rotor with longitudinal baffles and V-shaped baffles; the power consumed in the case of rotors with scrapers is somewhat higher, however, in the case of rotors with disc elements it is significantly lower.

In conclusion it must be stated that the results obtained in the present work have been utilized in the designing of industrial horizontal reactors for the synthesis of polyesters and polyamides.

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EXAMINATION OF TWO-PHASE REGIMES OF CRYOGENIC TURBINE EXPANSION ENGINES

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UDC 621.592.3.001.5

The two-phase regimes of cryogenic turbine expansion engines have been examined at a liquefaction degree of the working substance (air) of up to 6% [1-3]. However, considerably higher liquefaction degrees can be obtained in equipment using turbine expansion engines in two-phase regimes. For this reason, the N. E. Bauman Higher Technical School, Moscow, tested turbine expansion engines in two-phase regimes with liquefaction degrees considerably higher than those examined in [1-3], evaluated the structure of the flow at the outlet from the rotor, and determined the effect of condensation of a large part of the working substance on the performance and efficiency of a peripheral-admission turbine engine.

The experiments were conducted in air dried in a zeolite unit to remove carbon dioxide. Each of the five tested turbine engines represent a peripheral-admission stage with the radial-axial semiopen rotor, and the brake is represented by a centrifugal heater and the supports by gas-static bearings.

Main Parameters of Turbine Expansion Engines

	M28A	M28V	M28S	M28D	M500
Outer diameter of the rotor, mm	28	28	28	28	12
Reduced diameter of the rotor at the outlet	0.4	0.4	0.4	0.4	0.4
Width, mm:					
of rotor blades at the inlet b_r	1.5	1.5	1.1	1.1	0.5
nozzles of stator b_s	0.85	0.49	0.49	0.75	0.2
Nominal width of nozzles $b_{n.n.}$, mm	1.4	1.4	1	1	0.4
Height of stator nozzles, mm	0.7	0.7	0.7	0.7	0.4
Total passage cross section of the nozzles mm^2	14.3	8.2	8.2	12.6	0.8
Limiting frequency of rotation of the rotor, sec^{-1}	3650	3650	3650	3650	7500

The M28 turbine expansion engines were tested at low and medium pressures, and the M500 was tested at a higher pressure.

The structure of the two-phase flow is characterized by the dispersion of the liquid phase. The presence of a coarse-dispersion liquid in the flow behind the rotor may be detected visually from the presence of film flow at the walls of the outlet nozzle. The effect of condensation of the working substance on the efficiency of the turbine expansion engine was determined by comparing the efficiency of the machine in the two-phase and gas cold regimes (the expansion process is completed prior to reaching the saturated vapor line in the phase diagram) with the same pressures at the inlet and outlet of the turbine expansion engines and the reduced circumferential velocity at the inlet into the rotor $\bar{u}_1$. The efficiency

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TABLE 1

Turbine expansion engine	Relative width of the nozzles	Efficiency	
		in gas cold regime $\eta_{sg.c}$	in two-phase regime η^*_{st}
M28V	0.35	0.56	0.16
M28S	0.49	0.617	0.492
M28D	0.75	0.645	0.603

*The temperature at the inlet into the turbine expansion engine was T_0 = 120-123°K.
Comment. For all the regimes, the pressure at the inlet into the turbine expansion engine p_0 = 2 MPa, velocity $\bar{u}_1$ = 0.6-0.65.

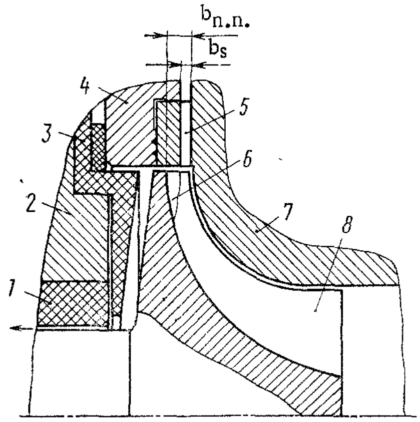

Fig. 1. Fragment of the longitudinal section of the M28 turbine expansion engine: 1) labyrinth seal; 2) housing of the bearing; 3) heat insulating vessel; 4) housing of the starter; 6) dead zone; 7) the stationary cover jaw; 8) rotor.

of the machines in the two-phase regimes was determined on the basis of the nature of their operation and the results of subsequent examination of the flow-through part.

The tests were carried out in a special testing unit [4, 5] which made it possible to regulate and measure the pressure and temperature of the flow at the inlet and outlet of the machine, the speed of rotation of the rotor, the flow rate of the working body via the turbine expansion engine, and the power of the electric heater at the outlet from the machine. The heater evaporated the liquid phase of the flow to determine the efficiency of the turbine expansion engine in the two-phase regimes [4]. The rms error in measuring the flow rate did not exceed 1.1%, in measuring the isoentropic efficiency η_s 6.7%, in measuring the degree of liquefaction g 2.9%, and in determining the reactivity ratio of the machine ρ_m 0.46%.

The M28A machine is constructed using a replaceable casing of special design which makes it possible to examine visually the nature of the flow in the outlet nozzle over a length of 150 mm, starting almost at the rotor. The area is illuminated using eight low-power bulbs uniformly distributed along the length of the optically transparent slit. To prevent blurring of the surface of the inspection glass due to the condensation of steam and CO_2, cold air from which CO_2 was removed was blown onto the glass. Visual examination of the flows in the two-phase and gas cold regimes shows that in the two-phase regime, the liquid forms in the flow-through part of the turbine expansion engine. The absence of the film flow on the walls of the outlet nozzle indicates that no large droplets form during spontaneous condensation in the turbine expansion engine.

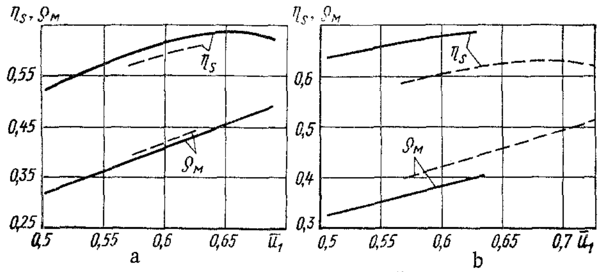

Fig. 2. Operating characteristics of turbine expansion engines: a) M28A (p_0 = 0.7-1.1 MPa; π = 0.155-0.186); b) M28D (p_0 = 2.2-2.5 MPa; π = 0.12-0.123); ——, gas cold regimes; ----, in two phase regimes.

The tested M28 machines differ from each other in the parameters b_r, $b_{n.n.}$, b_s, and $\bar{b} = b_s/b_{n.n.}$, and parameter $\bar{b}$ has a stronger effect on the efficiency of the turbine expansion engine in the two-phase regime than in the gas cold regime for the same degree of expansion π (see Table 1).

The large reduction of the efficiency of the turbine expansion engine with the lower value of $\bar{b}$ in the two-phase regime is caused by the fact that the size of the dead zone, i.e., the zone not filled with the flow (Fig. 1), in the rotor increases with decreasing $\bar{b}$. The liquid particles present in the flow at the outlet from the starter which formed during spontaneous condensation in the rotor and penetrated into the dead zone are ejected under the effect of centrifugal forces into the gap between the housing of the starter and the rotor. Part of the liquid phase separated in this manner is evaporated as a result of contact with the components of the structure, and part is carried away with gas leaks through the external labyrinth seal of the shaft into the cavity for collecting the gas from the bearings. The above-mentioned reason for the reduction of efficiency is confirmed by the fact that in the two-phase regime, the gap between the power at the shaft of the turbine expansion engine and its refrigerating capacity increased with decreasing $\bar{b}$:

Turbine expansion engine	M28V	M28D
Power at the shaft N_e, W	568	2460
Refrigerating capacity Q_0, W	190	2350

It is evident that the increase of the N_e-Q_0 difference is the result of the increase in the cold losses caused by the separation of the liquid phase from the dead zone. The loss of the liquid with the external leaks is also indicated by freezing of the main line for the collection of gas from the bearings. No freezing was observed in the corresponding gas cold regimes.

The factors influencing the efficiency of the turbine expansion engine in the two-phase regime also include the moisture content of the flow in the rotor (in its initial sections). A reduction of the moisture content, e.g., by increasing the temperature of the gas at the inlet into the machine at constant pressure and other conditions being equal, increases the efficiency as a result of a lower degree of separation and removal of the liquid phase from the dead zone.

The operating characteristics of the M28A and M28D turbine expansion engines are shown in Fig. 2 which indicates that the dependences of η_s on $\bar{u}_1$ in the gas cold and two-phase regimes are virtually equidistant for each machine. The difference in $\eta_{s.g.c.}$ and η_{st}, equaling 2 and 6% respectively, is explained by an increase in the heat inputs in the transition from the gas cold to two-phase regime. The degree of liquifaction of air in these turbine expansion engines reached respectively 8 and 14%.

The efficiency of the M500 turbine expansion engine in the two-phase regimes remained almost identical with the efficiency level of the gas cold regimes, regardless of the low value $\bar{b}$ = 0.5 and the high degree of liquefaction g = 13%. The absence in this machine of the cold losses associated with the removal of the liquid phase is explained by the fact that in the tests a large part of the heat gradient was used up in the gas region (the parameters of the working body at the inlet greatly differed from those in the two-phase region in equilibrium diagram).

All the machines operated in the two-phase regimes in a stable manner with small variations (amplitude 0.01 MPa) of the static pressure at the inlet and outlet of the rotor. The pressure variations were caused by the fact that the supply of latent heat in condensation in the stator destabilizes the operating regime of the nozzles and the flow (mass flow rate) pulsates.

During the tests of the M28 turbine expansion engines (∿180 h), no visible traces of erosion of the blades of the stator and the rotor made of respectively brass and AK-6 aluminum alloy without any coating were found.

The following conclusions may be drawn as a result of these tests: Condensation of air takes place in the flow-free part of the turbine expansion engine and the structure of the two-phase flow is finely disperse; the performance of the turbine expansion engines in the two-phase regime is not disrupted (the degree of liquefaction reached 20%); condensation of a large part of the working body in the machines with $\bar{b} \geqslant 0.75$ has almost no effect on their efficiency; at $\bar{b} < 0.75$, the efficiency of the turbine expansion engine in the two-phase regimes decreases to a considerably greater extent than in the gas cold regimes. This difference in the efficiency increases with an increase in the content of the liquid phase in the initial sections of the rotor and with a reduction of $\bar{b}$.

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EFFECT OF ENERGY LOSSES ALONG THE LENGTH OF THE FLOW ON THE DISTRIBUTION OF LIQUID IN EQUIPMENT WITH A DRIPPING FILM

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The current requirements of chemical engineering are satisfied to the largest extent by vaporizers with the dripping liquid film. These systems are characterized by a high intensity of heat transfer at low temperature heads combined with the short time during which the product stays in the zone of heating. Consequently, thermolabile substances can be processed. To achieve effective operation of these systems, it is necessary to ensure the uniform distribution of the sprinkled liquid both on the internal surface of the heating pipe and over the entire section of the pipe bundle.

Of the existing distribution devices, the distribution trays [1] are used most extensively. To ensure uniform sprinkling of the pipes, the level of the liquid on the tray must generate the hydraulic head H slightly greater than the specific value H_{min} sufficient to prevent vortices (whirls) which greatly reduce the throughput capacity of the discharge holes. The ideal distribution of the liquid is the one in which the constant level is ensured at all points of the tray. However, the liquid on the tray represents a flow with smoothly changing motion. In peripheral supply, the liquid flows towards the center and loses part

*Deceased.

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