure at various outputs of the auxiliary pumps $G_I<G_{II}<G_{III}$. I — during the output of auxiliary pump G_I; II — during G_{II}; III — during G_{III}.

Let us consider two cases: first — when the stream, issuing from the nozzle, enters the cylindrical mixing chamber, and second — when the stream enters the conical mixing chamber (Fig. 4-28). The diffuser in both cases is absent. Let us assume that in both cases the injection pressure at first corresponds to the maximum vacuum. If one were to increase the outlet pressure, then, as already noted above, stream compression shock will appear in the vapor stream, shifting with the increase of the outlet pressure towards the nozzle. If the injection pressure starts to increase at the time of the extreme position of compression shock preceding the breakaway of the stream from the walls, then in first case (a) due to the narrowing of the stream (stream is folded by the injection pressure) its breakaway from the walls will occur and overflow of the gas from the region of preliminary rarefaction towards the side of the inlet will start; in second case (b) due to the narrowing of the stream, the front of the compression shock will be shifted in a narrower part of the housing and breakaway of the stream from the walls will not occur. In other words the higher the injection pressure, the less the peak outlet pressure should be — in first case, and those higher should apply — in the second case.

Figure 4-29 gives the dependence of the output and the peak outlet pressure for two experimental vapor-oil jet pumps with a conical mixing chamber. As can be seen from Fig. 4-29, the peak

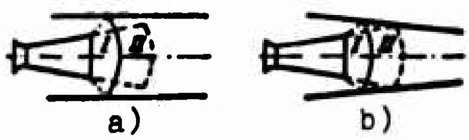


Fig. 4-28. Formation of compression shock in cylindrical (a) and conical (b) mixing chambers of a jet pump. I – extreme position of the compression shock; II – extreme position of compression shock with an increase in the injection pressure.

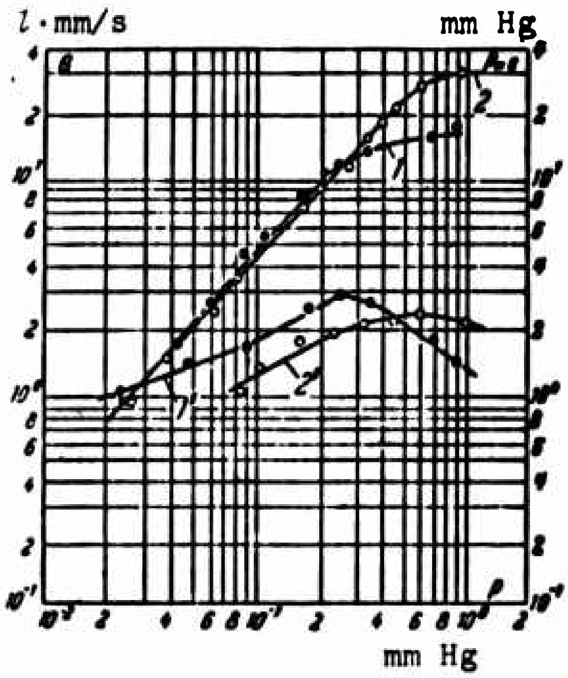


Fig. 4-29. The dependence of output and the peak outlet pressure on the value of injection pressure for two experimental vapor-oil jet pumps with a conical mixing chamber. 1, 2 – output; 1', 2' – the peak outlet pressure.

outlet pressure of jet pump increases with an increase in the injection pressure up to a certain maximum value, corresponding to the maximum output, but then it decreases. Presence of maxima on curves $p_{c(\text{пр})} = f(p_{\text{вп}})$ and $G = f(p_{\text{вп}})$ can be explained by the fact that in the contracting mixing chamber the increase in the peak outlet pressure with an increase in the inlet pressure continues as long as the front of the compression shock in its extreme position does not coincide with the section of the outlet hole of mixing chamber; at this instant the maximum value of the peak outlet pressure is attained. With a further increase in injection pressure the stream narrows so much that it ceases to completely fill the outlet, and the gas from the side of preliminary rarefaction starts to overflow into the region of the injection pipe; the peak outlet pressure of the pump in this case decreases, and the output no longer increases. (Actually, the pumping action of the jet pump in this case decreases and, finally, its role amounts to a certain system through which the high-pressure vacuum pump pumps the gas.)

Vapor-oil jet pumps operate at a comparatively low pressure of vapor in the boiler, from several to tens of millimeters of mercury. This is caused, first, by the small specific gravity of the oils, and the necessity in connection with this, to devise large hydraulic oil seals in the overflow pipes for great pressures and, secondly, by the decomposition of oils at high temperatures in the boiler.

Due to the small pressure in the boiler, and accordingly, the small value of expansion of vapor $E \approx 20$-100 in the nozzle with a compression ratio, $\xi = 5$-10, usually maintained in the oil jet pumps, the coefficient of ejection of jet pumps has a very small value, $\nu \approx 10^{-2}$-10^{-3} (see character of the change in ν on E and ξ in Fig. 4-12).

A theoretical calculation of vapor-oil jet pumps at present turns out to be impossible due to the lack of the necessary thermodynamic characteristics of the vacuum oils. Unknown, in particular, is the value of the adiabatic index, $\kappa_{п}$, for oil vapors; the phase diagram of the oil vapor is lacking.

The design of vapor-oil jet pumps is done on the basis of experimental data with subsequent experimental adjustments. In this case the above-indicated principles of similarity of jet pumps are used, based on the basic geometric parameters $f_3/f_{п.к}$ and $f_{п1}/f_{п.к}$ under identical operating conditions.

c) *Working fluids.* The basic requirements, demanded of working fluids in vapor-oil jet pumps, are: high thermal and thermal-oxidation stability, high vapor tension at the operating temperature in the boiler of the pump, small value of heat of vaporization. In contrast to oils for other vacuum vapor-jet pumps, the oils for the jet pumps should not necessarily have a low vapor pressure at room temperature, since the critical vacuum of such pumps does not have practical value and can be sufficiently large.

In principle, one should use the lightest oils with a high vapor pressure for jet pumps. However, at present the same working fluids are used for them as for booster vapor-oil pumps: "G," VM-3 and PFMS-1 oils (see Chapter V). The vapor tension of these oils is sufficiently low: 10^{-4}-10^{-6} mm Hg (at 20°C); therefore, their application in jet pumps in no way can be considered effective enough.

d) *Exploitation of pumps.* Vapor-oil jet pumps are used for pumping out installations, operating in the range of pressures of 10^{-3}-1 mm Hg, with the maximum elimination of gases at pressures of 10^{-1}-1 mm Hg. Most valuable is their use in installations, where the pumping of vapors of organic compounds, soluble in oil, with a high vapor pressure (distilling or impregnating installations, and others) is required. In this case the application of oil rotary pumps is inconvenient, since the pumped products, being dissolved in oil, increase over the course of time by the produced pump pressure. In jet pumps these products will be easily driven off due to the high vapor temperature in the boiler, but the entrapment of products in the mechanical pump mounted behind the jet pump will not even result in the impairment of the suction pressure due to the high value of the peak outlet pressure of the jet pump. Special advantages are gained with the pumping out of vapors of liquids using vapor-oil jet pumps; these can apply for pumping, for example, esters, fatty acids, vaseline oils, and others.

Due to the very simple design the exploitation of vapor-oil jet pumps does not create difficulties. When using stable working fluids (for example, PFMS-1) pumps can operate for months without bad effects on their characteristics. In these cases the service life of the pump is limited by service life of the heater and oil leakage. Losses of oil in the pump are caused mainly by the removal of vapors through the outlet pipe. In this case the biggest losses of oil take place during operation in the transfer section of the characteristics. Therefore, when selecting a jet pump it is necessary to commensurate its output with the quantity of pumped gas in such a way that pump would work a minimum time in the transfer section of the characteristics (see Chapter IX).

To ensure the normal operation of the pump it is necessary only to watch for a decrease in oil in the boiler and to periodically add to it, and also to make a prophylactic cleaning of the pump at set periods depending upon the conditions of exploitation. A prophylaxis of the pump consists of disassembling the pump and washing it in some solvent. Usually for this purpose purified nonethylated gasoline, purified kerosene, acetone, or dichlorethane are used. It is necessary to consider that all these products are toxic, and to work with them should be done under hoods or in specially equipped locations with effective ventilation. The operation should include the use of rubber gloves, and in case of open washing, respirator masks or gasmasks.

4-5. Mercury-Vapor Jet Pumps

a) *Design.* The application of mercury as a working fluid in mercury-vapor jet pumps is governed by a number of its features: a) high vapor tension during the operating temperature in the boiler, permitting one to produce a large maximum counterpressure with small expenditures of power upon vaporization; b) the stability of mercury: being a simple substance, it does not decompose into any components; c) small tendency to dissolve gases; d) high specific gravity, allowing an equalizing of pressure in the boiler by the small mercury seal in the overflow pipes. These features of mercury would govern the use of mercury-vapor jet pumps in the following instances: a) for pumping rare and valuable gases from a container with a low pressure, into a container with a high pressure without the risk of contaminating the gases with organic products or without their partial loss due to dissolution in oil (in the case of using oil pumps); b) for pumping out radioactive gases without contamination by their organic products (in the case of oil pumps; furthermore, radioactive gases destroy oil in a pump which makes the application of an oil pump unsuitable for this purpose); c) for pumping and compressing gas mixtures during a gas analysis without the contamination and loss of gases; d) as auxiliary pumps, ejecting gas in atmosphere, in pumping systems without oil.

Mercury-vapor jet pumps are usually designed in multistage with the peak outlet pressure in tens of millimeters of mercury to 1 atm(tech.).

Figure 4-30 gives a diagram of a device of a three-stage mercury-vapor jet pump [119]. The first stage of the pump has cylindrical mixing chamber without a diffuser, the second and third stages - usually, with effusers. In all stages the mixing chamber and diffusers 4 are placed inside unique surface condensers comprised of two coaxially arranged pipes, 5 and 6, around which a coil of wire is wound. Vapor, coming out of diffuser 4, passes through the holes at the bottom of the inner pipe 5 and, rising along the coiled channel between the pipes upwards, is condensed on water cooled walls of outer pipe 6. The condensate flows downwards and through the overflow pipe 7, returns to boiler 1. The overflow of the condensate from the first and the second stages is carried out by a lock between the stages. The filling of the boiler with mercury is done through tube 9, which has a fin for cooling, and a moveable joint for rotating the vertical section of the tube by 180°. Through tube 9 when its rotation is downwards by 180° the overflow of mercury from the boiler can be made. Boiler and the vapor-feed pipe have asbestos thermal insulation. The pump has an output based on air of 30 $l\cdot$mm/s at 1 mm Hg, the peak outlet pressure of 100 mm Hg, maximum vacuum using air of 10^{-3} mm Hg; it consumes 6.4 kW of power.

Figure 4-31 gives the appearance of a four-stage EN-100R mercury-vapor jet pump. The pump has a fast pumping action; 100 l/s at a pressure of 10^{-1} mm Hg and a maximum vacuum of $1.5\cdot10^{-6}$ mm Hg; a peak outlet pressure of 50 mm Hg; it consumes 4 kW of power. By being connected in series with a water-jet pump, it ensures initial vacuum pumping without oil of large volumes. Based on its output it is equivalent ot a VN-6 mechanical pump. Its overall dimensions: height 1,370 mm, area in plan of 560 × 570 mm^2.

Diagram of experimental design construction of a mercury-vapor jet pump, operating with the exhaust in the atmosphere, is shown in Fig. 4-32. A three-stage pump has been made out of stainless steel.

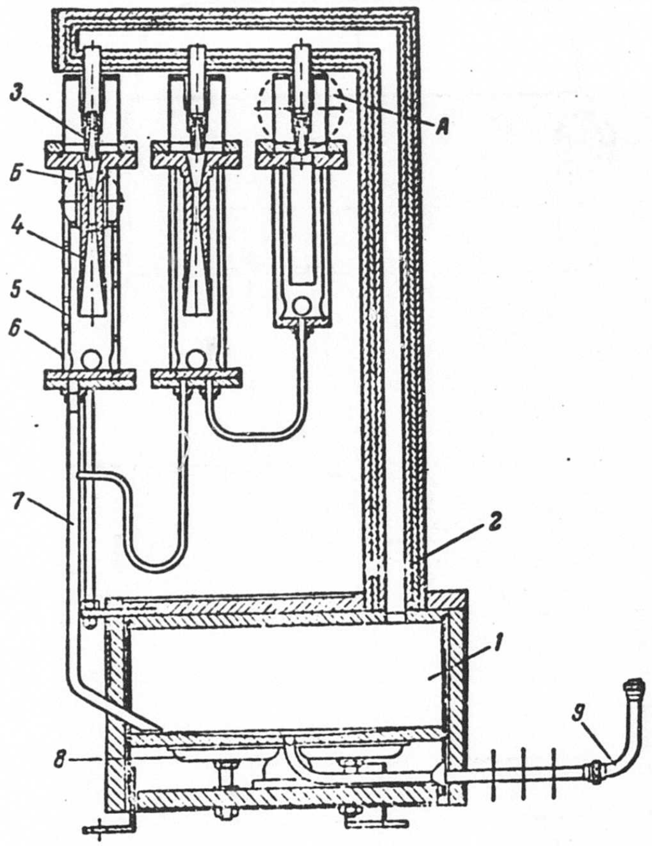


Fig. 4-30. Diagram of the design of a three-stage mercury-vapor jet pump. 1 – boiler; 2 – vapor-feed pipe; 3 – nozzle; 4 – mixing chamber with a diffuser; 5 – inner pipe of the condenser; 6 – outer pipe of the condenser; 7 – overflow pipe; 8 – heater; 9 – tube for filling the pump with mercury and its overflow; *A* – position of the injection pipe; *B* – position of the outlet pipe.

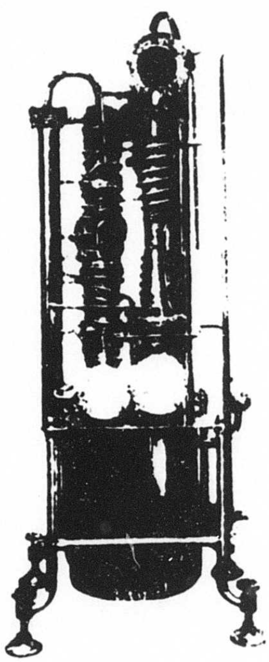


Fig. 4-31. View of EN-100R mercury-vapor jet pump.

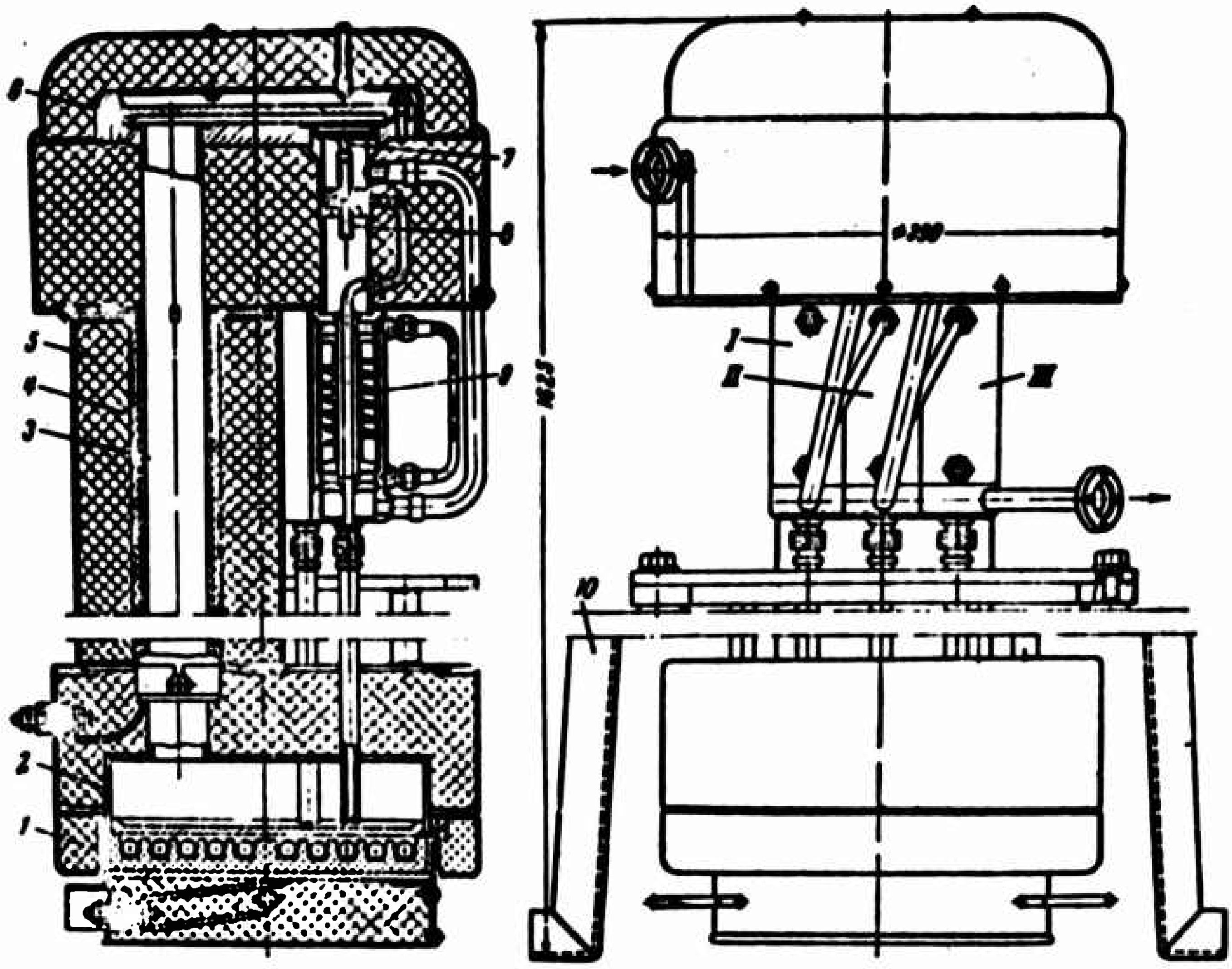


Fig. 4-32. Diagram of the design of a three-stage mercury-vapor jet pump, working with exhaust in the atmosphere. 1 – electric heater; 2 – boiler; 3 – vapor-feed pipe; 4 – electroheating; 5 – thermal insulation (asbestos); 6 – vapor header; 7 – nozzle; 8 – diffuser; 9 – condenser; 10 – frame; I – first stage; II – second stage; III – third stage.

In first two stages supersonic nozzles are expanded, in last stage a contracting sound nozzle is installed. The pump consumes 4.5 kW of power. The characteristics of the pump are shown in Fig. 4-33. The residual pressure, created by the pump, amounts to 20 mm Hg. By increasing the number of stages in the pump, one can produce a pressure lower than 10^{-4}-10^{-5} mm Hg, whereby the jet pump principle transcends into pure diffusion.

For a pump, operating with the exhaust of the gas into the atmosphere the shielding of the pump from the ejection of mercury vapors through exhaust duct is a serious problem. As the experiment

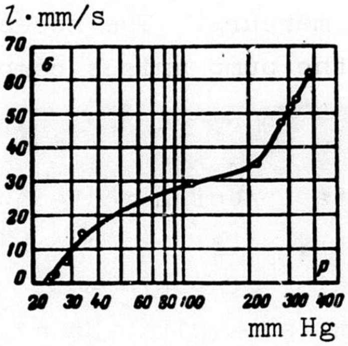


Fig. 4-33. Dependence of the output of a three-stage mercury-vapor jet pump on the injected pressure.

shows, for this purpose an installation behind the last-stage of the condenser of a pump, made of carbon filters from a gasmask, or the installation of a trap with water in which a vapor-gas mixture bubbles through the layer of water coming out of pump can be used. Independently of the quality of the filters, installed behind mercury-vapor jet pump, the exhaust of gas from the pump cannot be carried out on location; an exhaust line for the pump behind the filter, connected to an exhaust pipeline removed from the location, is necessary.

In mercury-vapor jet pumps ordinary commercial mercury is used. Special requirements for purifying the mercury are not imposed. In domestic pumps mercury brands P-1, P-2 and P-3 are used.

The high chemical activity of mercury governs the selection of structural materials for the pumps. Usually, mercury-vapor jet pumps are made of stainless steel which interacts weakly with mercury. For the sealing of linings in jointed couplings exposed to mercury vapor at high temperatures, soft sheet metal and nickel can be used. The design and dimensions of such couplings should be selected based on the available normals for jointed heated couplings [287].

b) *Exploitation.* Exploitation of mercury-vapor jet pumps is very simple. The pumps can work for a prolonged time without impairment of their characteristics. Thus, just as in the case of vapor-oil jet pumps, the period of continuous work of mercury-vapor jet pump is limited by the service life of the heater and by the reduction of mercury from the pump. Since mercury does not undergo changes during the operation of the pump, then it is necessary only to replenish periodically the decrease of mercury. The necessity for prophylactic dismantling and cleaning of the pump arises comparatively rarely.

The important features in the exploitation of mercury pumps are connected with the toxicity of the mercury vapors. This circumstance demands special requirements for the location of equipment, as to where the pumps are installed, and where to work with the mercury (see [284]).

4-6. Steam-Water Jet Pumps

a) *Designs.* Steam-water pumps are the most widely used compared to other types of vapor-jet pumps. This is due to the simplicity of steam as a working medium (uniform in composition, does not decompose, is not oxidized, and so forth), its accessibility and potential for practically unlimited consumption with respect to the contemporary level of boiler design. The last circumstance along with the potential for consumption of the necessary quantity of cooling water makes it possible to design steam-water jet pumps for any required output. Thus, at present there are pumps with an operating speed of hundreds of thousands of liters per second [118].

A typical diagram of a two-stage steam-water jet pump with an intermediate condenser is shown in Fig. 4-3. In contrast to vapor-oil and mercury-vapor jet pumps, the steam in steam-water pumps is not prepared in its own boiler, but is fed through nozzles from the main steam pipelines from the heat and electric power plant (TETs) or boiler under raised pressure, usually on the order of 5-10 atm(tech.).

According to the shown diagram in Fig. 4-3, steam-gas mixture from first stage enters the mixing condenser where it flows downward along plates to the water. As a result of agitation and heat exchange between them, the vapor is condensed, and the gas, saturated with the vapors of cooling water, is pumped by the following stage. The condensate from the condenser leaves by gravity flow through the overflow barometric pipes into the cistern. A hydraulic seal is set in overflow pipes thereby equalizing difference in pressures between atmospheric pressure and the pressure in the condenser. The height of the seal is nearly 10 m. Accordingly, the condenser is located at a height of nearly 11 m from the level of water in the cistern. Condensers with a barometric discharge of the condensate go by the name of barometric condensers.

Along with mixing condensers in steam-water jet pumps, surface condensers are also used, usually the tubular type. Here, in contrast to mixing condensers heat exchange occurs not as a result of the direct contact of the steam-gas mixture with the cooling water, but through the walls of the tubes, inside of which the water moves, but on the outside - the steam-gas mixture.

Mixing condensers are widely used thanks to their simplicity, lower water consumption (Fig. 4-34), feasability of manufacture from any materials, lower market value and operating cost. It is exceedingly rare that they require cleaning. According to [117] a four-stage pump with a mixing condenser having an output of 14 kg/h costs 1.5 times less than the same pump with a surface condenser. Mixing condensers can operate with contaminated water, and without fear of particles from the machinery; these are washed away by the flow of water in the overflow pipes, and even less fear of corrosion, since the corroding medium entering the condensers is diluted by the cooling water.

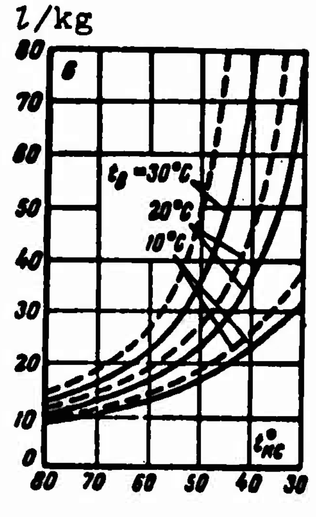


Fig. 4-34. Consumption of water in a condenser depending upon the temperature of the incoming water and the temperature of the condensate (solid curves - for a mixing condenser; dotted curves - for a surface condenser).

At the same time surface condensers possess a number of advantages, which in certain cases make their use more preferable than the use of a mixing condenser. Thus, the cooling water in them is not mixed with the condensate, which permits the return of clean condensate in the system. Less output is required in the drawing off of condensate by a condensate pump than by a mixing condenser. If corroding, poisonous or radioactive components are contained in the condensate, then far fewer problems arise, associated with trapping these components in using a surface condenser than in using a mixing condenser.

The question as to the application of some type of condenser can be solved depending upon the actual conditions of use of the vapor jet pump.

Figure 4-3 shows a typical diagram of the cistern or, as it is sometimes called, the barometric cistern. This cistern is of a certain capacity, and is made of concrete or metal. Its volume is calculated in such a way so as to ensure the filling of the barometric pipes with water with an installation of a trap. Usually, the capacity of the cistern is taken as not less than one and a half the capacity of all the pipes. The well is divided into two parts by a high threshold, through which the water overflows from the main chamber. Such an arrangement is convenient, since it converts the main trap chamber into a kind of a settling tank for different contaminations carried by the water from the pump. This is important, for example, in the case of the pumping out of dust-laden media with pumps.

Along with barometric condensers so-called low-level condensers are also used, from which the condensate leaves not through the barometric overflow pipes, but is drawn off by condensate pumps. In this case the positioning of the condensers at a great height is not required; thus, they acquire the name, low-level. Figure 4-35 gives a diagram of a four-stage vapor jet pump with a condensate pump. The condensate is removed from the condenser through overflow pipes in a closed steam trap, where it is drawn off by a pump. The height of the overflow pipes in this case can be set at nearly 1.5 m which is sufficient to equalize the difference in pressures between the individual condensers. In order to separate the high pressure in the last condenser from the low pressure in preceding condensers, the steam trap has a partition with a float throttle valve.

It is necessary to consider that in the case of sudden stopping of the condensate pump, water will be sucked into the condenser

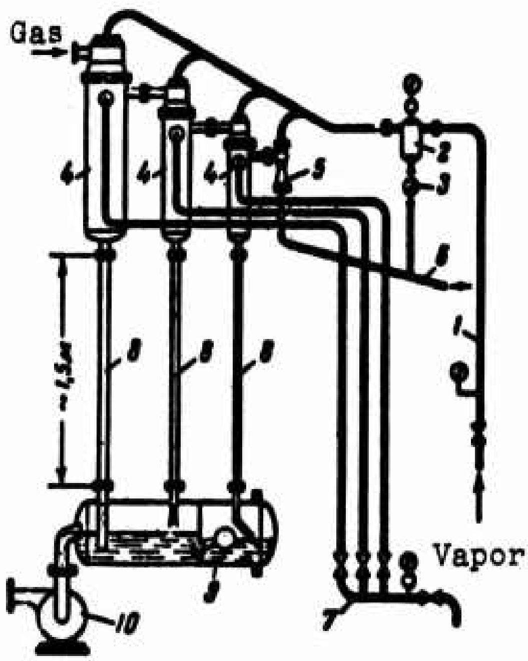


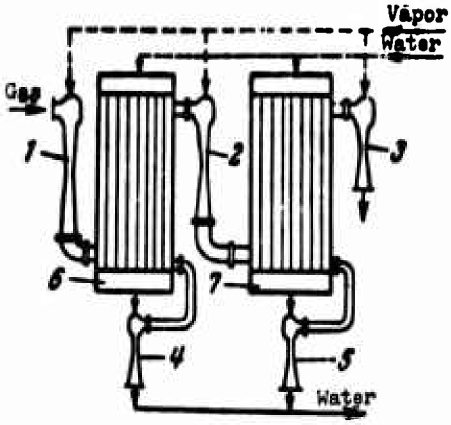
Fig. 4-35. Diagram of a four-stage steam-water jet pump with a condensate pump. 1 - steam pipeline; 2 - water separator; 3 - steam trap; 4 - condensers of Soviet design inside the jet pumps; 5 - jet pump; 6 - exhaust pipe; 7 - water-feed pipe; 8 - overflow pipe; 9 - condensate cistern; 10 - condensate pump.

through the overflow pipes and can enter the pumped volume. To avoid this, it is necessary to provide for an automatic cutoff of vapor and water in the case of the stopping of the condensate pump, and also to provide a cover for the valve on the suction pipeline (if there is one).

In addition, the diagram for the removal of condensate by gravity flow through the barometric tubes is more simple and reliable. Therefore, condensate pumps are used only in those cases, when the possibility of exploitation is limited by the overall dimensions of the pump.

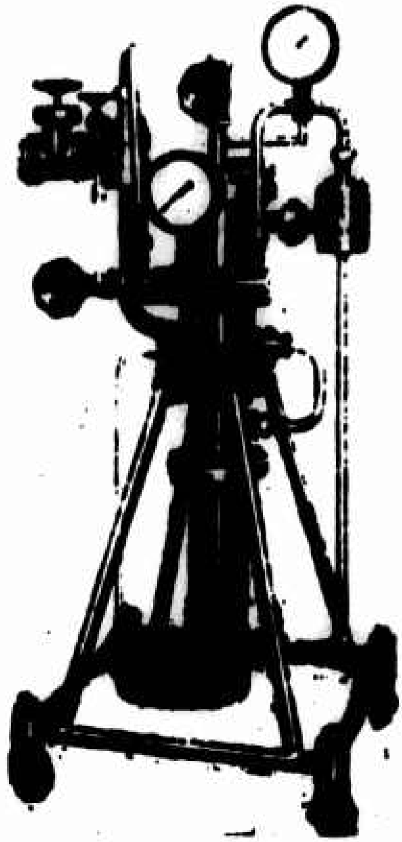
In pumps, which are short-term under actual operating conditions the tapping off of condensate from the condensers can be done through short enclosed pipes - collectors, which are under the same pressure as the corresponding condensers. The volume of the collector is selected equal to approximately one and a half the volume of the condensate emptying into it after a work cycle of the pump.

A circuit with intermediate surface condensers and water-jet pumps for drawing off the condensate from them (Fig. 4-36a) turns out to be very convenient and economical for small pumps. In this



a)

Fig. 4-36. Diagram of a steam-water jet pump with the removal of condensate from condensers by water-jet pumps (a); view of NEV-0.2×20 (b). 1, 2, 3 - steam stages; 4, 5 - water-jet pumps; 6, 7 - surface-condensers.



b)

case water-jet stages can be fed water coming out of the condensers which makes it possible to avoid an additional consumption of water. Pumps, operating according to such a circuit can have very small dimensions.

The last stage of the pump usually ejects a steam-gas mixture directly into the atmosphere. But sometimes, this happens to be inconvenient or is economically inexpedient. In these cases the condenser is installed behind the last stage whereby the vapors ejected from the stage are condensed, or the vapors in the cistern are collected under a layer of water, as shown in the diagram in Fig. 4-3.

Figure 4-36a gives a view of a small-size two-stage NEV-0.2×20 pump, operating according to the diagram analogous to Fig. 4-36a.

The steam-gas mixture from the second stage of the pump enters the mixing condenser, bathed with the water ejected from the water-jet stage. Pump has an output of 0.5 kg/h at a pressure of 20 mm Hg, the critical vacuum is 10 mm Hg. Steam consumption is 12 kg/h at a pressure of 2.6 atm(tech.), water consumption is 0.35 m^3/h at a pressure of 1.0 atm(gage). The overall dimensions of the pump: height of 1025 mm, the area in plan, 500 × 445 mm^2.

Usually, the steam-water jet pumps are calculated for operation based on dry saturated vapor. If the vapor turns out to be slightly overheated (Δt = 20-30°C) during the operation of the pump, then this is hardly reflected in the operation of the pump. If however, the vapor turns out to be humid, then the characteristics of the pump can be perceptible worsened. Therefore, when operating on humid vapor a moisture separator (Fig. 4-35) is recommended to be installed on the vapor-supply line. The drying of the vapor also occurs when choking it from an elevated pressure (see Section 3-4).

In vapor jet pumps with intermediate condensers the pressure, produced by the first stage, substantially depends on the temperature of the water in the condenser installed behind it. This is caused by the fact that the pressure behind the stage cannot be produced lower than the pressure of the saturated vapor of the cooling water in the condenser. At a water temperature of 25-30°C this pressure is nearly 30 mm Hg, and consequently, at a compression ratio of 4-10, an inlet pressure in the stage lower than 3-8 mm Hg cannot be produced. Such a pressure is usually produced in a three-four-stage pump with two-three intermediate condensers. To produce a lower pressure, multistage ejector pumps are designed in which the condenser and pressure behind it are lacking after the first stage; consequently, the pressure is governed only by the output of the following stage. A diagram of a five-stage NEV-100×1 pump with the first stage without a condenser, is shown in Fig. 4-37. Such a pump is able to produce a pressure of 0.5-1 mm Hg. By installing two, three and more stages without condensation, it is possible to devise pumps producing a pressure of 10^{-1}-10^{-2} mm Hg and lower. The number of stages essentially depends on the temperature of water, entering the condenser. At a water temperature of 20-25°C, four-stage

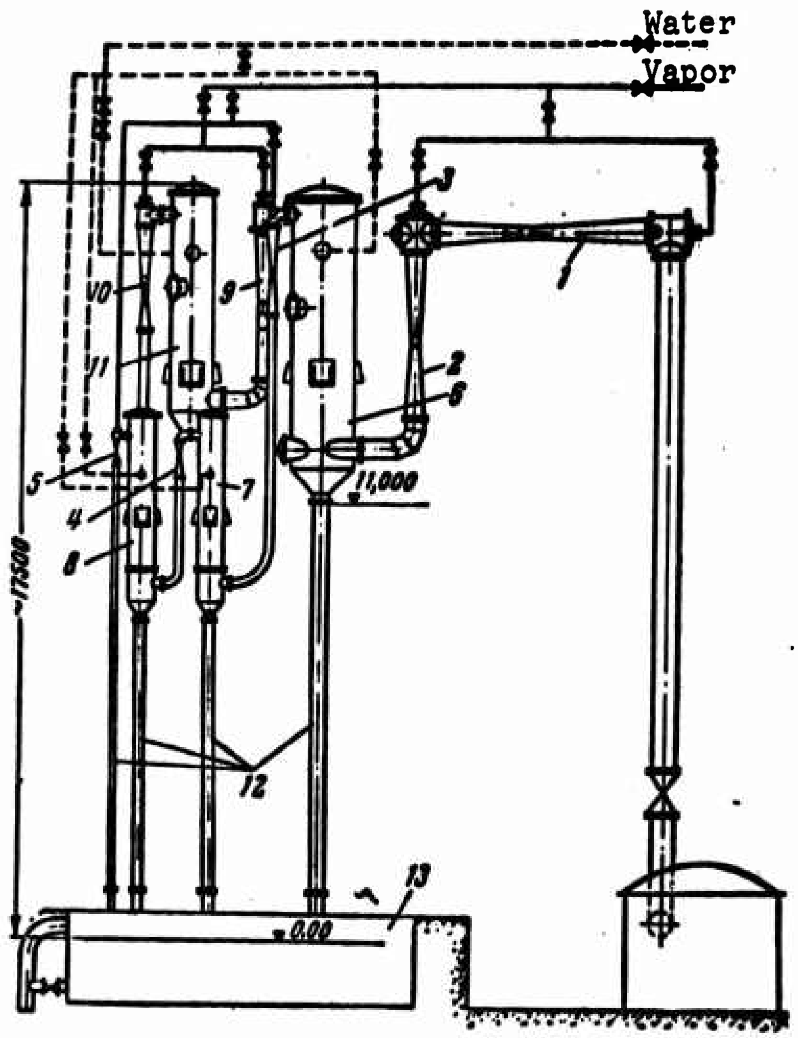


Fig. 4-37. Diagram of a five-stage NEV-100×1 steam-water jet pump. 1, 2, 3, 4, 5 - stages of the basic pump; 6, 7, 8 - condensers; 9, 10 - stages of starting pump; 11 - condenser of the starting pump; 12 - barometric overflow pipes; 13 - barometric cistern.

pumps usually have one stage without condensation and two with intermediate condensers, a five-stage pump - the first two stages without condensation, six-stage pumps - the first three stages without condensation.

The value of residual pressure depends upon the number of stages in a pump [117]:

1st stage -	50	mm Hg
2nd stage -	5	mm Hg
3rd stage -	2	mm Hg
4th stage -	0.2	mm Hg
5th stage -	0.03	mm Hg
6th stage -	0.005	mm Hg
7th stage -	0.0005	mm Hg

The possibility of producing such small pressures with help of a steam-water pump at first glance seems surprising. Moreover, as long as these pressures were not produced practically enough widespread opinion existed that with the help of a steam-water pump it was generally impossible to produce a pressure lower than the elasticity of steam at a temperature of the walls of the pumped system, i.e., lower than 15-20 mm Hg. This opinion was based on the fact that in the system the pressure of the steam is supposedly set at equilibrium. Meanwhile, in reality, this never happened, since the steam-water jet pump pumps steam very well. Therefore, the residual pressure, produced by the steam-water pump, can determine, in practice, only the number of stages in it.

In cases of pumps operating in stages at low pressures, operating at pressures lower than 4 mm Hg, an icing over of the flow-through part can occur which results in a substantial impairment of the characteristics and affects the critical vacuum of the pump. To prevent icing these stages are warmed up. Usually, steam is used for warming the heat derived mainly from the receiving chamber and effuser, sometimes the nozzle. For this purpose steam jackets are welded to these elements.

It is necessary to note that the application of stages in such pumps without intermediate condensers substantially increases the steam consumption per unit of pumped gas as compared to an ordinary diagram using condensers. Nonetheless, in those cases where producing low pressures are required, but where the operational cost is less important than a reduction in the overall dimensions of the pump, the pumps generally in use are sometimes without intermediate condensers.

Along with multistage vapor jet pumps, operating with an exhaust into the atmosphere, combinations of vapor jet pumps with other pumps are also used. In this case the vapor jet pump compresses the gas to a pressure of 50-100 mm Hg, but further compression of the gas into the atmosphere is done by an auxiliary pump. Thus, with vapor jet pumps of large output water-circulating mechanical pumps are used as auxiliary pumps. Schemes with water-circulating pumps are expedient at small vapor pressures (on the order of 1 atm(tech.)), since in this case, a large quantity of vapor would be needed (see Fig. 4-12) for the compression of the gas into the atmosphere in the vapor jet stage. When using water-circulating pumps, the advantages of the clean vapor jet pumping effect (reliability) are lost.

For pumps of low output it is expedient to use a water-jet pump as an auxiliary pump, which can not only fulfill the function of a pump with respect to the pumped gas but also serve simultaneously as a jet condenser for vapor ejected from the last stage of steam-jet pump. By combining the steam-jet stages without intermediate condensers with the water-jet pump, a very compact small-size pump can be devised. Figure 4-38 gives a design of a four-stage NEV-3 vapor jet pump without intermediate condensers and with a final stage - water-jet. The pump has an output of 1 kg/h at a pressure of 0.5 mm Hg during a flow rate of 140 kg/h of vapor at a pressure of 4 atm(tech.) and 14 m^3/h of water at a pressure of 4 atm(tech.). The area in plan is 1.05 × 1.15 m^2. The height is 2.5 m. The characteristics of the pump is shown in Fig. 4-39.

It is necessary to note that if the possibility of materializing pumps of very high output is determined mainly by the feasibility of the provision of vapor and water, then the minimum output of pumps would be regulated by the feasibility of the realization of the minimum dimensions of the nozzles. Thus, the output of pumps of nearly 0.3 kg/h, corresponding to an operating speed of 50 l/s

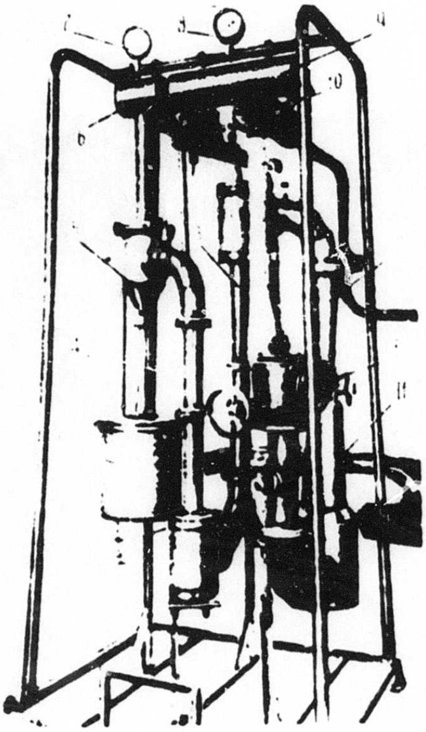


Fig. 4-38. Small-size NEV-3 steam-water jet pump. 1 - injection pipe; 2 - first stage; 3 - second stage; 4 - third stage; 5 - water-jet stage; 6 - water collector; 7, 8 - manometers; 9 - steam header; 10 - steam valves; 11 - heating of the first stage; 12 - overflow tank.

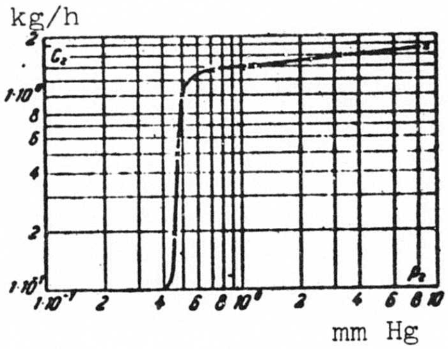


Fig. 4-39. Characteristics of small-size NEV-3 steam-water jet pump.

at 1 mm Hg, 500 l/s at 0.1 mm Hg, 5000 l/s at 0.01 mm Hg, etc., is apparently the minimum practicable at pressures of a working vapor of 5-6 atm(tech.), since critical diameters of the nozzles in this case are already of the dimensions of the order of 1 mm. One should as far as possible avoid using such small diameters of nozzles due to the significant boundary layer effect on the steam consumption through the nozzle. With this in mind the pressure of the working vapor is reduced in first stages of multistage pumps of small output to a value with which the critical throat diameter will be not less than 2-3 mm.

If problems arise in small pumps, associated with the small dimensions of the nozzles, then in large pumps problems arise associat with large dimensions of nozzles. Thus, in the first stages of highly productive pumps, operating at high flow rates and with large expansions of the vapor, the nozzles sometimes turn out to be so long that their manufacture becomes difficult. In these cases it is expedient instead of one nozzle to install a set of nozzles in the jet pump with a total area of critical section, calculated according to the required steam consumption [58]. The efficiency of a jet pump with a multinozzle head turns out to be higher than a jet pump with a single nozzle, due to the well-developed surface of the jets.

In pumps of very high output, the first stages usually are arranged in the form of a group of operating stages in parallel. This enables one to avoid an error of large dimensions in the stages. Thus, Wiegand [37] does not recommend devising a stage with the throat diameter of the diffuser larger than 250 mm. At the same time the presence of parallel stages creates possibility to control the pump capacity in process of work by means of a cutoff of one or several of the stages.

In certain designs of pumps stages of the jet pump are built inside the condensers (Fig. 4-40) which complicates their construction, but then it considerably reduces the overall dimensions of the pump [78].

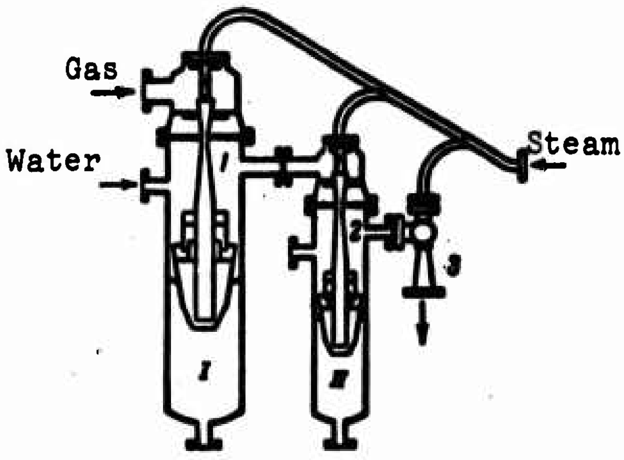


Fig. 4-40. Three-stage steam-water jet pump with stages, located inside the mixing condensers. 1, 2, 3 - jet stages; I, II - condensers.

Very frequently during the use of pumps of large output the necessity arises to rapidly pump the system from atmospheric pressure to a working pressure. Since vapor jet pumps possess a constancy of weight the output in contrast to mechanical pumps is characterized by a constant volume output; then, the vapor jet pump, calculated on a maximum weight output at low pressures, has a small volume output at high pressures. Therefore, pumping a volume from atmospheric pressure using such pump takes much time. For example, the pumping out of a volume of 50 m^3 with a pump of an output of 5 kg/h from atmospheric pressure to 3 mm Hg continues for several hours, although the pumping rate at 3 mm Hg is equal to 1000 m^3/h. In connection with this, in order to accelerate the pumping out process of a given volume to an operating pressure, the preliminary pumping out from atmospheric pressure to 75-100 mm Hg is done by a so-called starting vapor jet pump, possessing a large output at high pressures. The starting pump is either single- or two-stage. Frequently, the starting jet pump is not connected directly to the pumped volume, but through the main pump by means of a connection from the condensers, in which the pressure is equal to the final pressure of preliminary pumping. By employing such a connecting arrangement, during the period of preliminary pumping out, along with starting jet pump the last stage of the main jet pump also operates; it is installed behind the indicated condenser. Upon completion of the preliminary pumping out the starting jet pump is disconnected. Figure 4-37 gives a diagram of a two-stage starting jet pump, connected behind the first condenser of the main pump.

The construction of a steam-water jet pump essentially depends on its dimensions and material. Usually, jet pumps are made from the same materials as the pumping equipment. Thus, for example, the following materials are recommended:

Equipment made from steel casting	–	*the pump from cast iron or steel casting, the nozzle from bronze*

Equipment made from stainless steel	–	*pump made from stainless steel or entirely of cast iron and stainless steel*
Rubberized equipment	–	*pump is rubberized or made from porcelain*
Enameled equipment	–	*pump made from porcelain*

A pump made from porcelain is the stablest in the case of pumping out vapors of acids and solvents. In certain cases in the designs of condensers, in which strongly aggressive vapors are condensed, graphite is used. When making pumps out of steel casting it is recommended to make the nozzle from corrosion-resistant materials.

Two types of metallic jet pumps are made: cast (for small dimensions) and welded (for large construction). Figure 4-41 shows a typical design of a cast-iron jet pump.

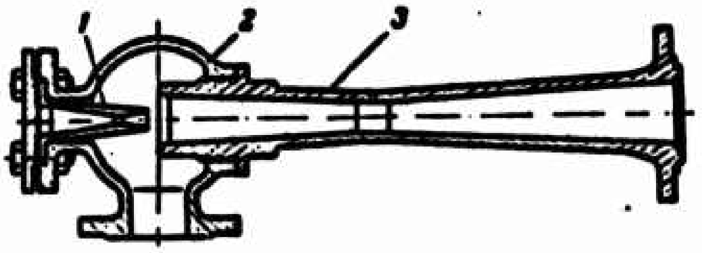


Fig. 4-41. Design of a cast-iron jet pump. 1 - nozzle; 2 - receiving chamber; 3 - diffuser with a mixing chamber.

The basic dimensions of the flow-through section of the jet pump are determined in accordance with the method and recommendations, given in Section 4-3. The dimensions of receiving chamber are determined based on design considerations. In this case the inlet pipe should possibly be made shorter and have a diameter whereby the flow rate in the pipe does not exceed 60 m/s.

Serious attention during the manufacture of the jet pump should be given to the treatment of the internal surfaces of the flow-through section. Especially the nozzle, whose internal surface is recommended for polishing, should be thoroughly treated. All angles in flow-through section should be rounded, especially the transitions from the conical surfaces to the cylindrical in the nozzle and diffuser.

b) *Characteristics.* The characteristics of four-, five- and six-stage steam-water jet pumps, expressed in the form of the dependence of suction pressure on the load, taken in percent from the calculation, are shown in Fig. 4-42. Points on the curves, corresponding 100% load, are the calculated points.

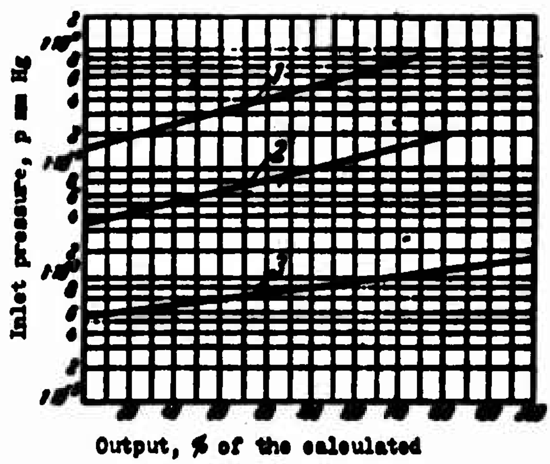


Fig. 4-42. Characteristics of four-, five-, six-stage steam-water jet pumps. Vapor pressure of 7 atm(tech.). Temperature of the cooling condenser of water is 30°C [116]. 1 - four-stage pump; 2 - five-stage pump; 3 - six-stage pump.

Figure 4-43 gives the dependence of the inlet pressure of the pump on the counterpressure, expressed in percent of the computed values. As can be seen, a small increase of counterpressure with respect to the computed value leads to a sharp increase in the inlet pressure.

Characteristics of a steam-water jet pump essentially depends on the composition of the pumped steam-gas mixture. Figure 4-44 gives typical characteristics of a pump when pumping out a steam-air

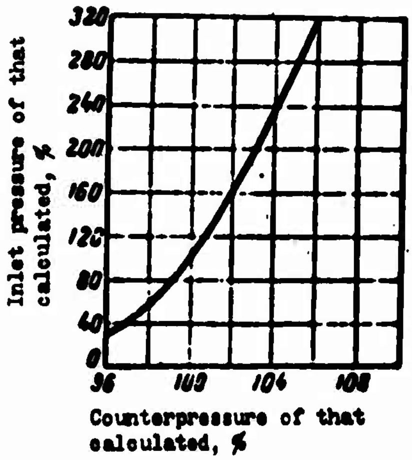


Fig. 4-43. Dependence of the inlet pressure on the counter-pressure for a steam-water jet pump, expressed in percent of the computed values [116].

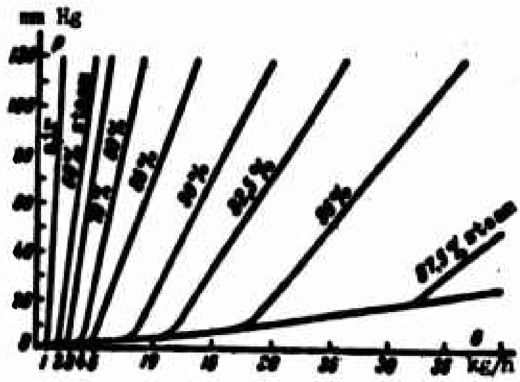


Fig. 4-44. Characteristics of $p = f(G)$ of a steam-water jet pump when pumping out a steam-air mixture [111].

mixture. As can be seen, the working section of the characteristics, short in the case of pumping out air, increases considerably with an increase in the content of vapor in the mixture. The maximum output when pumping out pure vapor exceeds the maximum output when pumping out air by tens of times. In this case the slope of the transfer sections become flatter with an increase in the steam content of the mixture.

Characteristics of the pump when pumping out a steam-air mixture essentially depends on the temperature of the mixture. The suction pressure and temperature of the mixture determine its composition, since the vapor in the mixture is assumed to be saturated. With an increase in the suction pressure with a constant temperature of the steam-air mixture its composition changes. Therefore, the characteristics, designed at a constant temperature of the mixture, correspond to a variable mixture of the ratio. Analogously, at a constant composition of the mixture with an increase in its temperature the suction pressure increases.

If the characteristics of a vapor jet pump when pumping out dry air are known, then it is possible to determine approximately its characteristics for a steam-air mixture of a known composition $g_{\text{в}} = G_{\text{в}}/G_{\text{см}}$, using the approximate relationship [78]:

$$G_{\text{см}} = (0{,}8 + 0{,}2g_{\text{в}})\,G_{\text{в(сух)}}, \qquad (4\text{-}55)$$

where $G_{\text{в(сух)}}$ – output for dry air at a pressure, for which $G_{\text{см}}$ is determined.

This dependence is useful for working sections of characteristics when $p_{\text{п}}$ = const and $g_{\text{в}} > 0.2$. If the composition of the mixture is not specified, and its temperature is $t_{\text{см}}$, then according to steam tables the pressure of saturated vapor $p_{\text{п}}$ can be determined corresponding to $t_{\text{см}}$, and then according to equation (4-52) the values $g_{\text{в}} = 1 - d$ can be calculated for every suction pressure.

The most important parameters, characterizing the operation of a steam-water jet pump, are the steam consumption (kg/h) and water (m^3/h) per 1 kg/h of pumped air. Figures 4-45 and 4-46 give the curves of flow rates of working vapor and cooling water depending upon the value of injection operating pressure of the pump, designed according to [37, 116, 117]. As can be seen from the graphs, the required steam consumption and water increase sharply with a decrease of the working injection pressure. Steam consumption and water essentially depend on the temperature of the water. Thus, for example, with an injection pressure of 5 mm Hg, a temperature of cooling water of t_1 = 15°C, a drop in the condenser of Δt = 10°C, i.e., at an outlet temperature of the condenser of t_2 = 25°C (pressure of the saturated vapor p equal to 24 mm Hg), a compression of the gas occurs from 5 to 24 mm Hg; if however, t_1 = 25°C and Δt = 10°C, then at t_2 = 35°C, p = 42 mm Hg, i.e., compression occurs

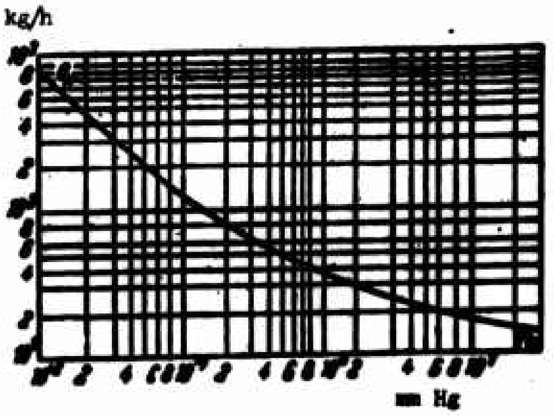


Fig. 4-45. Dependence of the flow rate of a working vapor per 1 kg/h of pumped dry air in a multistage steam-water jet pump, upon the value of the injection pressure at the calculated point (temperature of water is 30°C, vapor pressure is 7 atm(tech.).

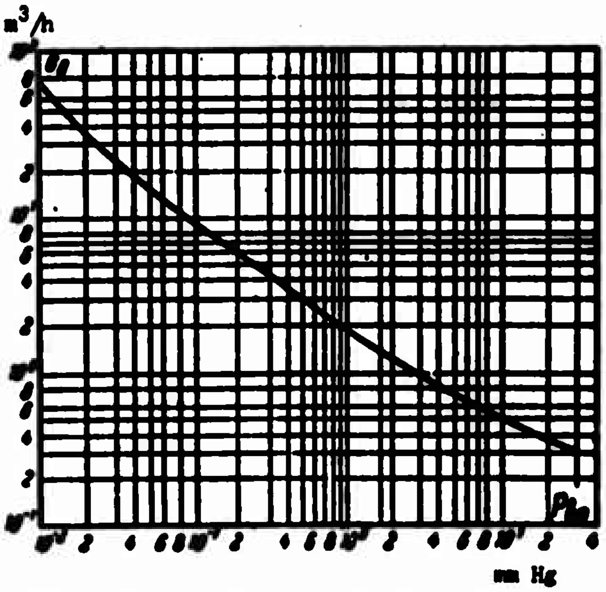


Fig. 4-46. Dependence of the flow rate of cooling water per 1 kg/h of pumped dry air in a multistage steam-water jet pump, upon the value of the injection pressure at the calculated point (temperature of water is 30°C).

from 5 to 42 mm Hg. Consumption of the vapor in this case doubles. The dependence of steam consumption and water on the temperature of water has an especially strong effect when operating lower than 1 mm Hg. Curves of the dependences of steam consumption and water on the temperature of water are shown in Fig. 4-47.

The consumption of vapor and water in the pump also depends on the vapor pressure, by decreasing with its increase (Fig. 4-48). However, the advantages of a large vapor pressure are frequently lost when operating at very low injection pressures, since in this

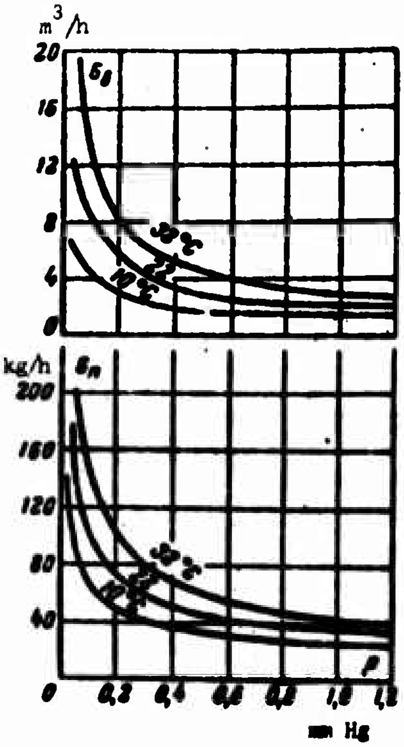


Fig. 4-47. Steam consumption of G_n and water G_B per 1 kg/h of pumped dry air depending upon the injection pressure of a five-stage steam-water jet pump at various temperatures of the water [116].

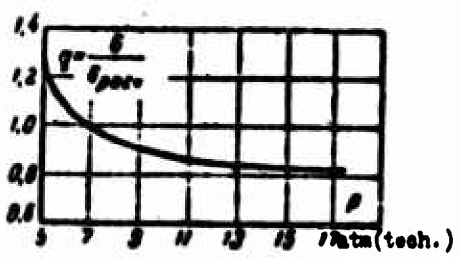


Fig. 4-48. Steam consumption in a single-stage steam-water jet pump depending upon the vapor pressure (computed pressure of p_p = 7 atm(tech.) [115].

case very small dimensions of nozzles are obtained. In a number of cases with a small output and low injection pressures the gases, for the sake of an increase in the dimensions of nozzles, operate at a vapor pressure of less than 1 atm(tech.).

The dependence of the characteristics of a steam-water jet pump upon different geometric and conditioning factors obeys the general laws, given in Section 4-3.

As already indicated in Section 4-3, the existing methods of calculation permit one to compute the steam-water jet pumps within the limits of those conditions and design parameters, for which the necessary empirical dependences and constants are determined.

Thus, one may assume that the existing methods permit one to calculate jet pumps with expansions of not more than 5000 (see Fig. 4-12) which corresponds, in general, to suction pressures on the order of 1 mm Hg. The calculation of jet pumps at a lower pressure, as the experiment turns out, yields incorrect results.

Investigations we conducted of steam-water jet pumps, operating at pressures of $\sim 10^{-2}$ mm Hg, showed that under such conditions a boundary layer has a substantial effect on the operation of the nozzle. Figure 4-49 gives the results of the measurement of the total pressure after compression shocks in the steam flow at the nozzle section. As can be seen from the figure there is a certain region of isentropic flow in the flux, where the total pressure remains constant, and there is a region near the walls of the nozzle, where the total pressure drops sharply. The region near the walls is the boundary layer. As for the investigated nozzle the thickness of the boundary layer in the outlet section with a diameter of 54 mm, amounts to ~ 8 mm, i.e., $\sim 50\%$ of the area of nozzle outlet section is occupied by the boundary layer. In this case approximately 50% of the critical section of the nozzle is also covered by the boundary layer, as a consequence of which, the actual steam consumption through the nozzle turns out to be almost one-half of that calculated. Consequently, during the calculation of similar types of jet pumps it is necessary to consider the thickness of the boundary layer in nozzles.

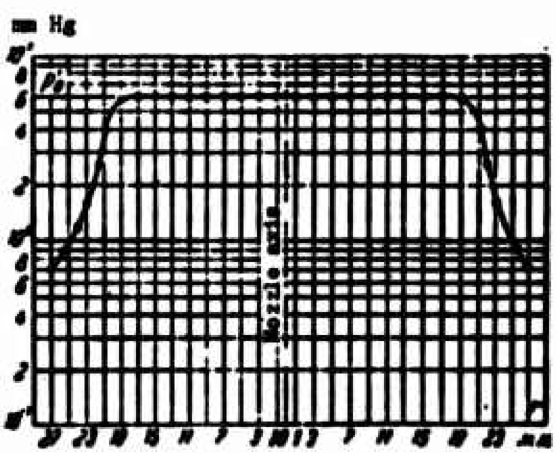


Fig. 4-49. Distribution of total pressures, measured by a Pitot fitting in the nozzle outlet section. $d_{\kappa} = 2.5$ mm; $d_1 = 54$ mm; angle of the nozzle, 25°; overheated steam: $t_{\Pi} = 190°C$, $p_{\Pi} = 0.5$ atm(tech.); $p_{\text{вп}} = 1.6 \cdot 10^{-2}$ mm Hg.

When operating jet pumps in the region of pressures of 10^{-1} to 10^{-2} mm Hg and below, the calculated expansions of the vapor in the nozzles goes into the tens and hundreds of thousands. With such expansions of the vapor, the calculated lengths of the nozzles turn out to be especially large which inevitably also leads to an increase in the losses of the flow rate. Figure 4-50 gives the dependence of the injection pressure, produced by the experimental jet pump at a constant critical load and at a constant counterpressure behind the diffuser, upon the value of the expansion of the nozzle. As can be seen from the figure, the best results are obtained during the operation of a jet pump with a nozzle, having an expansion of ∿0.4 of that calculated. The coefficient of speed ϕ_1 for a nozzle with such an expansion, based on the measurements conducted by us, amounts to ∿0.96, i.e., it differs little from that usually taken in the calculated values. Consequently, for vacuum jet pumps, operating with large expansions of vapor, one should design a nozzle with considerable underexpansion with respect to the computed value (∿0.4).

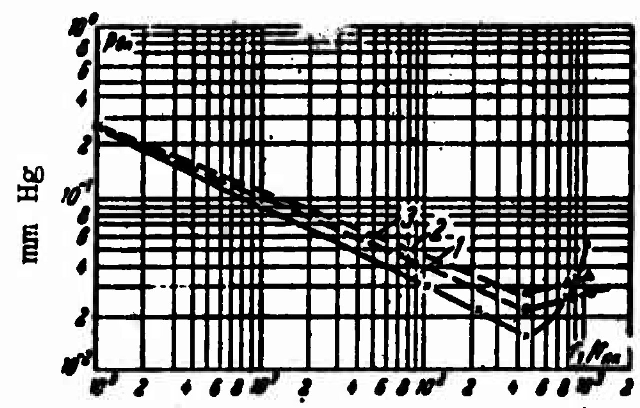


Fig. 4-50. Dependence of the injection pressure of a jet pump on the expansion of the nozzle at a constant critical load and constant counterpressure (nozzle: $d_к$ = 2.5 mm; overheated steam: $t_п$ = 170°C; $p_п$ = 0.5 atm(tech.); calculated expansion of the nozzle, 1250). 1 - angle of nozzle, 19°; 2 - angle of the nozzle, 30°; 3 - angle of nozzle, 40°.

Figures 4-51 to 4-53 give the dependences of the pressures produced by the vacuum jet pump upon a number of other geometric

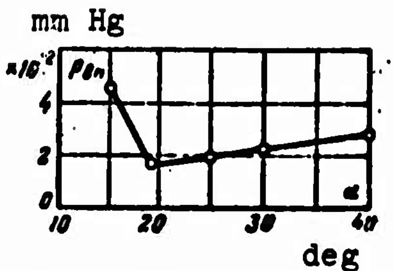


Fig. 4-51.

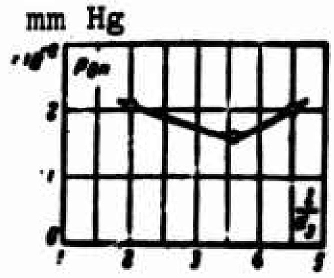


Fig. 4-52.

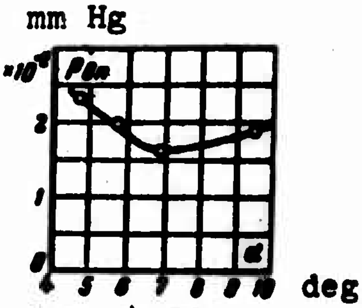


Fig. 4-53.

Fig. 4-51. Dependence of the injection pressure of the jet pump upon the cone angle of the nozzle at a constant critical load and constant counterpressure (nozzle: $d_{к}$ = 2.5 mm; d_1 = 54 mm; overheated steam: $t_{п}$ = 170°C; $p_{п}$ = 0.5 atm(tech.).

Fig. 4-52. Dependence of the injection pressure of the jet pump on the length of the throat of the diffuser at a constant critical load and constant counterpressure (nozzle: $d_{к}$ = 2.5 mm; d_1 = 54 mm; effuser: d_2 = 187 mm; l = 745 mm; throat d_3 = 100 mm; diffuser: d_4 = 150 mm; l = 535 mm; overheated steam: $t_{п}$ = 170°C, $p_{п}$ = 0.5 atm(tech.).

Fig. 4-53. Dependence of the injection pressure of the jet pump on the cone angle of the effuser at a constant critical load and constant counterpressure (nozzle: $d_{к}$ = 2.5 mm; d_1 = 54 mm; effuser: d_2 = 187 mm; throat: d_3 = 100 mm; l = 350 mm; diffuser: d_4 = 150 mm; l = 535 mm; overheated steam: $t_{п}$ = 170°C; $p_{п}$ = 0.5 atm(tech.).

parameters. As can be seen from the figures, the best characteristics of the jet pump are obtained at a cone angle of the nozzle ~19°, length of the throat of the diffuser, $l = 3.5d_3$ and the angle of the effuser $\alpha = 6.75°$, corresponding to a conicity of 1:85. The conducted investigations also showed that the best characteristics of the jet pump are obtained with the location of the nozzle outlet section in the plane of inlet section of the effuser, with the conicity of the diffuser 1:10, and at flow rates in the outlet section of the diffuser of 70-80 m/s.

c) *Exploitation. Application.* Steam-water jet pumps are used widely in different branches of industry. They are used for pumping in various kinds of distillation installations, evaporating apparatuses, vacuum-crystallizers, deaerators, condensers of vacuum-filters, and vacuum-impregnating equipment. They are widely used also in refrigerating equipment, in metallurgy for pumping out vacuum arc and induction furnaces, and extrafurnace installations in the degasification of liquid metal [160].

Advantages. Such extensive use of steam-water jet pumps can be attributed to a number of their important merits:

1. They can be designed and made for practically any arbitrarily large volume output, whereas other types of vacuum pumps have fully defined limitations on output, governed by their commercially feasible and expedient dimensions. Thus, for example, one vapor jet pump based on an average output of 100 kg/h at 0.5 mm Hg (volume output of 32,000 l/s) is able to replace over 270 such big mechanical pumps, as the VN-6. The effect of such a replacement is seen from Table 4-1.

Table 4-1.

Type of pump	Quantity	Total output, l/s	Area in plan, m^2	Weight, t	Water consumption, m^3/h	Consumed energy
Steam-water jet pump NEV-100×0.5..	1	32000	15	10	300	Steam 7 atm(tech.) 5000 kg/h
Mechanical vacuum pump VN-6.........	270	32000	3000	560	280	5600 kW

2. They do not have moving parts and have extraordinarily simple construction. Due to this their operation is very reliable, and their service-life is practically unlimited.

3. They can be made from any materials and designed to pump any gases, including aggressive gases and those contaminated by

mechanical impurities. When pumping out a dust-laden medium, filters need not be installed, thanks to their all-consuming output.

4. They have little weight. Foundations are not required for their installation.

5. Thanks to their simplicity of construction, the manufacturing cost of vapor jet pumps is low. Thus, according to [116] a steam-water jet pump with an output of 2 kg/h at pressures of 10^{-2} mm Hg costs almost 2.5 times less than a set of booster and mechanical pumps of the same output. The cost of vapor jet pumps increases with an increase in the output (Fig. 4-54).

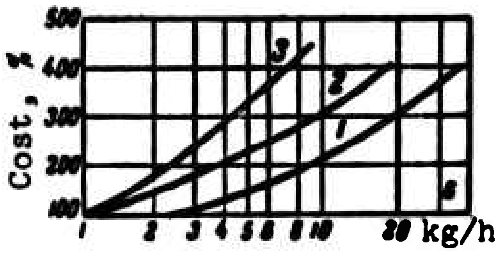


Fig. 4-54. Relative cost of steam-water jet pumps (in percent) depending upon their output (averaged curves, plotted according to [117]). 1 - four-stage pumps (0.25-2 mm Hg); 2 - five-stage pumps (0.05-0.2 mm Hg); 3 - six-stage pumps (0.005-0.03 mm Hg).

6. They are simple and cheap to operate. According to [114] the cost of running a vapor jet pump turns out to be one-half that of the cost of running a system consisting of a vapor-oil booster pump in a set with a mechanical one, even under conditions whereby operating speed of the jet pump is twice that of the operating speed of the booster pump.

7. They are compact. Separate elements of the pump can be placed on walls, columns, ceilings, so that the occupied space or industrial area turns out to be insignificant. As for the location of the pumps, as a rule, an isolated location is not required. They can be placed either in the workshop, or outside.

Steam and water. The main problems with using steam-water jet pumps are the safe supply of steam of the required quality and at the required pressure, as well as the water at the required temperature. Vapor pressure in any case should not be less than that calculated for the normal operation of the pump. If steam of greater pressure is present, then it can be choked to the operating pressure in the pump either by means of a reduction valve, or by installing a diaphragm in the steam-feed pipe. In the latter case the computed dimensions of the diaphragm can be done based on the graph in Fig. 3-20 in accordance with instructions given in Section 3-4. With the choking of the steam to the operating pressure, as a rule, the normal operation of the pump is ensured. For the normal operation of a pump with intermediate condensers a variation of vapor pressure of more than 10% is not permissible and for pumps with stages without intermediate condensers - not more than 5%.

As already indicated, the characteristics of a pump essentially depends on the quality of the steam. If the steam received from the boiler room is humid, then in front of its inlet it is necessary to install a moisture separator with condensation cistern on the steam-feed line in the pump. When using multistage pumps, operating at pressures lower than 1 mm Hg, slightly overheated steam (by 20 to 30°C) is recommended. With strongly superheated steam the operation of the pump, designed for saturated vapor, becomes unstable.

The temperature of the water, issuing from the cooling condensers, should not be higher than that designed for their normal operation. In this case it is essential, as to what kind of water is cooling the condensers: artesian, river or white. With river and white water considerable variations of temperature are possible depending upon the season. Therefore, in the appraisal of the possibility of using such water in a pump it is necessary to establish the highest seasonal temperature. With artesian water whose temperature, as a rule, turns out to be lower than the operating temperature in the condensers of the pump, it is possible to economize on steam by reducing it to such a figure in order not to worsen the characteristics of the pump.

If the vapor pressure and temperature of the cooling water have values, equal to the operating values for the pump, then the characteristics of the pump have a fully defined quantitative dependence of the load upon the injection pressure, measured based on its tests. Therefore, it is possible to use the characteristics of vapor jet pump with a normal safeguard of its vapor and water for an appraisal of the quantity of pumped gas based on the measured injection pressure.

Control of output. Usually, the load in operation on a vapor jet pump can be changed in the process of pumping out. In this case frequently the full pump capacity is used only during the initial period of work, but subsequently it can be substantially decreased without damage to the operation of pumped installation. In contrast to mechanical pumps with electric motor drive, where the reduction of the load automatically lowers the consumption of power, the steam consumption in vapor jet pumps remains constant independently of the change of load. In connection with this during the exploitation of vapor jet pumps the compulsory control of the pump capacity has great importance. Usually, such control has, as an aim, the saving of steam consumption and water with a reduction of the load on the pump. However, in certain cases the control of output necessitates maintaining a pressure in pumped installation to an assigned level based on assigned program. In this case, economic considerations play a secondary role.

The control of the pump capacity, as already indicated, can be carried out conveniently by means of a cutoff of one or several working stages of the pump in parallel. In this case a reduction of the output is accompanied by a reduction of the steam consumption and water. In principle, it is also possible to control the pump capacity by means of a reduction of the vapor pressure and consumption of water. However, this method is much more complex than the first, since it requires the knowledge of the characteristics of the pump depending upon the change in vapor pressure and consumption of water and high accuracy in their adjustment. Usually, this method is not used in operation.

For the indispensable work of a pumping installation based on an assigned program of a change of the injection pressure, the most convenient method of control of output of vapor jet pump is the reduction of the pumped flow of gas by a valve or lock on the suction line. In this case, control can be done automatically owing to the manipulation of the drive of the valve or lock from the signal, given by the pressure sensor, installed in the pumped system.

Diffusion of vapor from the pump in the system. When using steam-water jet pumps the problem of the penetration of steam from the pump into the pumped system is important. The amount of penetration of vapor in the system can be conveniently characterized by the moisture retention or partial pressure of the vapor in the system. Moisture content is connected with the partial pressure by the equation

$$x = \frac{M_n}{M_r} \frac{p_n}{p - p_n}, \qquad (4\text{-}56)$$

where M_r and M_n - molecular weight of gas and vapor; p - total pressure in the pumped volume; p_n - partial pressure of vapor in the pumped volume.

Thus, in this case or in some other case, the problem of determining the quantity of steam, penetrating into the pumped volume, amounts to the determination of the partial pressure of vapor, which can be simply established based on the measurement of the dew point.

The diffusion of steam from a vapor jet pump essentially depends on the construction, its operation conditions, produced pressure and on the value of external gas load. Figure 4-55 gives the dependence of the dew point by volume, pumped by a NEV-3 pump from the flow of air in the pump, produced near the injection pipe. The dew point was measured by a condensation hygrometer. As can be seen from Fig. 4-55 the dew point in the pumped volume considerably decreases with an increase in quantity of injected air in the pump. Furthermore, the dew point essentially depends on the degree of dryness of the injected air. Thus during the operation of the pump

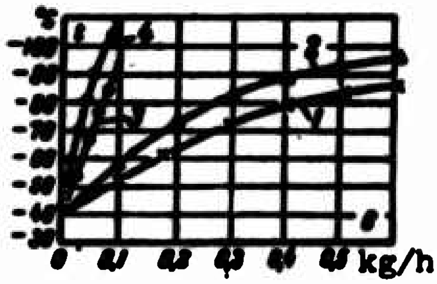


Fig. 4-55. Dependence of the dew point in the pumped volume, upon the flow of air in the NEV-3 pump. 1 - humid undried air; 2 - air passed through one U-shaped drier, filled with silica gel; 3 - air passed through two silica gel driers; one cooled by liquid nitrogen; 4 - air passed through two silica gel driers, cooled by liquid nitrogen.

without a gas load, under conditions of critical vacuum, the dew point in the pumped volume is equal to -40°C which corresponds to a vapor pressure of $1 \cdot 10^{-1}$ mm Hg, and a moisture content of 0.31 kg/kg (critical vacuum of the pump, p = 0.3 mm Hg). With a load of 0.1 kg/h, it comprises ~10% of the critical pump capacity of the NEV-3 (see Fig. 4-39); for humid atmospheric air (curve 1) the dew point decreases to -52°C which corresponds to a partial vapor pressure of $2 \cdot 10^{-2}$ mm Hg and a moisture content of $4.4 \cdot 10^{-2}$ kg/kg. For dried air (curve 4) the dew point with the same load falls to -110°C which corresponds to a partial vapor pressure of $\sim 1.5 \cdot 10^{-6}$ mm Hg and a moisture content of $\sim 2 \cdot 10^{-6}$ kg/kg.

Thus, by creating constant leakage in the pump, the air or some other gas, especially dried, can reduce the diffusion of vapor from the pump to a very small value. Suppression of diffusion in this case is caused by the fact that the diffusion of vapor toward viscous or turbulent flow of the gas is substantially hampered, and to an even greater degree, the larger the flow value. In connection with this, in those cases where the technological process at the pumping installation goes on without the isolation of the gases, a pump is specially loaded with a small quantity of air or some inert gas [114] to prevent the diffusion of vapors of the water in the installation.

Assembly. In spite of the simplicity of the device and the use of vapor jet pumps, it is necessary to observe specific guidelines during their assembly.

The location of the pumps, as already indicated, can be inside the building or outside it. The stages of the pump can be arranged arbitrarily depending upon the convenience of their location and the barometric mixing condensers should always be erected strictly vertically. During the installation of a pump it is necessary to ensure free access to all its jointed connections, valves, bolts and measuring instruments. The first stage of a pump should be located as near as possible to the pumping chamber.

The diameter and length of the connecting pipeline between the pump and the pumped chamber should be selected taking into account the minimum losses of its output (see calculations, Chapter 9).

The steam-feed pipes leading to the stages are designed in accordance with the steam consumption in the stage and taking into account the tolerated speed of the vapor in the pipe, 30 m/s. Table 4-2 gives the quantity of vapor (in kg/h), flowing at a speed of 30 m/s through pipes of different diameter and at a different pressure.

Table 4-2.

Vapor pressure	Internal diameter of a steam pipeline, mm							
	20	25	30	40	50	70	80	100
1	37,5	58,5	96	150	236	480	600	940
1,5	47	73,5	121	188	295	575	750	1 175
2	55	86	142	220	345	675	875	1 375
2,5	63,5	99,5	164	255	400	780	1 015	1 595
3	72	117,5	196	302	475	925	1 200	1 885
4	89	138	230	357	555	1 080	1 410	2 220
5	105	165	272	420	650	1 280	1 670	2 620
6	122	190	310	490	765	1 500	1 980	3 000
8	155	242	388	620	970	1 900	2 470	3 870
10	187	292	460	750	1 170	2 300	2 980	4 670
12	220	343	565	880	1 370	2 700	3 500	5 500
15	269	420	690	1 080	1 690	3 300	4 300	6 740
18	318	494	815	1 270	2 000	3 900	5 070	8 000
20	349	545	895	1 400	2 190	4 270	5 600	8 720
25	432	675	1 100	1 730	2 700	5 300	6 900	10 870

Using a steam header it is necessary to provide drain tube with a valve in order to blow the steam system and collect the accumulated condensate before starting up the pump. All steam-feed pipes should be well heat-insulated.

On steam hook-ups in front of nozzles with a $d_{\text{кр}} \leq 5$ mm it is necessary to install screens, protecting the nozzle from obstruction. The functional cross section of the screen should be at least 10 times more critical than the section of nozzle $f_{\text{п.к}}$. The holes in the grid should be approximately one-half of the $d_{\text{кр}}$ of the nozzle.

The barometric condensers one should set at a height of not less than 11 m above the level of the water in the barometric cistern. The overflow barometric pipes should be as straight as possible. Horizontal sections in the piping should be avoided, since the air dissolved in the water can be separated out in them and thereby interrupt the flow of water. The ends of the barometric pipes immersed in water, should be at some distance, based on the diameter of pipe from the bottom of the barometric cistern, but not less than 100 mm.

The supply of water in the condensers should be fed through sufficiently wide pipes, while excluding large losses of pressure. The selection of the pipe sizes depends upon the flow rate; see Appendix 10.

The accident-free operation of the vapor jet pump essentially depends on the reliability of control of its operation. With this in mind it is necessary to install controlling and measuring equipment on the pump. On the suction line of every stage a manometer is installed to control the operating pressure. On the steam header a manometer is installed to control the vapor pressure, and a thermometer to control its temperature. On the water collector a manometer and thermometer are installed; on every water line, leading to the condenser, flowmeters are installed. On the overflow pipes behind the condensers thermometers are installed to control the temperature of the discharged water; in this case, the thermometer bulb should not substantially reduce the section of pipe. During the pumping out of dust-laden gases, the dust may be precipitated in the condensers and overflow barometric pipes; in this case it is necessary to set a signalling apparatus level in the lower part of the condenser in order to detect the obstruction in the overflow pipe within due time.

Modern vapor jet pumps have been improved with automatic control and operational control. The basic requirement for automatic control amounts to the automatic opening and closing of the necessary valves and gates by signals issuing from the control panel and from sensors of control-measuring equipment. One can run the automatic control with the corresponding signalling and blocking, including: the pressure of the working vapor, temperature of incoming and outgoing water of the condenser, the pressure of pumped gas on the suction line, the level of water in the barometric pipes. The reduction of the vapor pressure, increase in the temperature of the outgoing water, and the level of water in the barometric pipes within more fixed limits is evidence of the critical operation of the pump. In each of these cases signals have to be given for the automatic shutting of valves and gates on the suction line of the gas, for the pipelines supplying vapor and water.

Starting and stopping. Prior to the first starting the pump should be checked for airtightness. A check on airtightness can be done by means of pressing the pump internally by a gas test using high pressure. For this purpose the pump is disconnected from the suction line, steam and water hook-ups, and is sealed vacuum-tight. When searching for leaks, one can use a halide and differential leak detectors. It is also possible to use halide leak detectors without pressing the pump internally. For this purpose, an atmospheric sensor is placed in the exhaust pipe of the pump; the pump is put in operation and the lock on the suction line is covered. Then, one produces blowoff of jointed vacuum connections of the pump with haloid-containing gas (freon). The test gas, penetrating through any loose connections, will be compressed in the pump and ejected in the exhaust pipeline, where it will be registered by the leak detector.

The check on airtightness is done during the first starting of the pump. In process of operation there is no need to conduct the test for airtightness, if the pump produces a normal residual pressure.

During the starting of the pump the water at first is injected into the condensers, but then vapor enters the stage. In pumps with the water-jet stage the water first moves into the water-jet stage. The quantity of water passed into the condensers is controlled by flowmeters. During the first starting of the pump the quantity of water is adjusted according to temperature of outgoing water of the condensers. Before injecting the vapor in the stage it is necessary to blow the line with vapor using the open depressurizing valve on the steam header in order to discharge the condensate accumulated in steam pipelines. When first starting the pump it is recommended to blow with vapor or compressed air at high pressure to remove cinder and mud from the steam pipelines. After such blowing one should have cleaned the protective screens and nozzles.

When stopping the pump first the vapor is shut down, then the water. When stopping a pump having a water-jet stage the shutting off of the water passed in water-jet stage should be done carefully to avoid a sudden suction of water from the stage into the system. In small pumps without intermediate condensers it is recommended in this case to fill the pump with air in advance.

Malfunctions and their elimination. The need for detection and elimination of malfunctions arises mainly when the pump does not produce the necessary operating pressure in the pumped apparatus. First of all it is necessary to check whether there is poor pressure as a result of the defective operation of the pump or if it is caused by a lack of airtightness of the apparatus. In order to do this the pump is disconnected from the apparatus with a vacuum lock (in the absence of a lock, it is disconnected from the apparatus and muffled using a suction pipe) and its maximum vacuum is checked. If the pump produces normal maximum vacuum (tentatively for a single-stage pump – 50 mm Hg, two-stage – 5-10 mm Hg, three-stage – 2-3 mm Hg, four-stage – 0.1-0.2 mm Hg, five-stage – 0.02-0.05 mm Hg, six-stage – 0.003-0.005 mm Hg), then it is possible (as the experiment shows) to expect that the pump will operate normally and with a load. In this case the probable cause of the malfunctioning may be the unhermeticity of the apparatus. Therefore, one should measure the magnitude of leakage in the apparatus and compare it with the norm.

To measure the leakage, the apparatus is pumped to a certain minimum pressure, which can be produced in it, and then it is disconnected to stop the pumping out process. The increase in pressure Δp, in the volume of the apparatus V, is measured over a certain time τ at a constant temperature and the leakage is calculated by the equation

$$G=\frac{V\Delta p}{\tau}. \tag{4-57}$$

If the volume of the apparatus is unknown, then one can connect a known leakage G' to the apparatus and make two measurements: with the leak G', and without it. The detected leak then can be determined from the equation

$$G=\frac{G'}{\frac{\Delta p'}{\Delta p}\frac{\tau}{\tau'}-1}, \tag{4-58}$$

where $\Delta p'$ - increase in pressure in the volume V, during the time τ' in the presence of the known leak, G'.

The value G should be less than $G_{\text{норм}}$. The norm of leakage is usually furnished from the calculation

$$G_{\text{норм}}<0.1G_{\text{г(мин)}}, \tag{4-59}$$

where $G_{\text{г(мин)}}$ - output of a vapor jet pump, utilized for pumping out an apparatus, at a minimum operating pressure.

If, however, as a result of a preliminary check it turns out that the malfunctioning remains hidden in the pump, then to detect it, it is necessary to start checking the suction pressure by stages. The distribution of the suction pressure by stages should correspond to the normal distribution for a given pump. If, in some stage an excess of normal suction pressure is observed, then this is evidence of the defective operation of this stage. The basic causes of malfunctioning can be, as follows:

1. The vapor pressure, passed in the stage, does not correspond to the operating value. It is necessary to regulate the vapor pressure.

2. The increase in humidity of working vapor due to an interference of the operation of the water separator or steam superheater (when operating on superheated steam). This cause can be especially important for the first stages of multistage pumps, operating at a pressure lower than 1 mm Hg. It is necessary to check and to eliminate the malfunction in the water separator or in the steam superheater.

3. Clogged nozzle or protective screen in front of the nozzle. It is necessary to clean them mechanically or by blowing. The formation of a deposit on the walls of the nozzle is possible. Scale is easily removed by polishing.

4. Abnormal operation of the condenser in front of a stage, resulting in an increase of pressure in the condenser. This can occur because of the insufficient quantity of cooling water or because of the high temperature of the incoming water. In both cases the temperature of the outgoing water turns out to be higher than the normal. It is necessary to adjust the consumption of the cooling water. Furthermore, with an obstruction in the barometric overflow pipes, or a loss of their seal, or an incorrect setting can all cause the flooding of the condenser by water, thereby interfering with its operation. Frequently, the flooding of the condenser or a partial disturbance of the overflow of the condensate

appears as a pulsation of the pressure in front of the stage, as vibrations of the condenser, as dull water hammers in it. It is necessary to identify the defect in the overflow pipe and to eliminate it.

5. Leakage in the vacuum connection of the stage or the condenser ahead of it. One should find the leakage and remove it using the above-indicated method.

6. Counterpressure higher than normal behind a stage. The possible cause can be an obstruction in the pipe, connecting the stage with the condenser behind it. For the last stage, operating with exhaust into the atmosphere, the cause can be an obstruction in the exhaust line. In both cases one should clean out the dirty lines.

7. The clogging up of the flow-through part of a stage by scale or contaminants during the process of prolonged use. Cleaning can be done mechanically and by polishing.

8. The wear of parts of a stage: nozzle, mixing chamber, and diffuser due to erosion (and in a number of cases also by corrosion) during the process of prolonged use. The worn out parts should be replaced by new ones.

In the analysis of malfunctions in the operation of a pump one should consider that the variation of pressure on suction line of the pump during its operation with a load is not always evidence of a defect; in many cases they are the result of the operation of the pump at an overload point of its characteristics.

CHAPTER V

BOOSTER PUMPS

5-1. Working Principle and Design

Booster pumps are used for pumping out large quantities of gas from vacuum installations at pressures of 10^{-1}-10^{-4} mm Hg.

The basic requirements imposed on the characteristics of booster pumps are: high operating speed within a specified range of pressures and a large value of the maximum outlet pressure.

The region of operating pressures of booster pumps is characterized by the fact that the conditions of flow of the pumped gas is transition from purely viscous - on the upper limit of the range (10^{-1} mm Hg) - to purely molecular - on lower limit of the range (10^{-4} mm Hg). This feature of the operating conditions of the pump is determined to a considerable degree by the mechanism of pumping out the gas by the pump itself.

The increase in gas by the steam jet in the booster pump, as already indicated, is governed by the viscous friction at the jet-gas boundary and by the diffusion of the gas in the jet. At high pressures the process of the viscous "capture" of the gas is the determining factor, at low pressures - the process of diffusion. In connection with this for best pumping action the jet should be sufficiently dense at high pressures, at low pressures - sufficiently rarefied. But since the conditions of discharge of the steam jet do not hardly depend on the pressure of the pumped gas in the operating range of the injection pressures and since the process of pumping out remains constant, then it is necessary to select it of such a design, so as to obtain a sufficiently high operating speed throughout the entire operating region at both high as well as low pressures. It is natural that such conditions will not be optimum for separate operation at high and low pressures, but will be optimum for the entire operating range of pressures on the whole. Accordingly, the operating speed of the pump depending upon the injection pressure is depicted as a curve with a maximum at a certain average pressure for a selected range of operating pressures.

By changing the conditions of discharge of steam jet (for example, by a change in the power setting on the pump), it is possible to shift the maximum of curve within the range of high or low pressures.

To illustrate the described phenomena, Fig. 5-1 gives the dependence of the operating speed on the pressure at different degrees of preheating for a vapor-oil BN-3 booster pump. From Fig. 5-1 it is clear that, for example, under discharge conditions of the steam jet, corresponding to the power of 3.5 kW, the optimum for the range of pressures is $2 \cdot 10^{-3}$-$1 \cdot 10^{-2}$ mm Hg, the maximum operating speed of 500 l/s is attained at a pressure of $5 \cdot 10^{-3}$ mm Hg; at a pressure of $1 \cdot 10^{-2}$ mm Hg the specified power (accordingly, also the density of steam jet) turns out to be less than optimum (4.5 kW), and at a pressure of $2 \cdot 10^{-3}$ mm Hg - more than the optimum (2.5 kW).

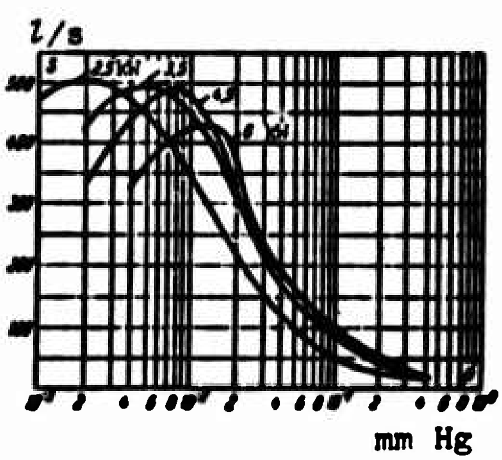


Fig. 5-1. Dependence of the operating speed of BN-3 vapor-oil booster pump on the injection pressure at various degrees of preheating.

To produce a high value of the maximum outlet pressure it is necessary to produce a steam jet with a large density. At the same time, to produce a high operating speed within a wide range of operating pressures, the density of the jet, as already indicated, does not have to be too high. To meet such contradictory requirements is possible by means of using multistage construction, which schematically can be represented in the form of several consecutively connected single-stage pumps (Fig. 5-2). In this case the discharge conditions of the steam jet in the output stage is selected in such a way as to obtain a high peak outlet pressure. In the preceding stage the density of the jet can be set according to the smaller one, since the jet works against the smaller outlet pressure; in the following stage, even less, etc. Under the initial discharge conditions, the optimum for the specified range of operating pressures is adjusted from the gas injection side of the jet.

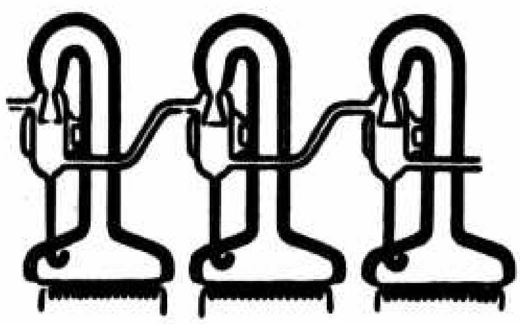


Fig. 5-2. Diagram of multistage pumping out.

When realizing a multistage design of a pump it is necessary that every subsequent stage provides for the operation of the preceding stage. The criterion for establishing characteristics of the separate stages is a continuity equation of the flow of G = const, i.e., $S_1p_1 = S_2p_2 = S_3p_3 = \ldots = S_np_p$; here S_1, S_2, S_3, ..., S_n - operating speed of the corresponding stages, p_1 - the maximum value of the injection pressure of the first stage in the specified range of operating pressures; p_2, p_3, ..., p_n - corresponding injection pressures of the remaining stages, equal by value to the values of the peak outlet pressure of the preceding stages, namely: $p_2 = p_{н.в}$ of the first stage; $p_3 = p_{н.в}$ of the second stage, etc.

The typical characteristics of a multistage booster pump are shown in Fig. 5-3. The intersection of the characteristics of every stage with the line of the peak output of the preceding stage in the operating section (dot-dash line) determines the injection pressure for the given stage, and accordingly, the counterpressure for the preceding stage. Thus, pressure p_2 is the peak outlet pressure for the first stage; pressure p_3' is the actual counterpressure for the second stage when $G_{макс}$, and p_3 - the peak outlet pressure of the second stage.

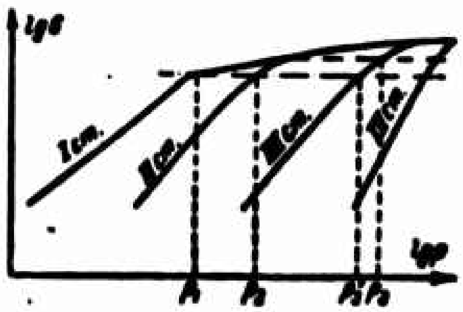


Fig. 5-3. Characteristics of a multistage booster pump [cm = stage].

Realization of multistage design with the use of axisymmetrical direct flow[1] of nozzles according to the diagram whoen in Fig. 5-2, is structurally very bulky and inconvenient. Therefore, in modern designs of booster pumps direct flow is not used, but a so-called "reversed" nozzle, in which the steam flow changes its direction.

The reversed nozzles are made into different designs; the most widely used nozzles are of the "umbellate" type. The typical construction of such a nozzle is shown in Fig. 5-4. The nozzle consists of two components: the pedestal (a) and the hood (b). The vapor proceeds through the steam-feed pipe, connected to the pedestal;

[1]By direct flow is meant a nozzle, in which the flow does not change its direction.

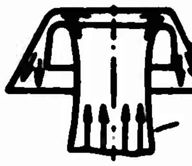


Fig. 5-4. Diagram of a reversed nozzle of the umbellate type. a) pedestal, b) hood of the nozzle.

then, by changing its direction, completely opposite, it passes through the smallest section of the nozzle (critical section) and is expanded in the annulus, formed by the pedestal and the hood. All elements of the Laval nozzle are inherent in such a nozzle: here, just as in the Laval nozzle the vapor, moving at subsonic speed in the steam-feed pipe, is accelerated to the critical speed in the smallest section of the nozzle and then, by passing through the expanded channel, attains supersonic speed. The application of the reversed umbellate nozzles permits one to make multistage pumps in the form of simple and compact designs. Usually, all the stages of the pump are placed in one housing and are supplied with vapor from one boiler.

The vapor pressure in the boilers of the booster pumps usually amounts to 10-20 mm Hg, but in certain cases, reaches 40 mm Hg [166]. The nozzles are made with an angle of inclination 60-70° to the horizontal in forming the hood, perpendicular to the axis of the pump. The magnitude of expansion of nozzles can be changed from 2-3 for outlet stages to 20-50, and sometimes more for injection stages.

Figure 5-5 gives the design of a two-stage BN-3 vapor-oil booster pump. The pump consists of a cylindrical housing with a welded boiler in the lower part. Inside the housing a system of nozzles and steam-feed pipes have been installed, called collectively, the "steam pipeline" of the pump. The steam from the boiler passes through the central steam-feed pipe to the first stage (upper nozzle) and through the annulus between the external and central pipe - to the second stage (lower nozzle). The condensate of the vapor flows along the wall of the housing and through the overflow pipe, back to the boiler of the pump.

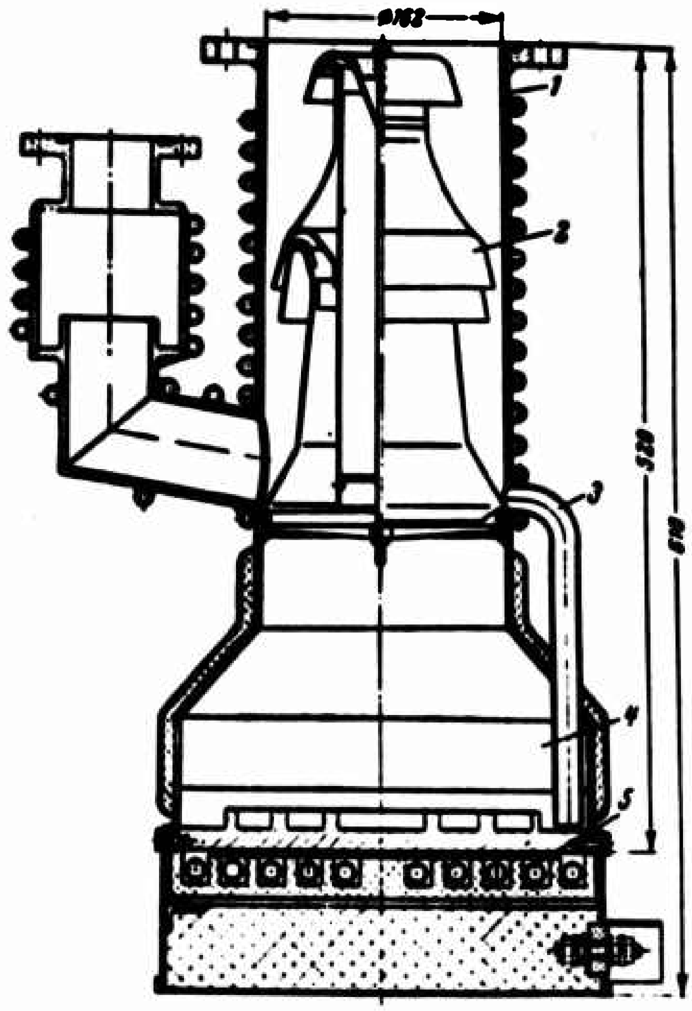


Fig. 5-5. BN-3 vapor-oil booster pump. 1 - housing; 2 - steam pipeline; 3 - overflow pipe; 4 - boiler; 5 - electric heater.

In the BN-3 pump, as well as in the majority of other booster pumps of similar type, the boiler has dimensions somewhat enlarged in comparison with the diameter of the housing of the pump. This is caused by the fact that the quantity of heat (power of preheating), necessary to operate the booster pump, is relatively large, and therefore, to avoid large specific thermal loads on the bottom of the boiler it is necessary to enlarge its surface. Thus, in the BN-3 pump the specific thermal load amounts to ~ 6 W/cm^2 of the bottom of the boiler. Even with such a thermal load, enough strong turbulence of oil in the boiler of the pump occurs, worsening the heat transfer from the bottom of the boiler to the oil and resulting, furthermore, in an irrational expenditure of power for lifting liquid drops from the surface of the oil during the turbulence. For the calm boiling of the oil without turbulence the specific thermal load should not exceed 3-4 W/cm^2. In the designs of boilers, similar to the boiler in the BN-3 pump, the specific thermal loads usually exceed this value, since otherwise the overall dimensions of the boiler turn out to be very considerable.

In general, boilers of such a type along with their simplicity in design possess a number of substantial deficiencies. In this case just as shown here - large thermal loads - one should also note the considerable thermal losses (∿20% of the input power) and the difficulty of realizing an overflow of the working fluid from the pump and its flushing (the presence of the electric heater suspended from beneath hampers the potential of the overflow pipes from the boiler of the device).

To a considerable degree the design of the boiler of the BN-4500 pump (Fig. 5-6) is free of these deficiencies. This boiler has the form of a parallelepiped, in one of the lateral walls of which six pipes of rectangular section are seated; inside the pipes electric heater elements are inserted, constituting nichrome coils, suspended in alundum tubing, secured on the metallic frame. In the boiler the oil is gushed in such a quantity that the pipes are completely immersed in it. The heat-transfer surface is the external surface of the pipes. The specific thermal load for this boiler amounts to ∿2.5 W/cm^2. The boiler has overall dimensions of 650 × 650 mm. If the boiler were made according to BN-3 type, then at a specific thermal load of 2.5 W/cm^2, the diameter of the boiler would be equal to 1 m, i.e., the overall dimensions would be larger. Thermal losses in the boiler amount to 5% of the input power, i.e., one-fourth less than in the boiler of the first type. The bottom of the boiler has a small cone, in the center of which is a connecting pipe for the overflow of the working fluid from the pump.

In the BN-4500 pump the two last stages differ from the stages with conventional nozzles of the umbellate type. Each of them constitutes a set of axisymmetrical direct flow nozzles, fixed

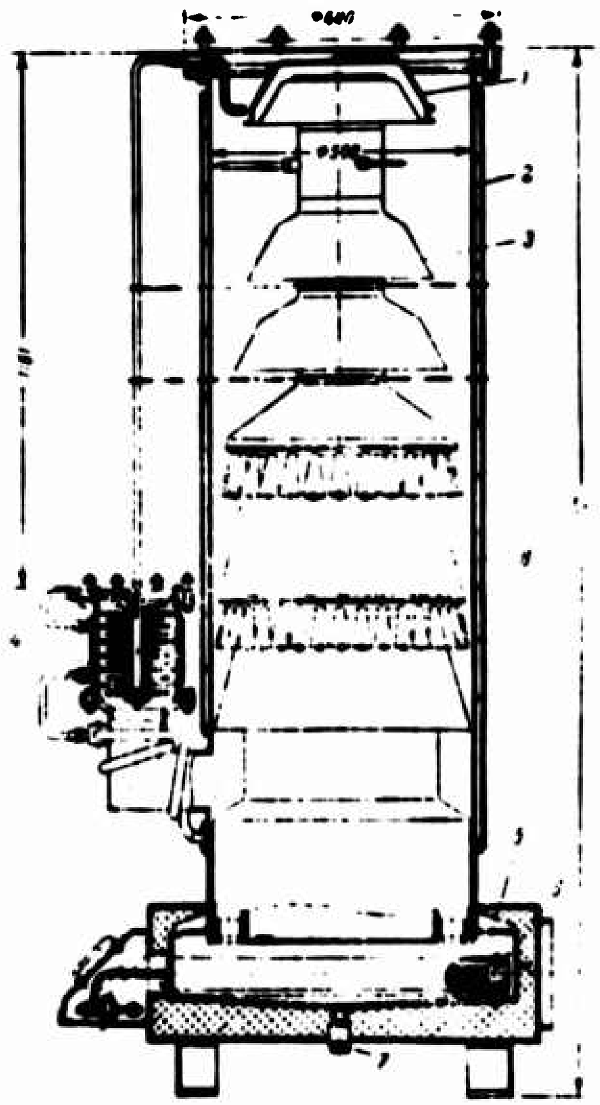


Fig. 5-6. Diagram of the construction of the BN-4500 vapor-oil booster pump. 1 - oil deflector; 2 - housing; 3 - steam pipeline; 4 - oil trap; 5 - boiler; 6 - electric heater; 7 - pipe connection for the overflow of oil; 8 - cowl.

generally as a collector, and working in parallel. Such a stage in contrast to the stage with a nozzle of the umbellate type permits one to produce at some given size a considerably larger surface of the steam jet, and consequently, at some given steam consumption, to produce a large output.

Figure 5-7 gives the comparative characteristics of two experimental stages with a nozzle of the umbellate type, and a set

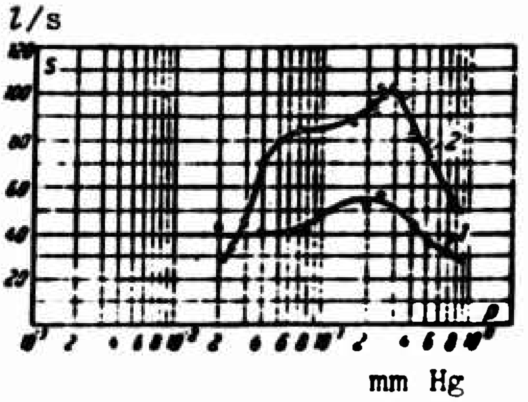


Fig. 5-7. Comparative characteristics of the stages of the booster pump. 1 - with a nozzle of the umbellate type; 2 - with a set of direct flow nozzles.

of direct flow nozzles with identical flow rates of steam, identical expansions of the nozzles, identical temperatures in the boiler and identical clearances between the nozzles and walls of the pump housing. As can be seen from Fig. 5-7, the stage with the direct flow nozzles has an operating speed twice that of the stage with the umbellate nozzle. For the operation of stages with direct flow nozzles the so-called "cowl" plays an important role, it is located under the stage. The experiment for developing the BN-4500 and BN-1500 pumps shows that without the cowl, the characteristics of the stage turns out to be worse than with the cowl (Fig. 5-8).

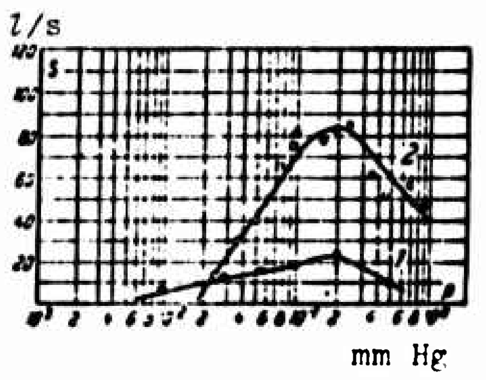


Fig. 5-8. Operating speed of stage with a set of direct flow nozzles depending upon the injection pressure. 1 - without a cowl; 2 - with a cowl.

Important also is the slope angle of the nozzles. Best results are obtained at an angle of the nozzles of 8-10° to the axis of steam pipeline.

To produce a high value of the peak outlet pressure an ejection stage is used as an output stage in many designs of booster pumps.

Figure 5-9 gives a diagram of a BN-2000 vapor oil booster pump with injection stage [167]. The injection stage of the pump is erected vertically and has a conical mixing chamber without a diffuser. Steam pipeline of the pump is three-stage, is located in the conical contracting housing. A boiler of such a type is just like that in a BN-4500 pump, but only with two pipes instead of six. The characteristic feature of the pump is rational organization of the steam feed of the separate stages. As for the operation of the injection stage it is necessary to maintain comparatively high vapor pressure in front of the nozzle, and in the boiler of the pump it is necessary to produce, accordingly, high vapor pressure. However, such a high vapor pressure is not required for the operation of the remaining stages; for example, for the first stage from the gas injection side the pressure of the vapor in front of the nozzle can be even a few times less than for the injection stage. Consequently, the supplying of vapor for all the stages at a particular (namely, a high) pressure, just as is done in many designs of pumps, only results in improperly expended vapor through the nozzles, and accordingly, results in unnecessary expenditures of power. In the BN-2000 pump the vapor feed for the separate stages is divided by means of installing diaphragms in the vapor feed pipe between the stages; in this case the vapor of high pressure from the boiler proceeds only so far as the injection stage, whereas the vapor is throttled in the diaphragms along the way to the remaining stages with a corresponding lowering of the pressure in front of each stage. Such organized vapor feed of the nozzle along with small thermal losses of the boiler ensures high technical and economic indices of the pump. The specific consumption of power based on liters per second for the operating speed of the BN-2000 pump amounts to 3 W/l/s, whereas for the BN-4500 pump it is equal to 5 W/l/s and for the BN-3 – 7 W/l/s.

Along with the proper distribution of vapor pressure by stages to increase the economy of the pump, the conical, contracting housing

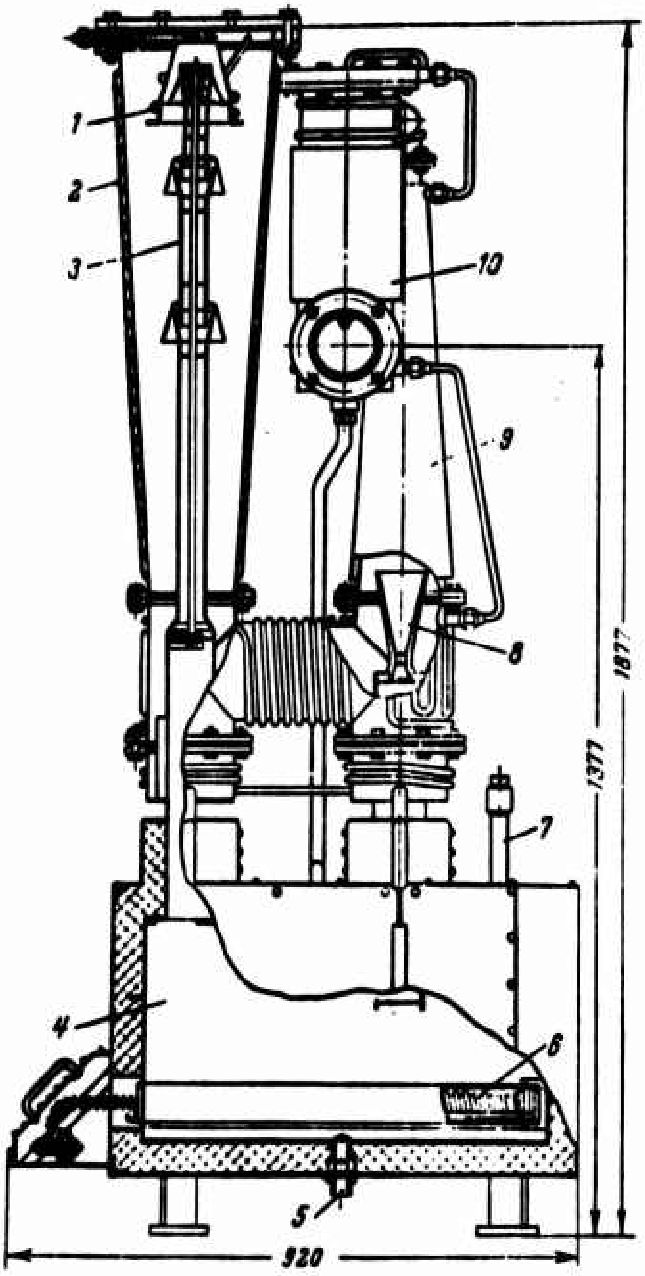


Fig. 5-9. BN-2000 vapor-oil booster pump. 1 - oil deflector; 2 - housing; 3 - steam pipeline; 4 - boiler; 5 - pipe connection for the overflow of oil; 6 - electric heater; 7 - filling tube for oil in the boiler; 8 - nozzle of the injection stage; 9 - effuser of the injection stage; 10 - trap for oil vapors.

also helps. In contrast to pumps with a cylindrical housing, where the diameter of the housing is determined by the magnitude of the operating speed of the first stage and is not changed during the transition to stages with a lower operating speed, in the pump with a lower operating speed is continuously reduced. In the first case due to larger diameter of the housing of the pump, the dimensions of the stages are consecutively increased from the first to the last, since their operating speeds, and accordingly, area of radial clearances between the nozzles and the housing consecutively decreases. Owing to the large dimensions of stages, the critical sections of nozzles in this case, and accordingly, the steam outputs turn out to be unjustified by overrating based on a purely design consideration, connected with the impracticability in large nozzles of small dimensions of critical sections. In case of conical housing its diameter and the dimensions of the stages consecutively decrease which permits the assurance of the required operating speed of the stages at the necessary (small) dimensions of the critical sections.

It is necessary, however, to consider that the height of the pump with a conical housing turns out to be considerably larger than the pump with a cylindrical body with identical characteristics (approximately by 1.5 times). In connection with this, in spite of the obvious economical advantage of pumps with conical housings, sometimes shorter cylindrical pumps are preferred.

With an increase of the output of booster pumps and their dimensions also increase accordingly, in particular, their height. Thus, for example, the BN-15000 pump with a conical housing with an output of 15,000 l/s, has an injection hole diameter of 900 mm and a height of 3.3 m. Realization of pumps of larger output, apparently, is already technically inexpedient owing to their extraordinarily great height, making their manufacture and exploitation difficult. With the necessity to have large output one should arrange a set of operating pumps in parallel.

For an appraisal of the height of a booster pump depending upon its operating speed we propose a semi-empirical relationship for the average distance between the two stages (in a cylindrical housing):

$$\bar{h}=0{,}25\,\mathrm{tg}\,\alpha\sqrt{S}\ [\text{мм}]. \qquad (5\text{-}1)$$

where $\bar{h} = H/n$ - ratio of full height between the first and last stages of a steam pipeline, to the number of stages; α - angle of inclination of the hood of the first nozzle, to the shear plane; S - operating speed of the pump, l/s. Equation (5-1) deviates from the experimental data by nearly 10%. By taking the corresponding angle of inclination of the hood of the nozzle (usually 65-70°) and selecting the necessary number of stages in the pump, it is possible, by using the equation (5-1), to determine the height of steam pipeline of pump H for a given operating speed. By adding the heights of the boiler, of the first nozzle, of the injection hole, of the oil seal, to the obtained value which in sum (for pumps with an output of more than 1500 l/s), amounts to approximately 20-30% of the H, we will obtain the full height of the booster pump with a cylindrical housing.

Thus, for a pump with an operating speed of 50,000 l/s and an angle of inclination of the hood of the first nozzle, $\alpha = 65°$, with a five-stage steam pipeline, the full height of the steam pipeline, $H = \bar{h}n$, will amount to $H = 6000$ mm in accordance with (5-1). By adding the unaccounted for heights (20% H), we will obtain the full height of the pump: $H = 7200$ mm. It is clear that the manufacture and use of a pump of such dimensions would turn out to be very difficult.

Vapor-oil booster pumps (housings, boilers) are usually made from common carbon steel, and the components of steam pipelines of the pumps - from aluminum. The cooling system for small pumps is made in the form of a coil, soldered on the housing, but for large pumps - in the form of a jacket, welded to the housing.

Two types of electric heaters for pumps are made: those with an open coil, placed in a ceramic base or wound on a ceramic pipe, and those with a closed coil, embedded in ceramics. Heaters of the first type have a coil with a short service-life (300-500 h), but can be easily repaired when burned out. Heaters of the second type have a long service-life (more than 3000 h), since access of oxygen to the coil is limited; however, in case of a burnout of the heater, its repair is impossible. At present in domestic pumps heaters with an open coil are widely used, in foreign pumps - heaters with a closed coil.

5-2. Working Fluids

Since booster pumps are designed for pumping out large quantities of gas at relatively high injection pressures (10^{-1}-10^{-4} mm Hg) and at considerable counterpressures, then the basic requirements imposed on the working fluids for booster pumps are: high thermal and thermo-oxidizing stability; high vapor pressure at the operating temperature in the boiler; low value of heat of vaporization; possibly a narrower fractional composition, excluding the noticeable change of characteristics of the oil due to the removal of light fractions from it during the operation of the pump. A very low vapor pressure at room temperature for the booster working fluid is not required. A vapor pressure at 20°C on the order of 10^{-5} mm Hg is sufficient.

Mainly special vacuum oils are used as working fluids for booster pumps. In domestic pumps three types of liquids are used: oil of petroleum origin, "G" and VM-3, and the silicon-organic compound - PFMS-1.

Oil "G" constitutes a product of distillation of medical vaseline oil (limits of distillation are 120-160°C at a pressure of 10^{-2} mm Hg). It is comparatively cheap. It possesses low vapor pressure at 20°C. Since during the purification of the initial product - medical vaseline oil - stabilizing resins are removed from it, oil "G" possesses low thermal and thermo-oxidizing stability (resistance to oxidation by the oxygen of the air at operating temperatures in the pump). During oxidation the oil darkens and changes its characteristics; resinous deposits are precipitated on the internal components of the pump. In connection with this the length of service-life of the oil in the pumps of large output is short.

Oil VM-3 [168-170] constitutes a product of the distillation of unrefined spindle oils, obtained from Caucasus and eastern oils. Oil VM-3 is almost 50 times more thermo-oxidizing resistant than oil "G." The operating characteristics of booster pumps when operating on oil VM-3 are ensured at a temperature of the oil of approximately 20-30°C lower than when operating on oil "G." At the same time, oil VM-3 has a lower boiling point, compared to "G," and therefore, the loss of oil from the pump when operating on VM-3 oil are higher than when operating on "G" oil (under identical conditions). Therefore, when operating on VM-3 oil one should use "more rugged" oil-catching devices.

Oil PFMS-1 consists of a polyphenylmethylsiloxane liquid. Just like all silicon-organic liquids, PFMS-1 oil possesses high thermo-oxidizing resistance, the highest compared to "G" and VM-3 oils. In connection with this the service-life of PFMS-1 oil in the pump turns out to be so long that the period of use of the pump is hardly limited at all during its trouble-free operation.

The characteristics of vacuum oils for booster pumps are given in Table 5-1.

Table 5-1.

Designation of the oil	Vapor pressure at 20°C, mm Hg	Viscosity at 50°C, cm	Molecular weight	Specific weight, g/cm^3
"G"	$5\cdot10^{[illegible]}$—$1\cdot10^{[illegible]}$	12,5—15,3	350	0,85
VM-3	$1\cdot10^{-4}$—$1\cdot10^{[illegible]}$	7—10	—	—
PFMS-1	$1\cdot10^{[illegible]}$—$7\cdot10^{[illegible]}$	3,6—4,6	700	1,0

The working fluids, used in foreign booster pumps, are: narkoil-10 (the United States); liquid "A" (England), liquid L-50 (FRG); they constitute a chlorinated diphenyl type of compound. These liquids possess high thermal and thermo-oxidizing resistance. Vapor pressure at room temperature is $\sim 10^{-4}$ mm Hg. The deficiency of the liquids is their toxicity (they act on skin and in respiratory tracts).

5-3. Characteristics

As already indicated, the dependence of the operating speed of the booster pump on the injection pressure has the form of curve with the maximum within the operating range of pressures. With an increase in the input power the maximum shifts in the region of high pressures, but with a decrease in power - in the region of low pressures. In this sense the dependence, shown in Fig. 5-1 for the BN-3 pump, with respect to its own character, is common for booster pumps and the output-injection pressure can be represented by coordinates just as shown in Fig. 5-10. With a reduction in input power the density of the jet decreases; accordingly, the diffusion of gas in the jet increases; this results in an increase of output in the region of low pressures and to a decrease of output generally in the region of high pressures.

Fig. 5-10. Dependence of output of a stage of a booster pump on the injection pressure at various temperatures in the boiler, ($T_1 < T_2 < T_3$).

An analogous dependence is obtained during the change in expansion of the nozzle (Fig. 5-11). During the expansion of the nozzle output is increased in the region of low pressures and output decreases in the region of high pressures.

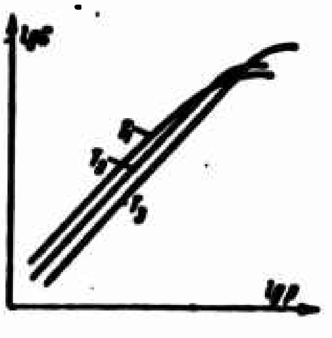


Fig. 5-11. Dependence of output of a stage of a booster pump on the injection pressure at various expansions of the nozzle ($A_1 > A_2 > A_3$).

The peak outlet pressure of the pump (or stage) also essentially depends on the degree of preheating and expansion of the nozzle being increased with the increase in the degree of preheating (Fig. 5-12) and with the decrease in the expansion of the nozzle (Fig. 5-13)

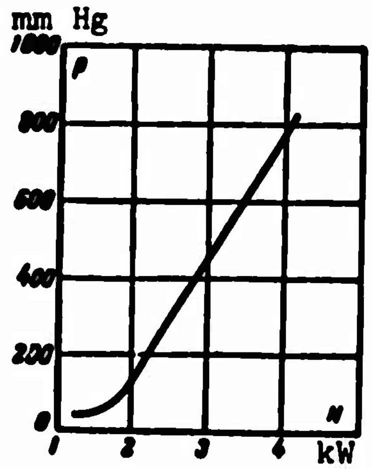


Fig. 5-12. Dependence of the peak outlet pressure of the booster pump on the degree of preheating (BN-3 pump).

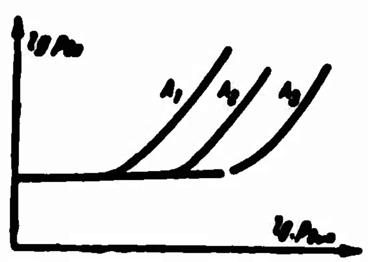


Fig. 5-13. Dependence of the injection pressure of the stage of the booster pump on the outlet of nozzle at various expansions ($A_1 > A_2 > A_3$).

By analyzing the dependences presented in Figs. 5-10 and 5-12, a conclusion can be drawn that the lowering of the output in the region of high pressures along with a decrease in the input power, and accordingly, with a decrease in the temperature and pressure of the vapor in the boiler occur owing to the decrease in the value of the peak outlet pressure of stage lower than the actual pressure, apart from the previous solution. If however, one were to maintain a counterpressure behind the stage sufficiently low, in such a manner that with a decrease in temperature and vapor pressure in the boiler the value of the peak outlet pressure of the stage would remain higher than the actual counterpressure behind it, then one could expect that the output in the region of high pressures would not decrease. These dependences confirm the above expressed consideration about the fact that it is not at all obligatory to have high vapor pressure in front of the first stage for its satisfactory operation; vapor pressure in front of the stage can be choked to a sufficiently small value, if in this case low counterpressure behind the stage is ensured which can be done by installing a sufficient number of stages in the pump.

Dependence of the operating speed of the stage on the pressure in the direction of preliminary rarefaction has the form presented in Fig. 5-14. As can be seen, with an increase in counterpressure behind the stage the operating speed at first changes, then after reaching the peak outlet pressure of the stage, it starts to decrease rapidly.

If, by holding the expansion of the nozzle and steam temperature in front of the nozzle constant, one changes the steam consumption in the stage due to the increase in the critical section of the nozzle, then when a constant area of radial clearance between the nozzle and the housing of the pump, the characteristics of the stage change in accordance with the dependence presented in Fig. 5-15.

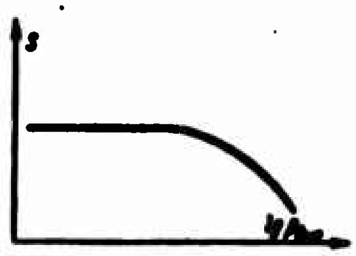


Fig. 5-14.

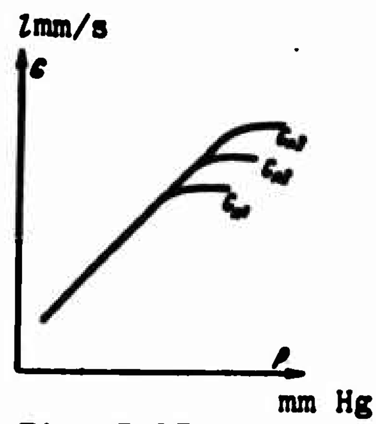


Fig. 5-15.

Fig. 5-14. Dependence of operating speed of the stage of the booster pump on the outlet pressure.

Fig. 5-15. Dependence of the output of the stage of the pump on the injection pressure at various flow rates of vapor through the nozzle (A = const; T = const; $G_{n1} < G_{n2} < G_{n3}$).

Due to considerable density of steam jets, passing from the nozzles of the booster pumps, they pump light gases well. Figure 5-16 gives the dependence of the operating speed on the injection pressure during the pumping out of air and hydrogen by the vapor-oil booster pump [166].

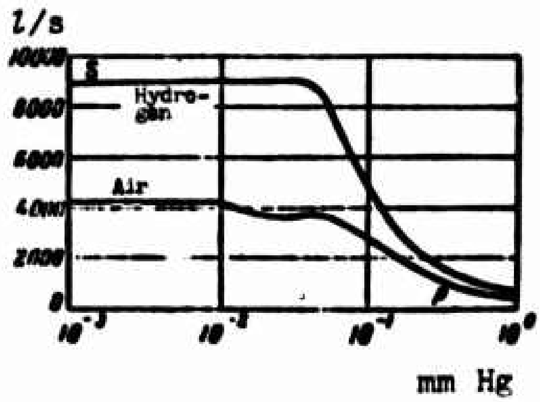


Fig. 5-16. Dependence of the operating speed of the booster pump on the injection pressure when pumping out air and hydrogen.

The magnitude of the operating speed of the pump essentially depends on the number of stages in it. This is governed by the fact that with a given diameter of the housing of the pump, the increase in the area of radial clearance between the nozzle and wall of the housing due to the reduction in the diameter of the nozzle leads simultaneously to a reduction in the peak outlet pressure of the stage. By increasing the number of stages in the pump and compensating thereby for the lowering of the compression ratio in the stage with an increase in the area of clearance between the nozzle and the housing body, it is possible to increase the operating speed of the pump. Naturally, in this case the height of the pump increases substantially. Thus, in an experimental booster pump with a diameter of the housing of 260 mm it was possible to increase the operating speed from 1500 to 3500 l/s due to the addition of two stages and a corresponding selection of operating conditions of the pump. In this case the height of the pump increased by almost 1.5 times. Similar results were also obtained according to Barrett and Dennis [166] with an addition of only one stage to a 18V3 pump (England).

In spite of the possibility of a considerable increase in operating speed of the pump with a corresponding increase in its height, in most cases it is preferred not to make pumps too high, and the increase in output can be accomplished due to the increase in the diameter of the housing of the pump. Thus, for example, the above-indicated increase in the operating speed from 1500 to 3500 *l*/s could be achieved due to the increase in the diameter of the housing of the pump from 260 to 380 mm with hardly any increase in the height of the pump.

The majority of contemporary booster pumps has a maximum output at a pressure of 10^{-2} mm Hg, and a peak outlet of 1-3 mm Hg.

The critical vacuum, measured by an ionization gauge, graduated by air, amounts to $(1\text{-}5)\cdot10^{-4}$ mm Hg for the majority of pumps. In this case the true value of the residual pressure is approximately of an order lower and turns out to be overrated by measurement owing to the increased sensitivity of the ionization gauge to oil vapors.

The thermodynamic efficiency of modern booster pumps amounts to $5\cdot10^{-4}$-$5\cdot10^{-3}$.

The characteristics of domestic booster pumps are shown in Fig. 5-17 and in Appendix 11.

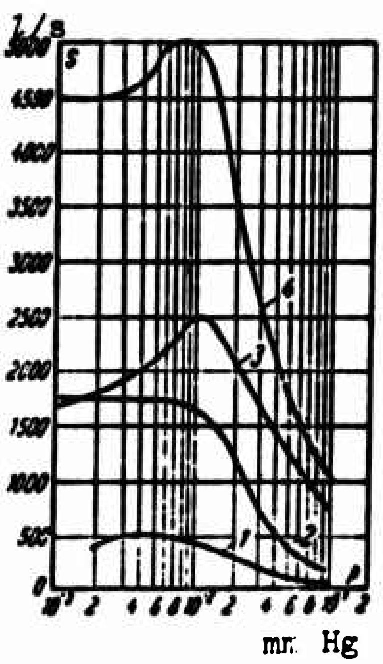


Fig. 5-17. Characteristics of vapor oil booster pumps. 1 - BN-3 pump; 2 - BN-1500 pump; 3 - BN-2000 pump; 4 - BN-4500 pump.

5-4. Migration[1] of Oil Vapor from the Booster Pump

During the operation of the booster pump migration of the oil vapor from the pump into the pumped volume and into the region of the preliminary rarefaction (outlet pipeline) takes place.

As already indicated in Section 3-3, the steam jet, passing from the supersonic nozzle of the vacuum steam-jet pump, also has a flow line of vapor, directed towards the injection pipe. The source of lubrication for the pumped system are, namely, these vapor flow lines.

To prevent the migration of oil in a pumped system a water-cooled copper oil deflector of the hood type is placed above the pump (see Figs. 5-6 and 5-9). Diagram of the action of the reflector is shown in Fig. 5-18. The action of the reflector is based on the "shearing" vapor flow lines such that the extreme flow line falls on the cold walls of the housing of the pump; in this case sufficiently complete condensation of vapor, migrating from the pump is ensured. Thus, during the normal operation of the BN-4500 pump with a hood reflector the presence of oil in the pumped system after 10 h of work is not visually disclosed. According to [166] the installation of a hood-type reflector above the booster pump lowers the migration of oil in the pumped system by almost 95%, so that the reverse flow of oil vapor behind the reflector (in a 18V3 pump) amounts to approximately 0.03 mg/min·cm^2 of area of the injection hole of the pump.

It is necessary to consider that the oil deflector works satisfactorily only within the operating range of the booster pump (for example, to 10^{-2} mm Hg for a BN-4500). At higher pressures the steam jet, passing from the nozzle, is broken up by the flow

[1]The term "migration" here conditionally signifies the penetration of oil vapor from the pump into the pumped system and into the initial vacuum pipeline.

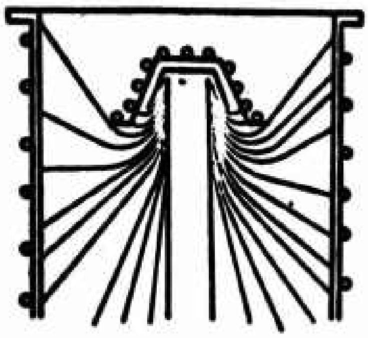


Fig. 5-18. Diagram of the action of a hood oil deflector.

of air, and the vapor starts to penetrate into the pumped system. Therefore, when using the pump it is necessary to keep watch so that the pressure in the system does not exceed the limits of the operating range of pressures of the pump.

An oil deflector of the hood type reduces the rate of pumping of the booster pump by only 5-10%.

In order to prevent the migration of oil from the booster pump in the region of preliminary rarefaction behind the jet stage of the pump a water-cooled disk condenser was installed (Fig. 5-19); it consists of a pack of alternated copper disks, blind and with holes, assembled on a water-cooled rod and placed in a cooled housing. The spout of oil through such a condenser during the operation of BN-2000 and BN-4500 pumps in the operating range of pressures does not exceed 0.4-0.6 cm^3/h (when operating on "G" oil).

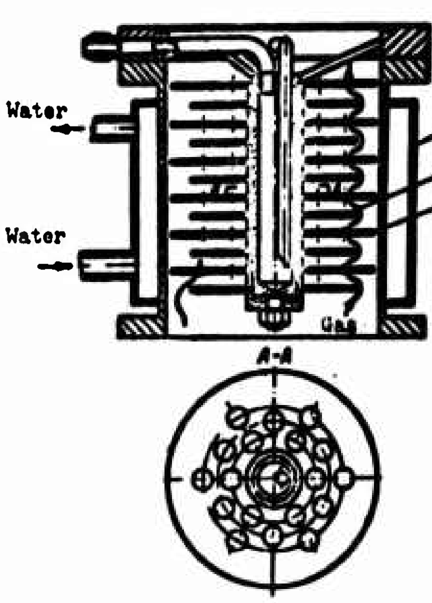


Fig. 5-19. Diagram of the disk condenser device for preventing the migration of oil vapor into the region of preliminary rarefaction. 1 - housing; 2 - blind disk; 3 - disk with holes.

5-5. Use

Vapor-oil booster pumps based on the range of operating pressures occupy an intermediate position between mechanical rotary pumps and high-vacuum steam-jet pumps. Therefore, they are used for pumping out installations, in which a considerable emanation of gases takes place, accompanied by an increase in pressure up to 10^{-1}–1 mm Hg, while at the same time the production of low pressures of 10^{-2}–10^{-4} mm Hg is required.

Thus, vapor-oil booster pumps are widely used for pumping out vacuum induction and arc metallurgic furnaces, installations for drying and impregnating electrical condensers, vacuum distillation installations, installations for metallizing plastic articles, supersonic wind tunnels, etc.

When pumping out heavily dust-laden gases the installation of filters is recommended in front of the booster pumps. Although there is settling of dust in the pump and although it does not have an effect directly on its operation, nevertheless, the accumulation of dust in the boiler over the course of time can result in an impairment of conditions of heat transfer in it and result in a change of the operating conditions of the pump. The installation of a filter usually results in a 50% reduction of the operating speed of the pump in air.

The wide use of booster pumps is caused, to a large degree, by the simplicity of the device and its operation.

Since the booster pumps operate in a set with auxiliary mechanical pumps, then economic assessment of the use of the booster pumps in comparison with other types of pumps should be made taking into account the total complexity of equipment included in the pumping installation: auxiliary pumps, valves, locks, etc. Figures 5-20 and 5-21 give the relative cost and power drain per 1 l/s of the operating speed of the pumping installation depending upon the injection pressure for the systems with the booster

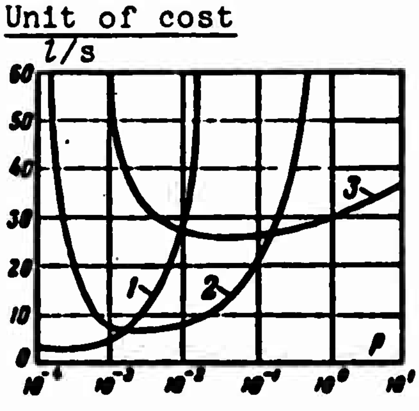


Fig. 5-20.

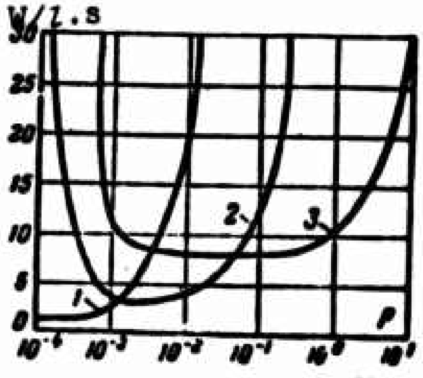


Fig. 5-21.

Fig. 5-20. Relative cost of pumping installations on the basis of a booster pump (curve 2), high-vacuum steam-jet pump (curve 1) and a double-rotary pump (curve 3), referred to in 1 l/s operating speed of the installation, depending upon the injection pressure.

Fig. 5-21. Power, consumed by pumping installations on the basis of a vapor-oil booster pump (curve 2), high-vacuum steam-jet pump (curve 1) and a double-rotary pump (curve 3), referred to in 1 l/s of operating speed of the installation, depending upon the injection pressure.

vapor-oil pump, high-vacuum steam-jet pump and a double-rotary pump (mechanical booster pump) [166]. As can be seen from the given curves, in the range of pressures of 10^{-3}-10^{-1} mm Hg the installation on the basis of vapor-oil booster pump consumes less power and has a lower cost in comparison with other installations. In the range of pressures below 10^{-3} mm Hg the system with the high-vacuum steam-jet pump turns out to be more economical and in the range of pressures above 10^{-1} mm Hg - the system with the double-rotary pump.

Just as in all steam-jet pumps in which moving parts are lacking, booster pumps are hardly subject to wear. The period of continuous operation of vapor-oil booster pumps is limited mainly by the service-life of the oil in the pump and by the service-life of the electric heaters. The service-life of the oil depends on the operating conditions of the pump and the kind of oil. As already indicated, oil "G" has the shortest service-life - due to its low thermo-oxidizing resistance, the longest - PFMS-1 oil. The higher the level of pressures at which the pump operates, the shorter the service-life of the oil. In connection with this, "G" oil is not recommended for use in large pumps, operating at high pressures.

When using vapor-oil booster pumps one should not permit an increase in the injection pressure in the pump of more than its maximum operating pressure (usually 10^{-2}-10^{-1} mm Hg) since otherwise the migration of oil from the pump in the pumped system and in the region of preliminary rarefaction strongly increases. Instances of thorough lubrication of the pumped volume taking place in the operation are, as a rule, the result of the prolonged operation of the pump on the transfer section of the characteristics [higher than the maximum operating point on curve $G = f(p)$]. During the normal operation of the pump the migration of oil supplied by the oil deflector upon injection and by the condenser at the outlet is insignificant and usually does not determine the period of continuous work of the pump.

During the pumping out of condensed vapors by booster pumps usually gas ballast pumps are used as high-pressure vacuum pumps. In this case it is necessary to watch that the pressure behind the booster pump will not increase above its peak outlet pressure, since this can lead to a breakdown of the operation of the pump. In a case where the peak outlet pressure of the booster pump is small (for example, 0.3-0.5 mm Hg for the BN-3), sometimes it is generally not possible to carry out the pumping out of condensed vapors, since in this case the injection pressure in the gas ballast pump during a full steam load can increase to 1 mm Hg, i.e., it becomes higher than the peak outlet pressure of the booster pump.

In order to cool the booster pumps well purified water with a low content of salts and absence of mechanical impurities should be used. The use of poorly filtered water leads to an obstruction in the water jacket of the pump and puts the pump out of commission. To clean the dirty water jacket with sand or with deposits of salts usually does not work; it is necessary in this case to cut away the water jacket and to weld on a new one. The cooling of the booster pumps is usually calculated on the basis of free pressureless overflow. Therefore, during the operation of the pump its water jacket is filled by water usually only in the course of its operation. During prolonged stops of the pump intense corrosion of the walls in the water jacket, unfilled with water can go on resulting in the clogging up of the channels of the jacket by oxidized iron, and sometimes even the perforation of walls of the housing of the pump. In order to prevent corrosion in the water jacket one should install a valve on the overflow water line and when stopping the pump, shut it off, leaving the water jacket filled with water.

Booster pumps should be connected to the pumped system through the shortest possible pipes. The diameter of the connecting pipe is usually taken to equal the diameter of the injection hole of the booster pump. The dimensions of the connecting pipe are selected with such a design as to ensure the minimum losses of operating speed of the pump (see design of systems, Chapter 9).

Between the pump and the pumped system the installation of a vacuum lock is recommended, allowing one to disconnect the system from the pump at the time of its starting and stopping, since during these periods an increased migration of oil vapors from the pump into the pumped system takes place due to the unformed steam flows in the stages of the pump. Furthermore, the presence of a lock makes it possible to protect the pump in the event of air accidentally getting in the pumped system.

The outlet pipeline, connecting the booster pump with the mechanical pump, should also be as short as possible. Its diameter is usually taken to equal the diameter of the outlet booster pump. The dimensions of the pipeline should be selected with such a design that with the selected mechanical pump one ensures obtaining a pressure behind the booster pump during its peak output smaller than the peak outlet pressure of the booster pump.

One should install a vacuum valve on the outlet pipeline, allowing one to disconnect the booster pump from the mechanical one during a sudden stop of the latter.

The starting of the booster pump usually takes 0.5-1 h. In connection with this, to avoid loss of time on starting the pump during frequent exhaust cycles of a commercial installation, the booster pump usually is not turned off, but the valve between the pump and the installation is merely closed, leaving the booster pump in the operating state.

During the normal cooling of the booster pump, the proper operation of the electric heater, absence of flow in the pump and proper operation of the auxiliary mechanical pump, the impairment of the characteristics of the booster pump can occur only due to the impairment of the quality of oil or a reduction of oil in the pump. Therefore, in safeguarding the indicated conditions, the elimination of malfunctions in operation of the booster pump when in actual use usually results in either an addition of the quantity of oil in the boiler, or to its replacement. The prophylaxis of the booster pumps are to be expected just as the prophylaxis of vapor oil jet pumps (see Section 4-4).

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13. ABSTRACT

The book contains generalized and systematized data on the development, investigations of operation of steam-jet vacuum pumps. Given are contemporary concepts on the operating mechanism of pumps and the latest achievements in the field of pump design. Special attention was given to the description of various physical phenomena taking place in pumps, characteristics of their operation, calculation and design of pumps. The book was written for a wide circle of engineers and scientists engaged in operation and development of steam-jet vacuum pumps; it can be useful also to college students studying vacuum techniques.

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Vacuum Technique						
Thermodynamic Calculation						
Vapor Pressure						
Vacuum Oil						
Booster Pump						
Vapor Pressure						
Vaporization						