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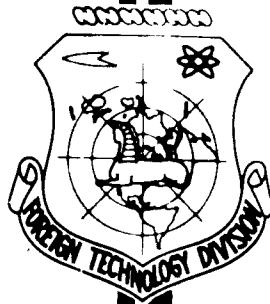
## TRANSLATION

THE PRODUCTION OF LIQUID OXYGEN USING AN AVERAGE PRESSURE  
SYSTEM WITH NITROGEN CIRCULATION

By

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et. al

FOREIGN TECHNOLOGY  
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AIR FORCE SYSTEMS COMMAND

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## UNEDITED ROUGH DRAFT TRANSLATION

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CIRCULATION.

BY: A. G. Golovintsov, M. B. Stolper,  
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THE PRODUCTION OF LIQUID OXYGEN USING AN AVERAGE-  
PRESSURE SYSTEM WITH NITROGEN CIRCULATION.\*

A. G. Golovintsov, Doctor, Professor of Technical Sciences, deceased; M. G. Stolper, Engineer; and A. M. Arkharov, Candidate of Technical Sciences.

Due to the growing need of different branches of technology for liquified gases, in particular oxygen, the question is raised concerning the development of optimum technological arrangements for the production of large quantities of liquid oxygen. The recent successful development of Soviet turbocompressor construction makes it possible to use compressors of average pressure  $p = 20 - 40$  atm for the setting up of large-scale deep-cooling installations for the separation of air involving the extraction of liquid products.

In the Deep-Cold Problem Laboratory at the Bauman Moscow Higher Technical School, research was conducted on cycles of deep cooling with nitrogen circulation and with the use of turbomachines. These

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\*The calculations used in this article were made by engineers L. Buchek, Z. Katz, and L. Rusanova under supervision of the authors.

cycles make possible complete separation of treated air in a single-column device without contamination of the oxygen by oil, do not demand special removal of carbon-dioxide and drying of the treated air, and have good economic indexes.

In this work we give the results of research on the basic characteristics of this cycle, and show the optimum values of the starting parameters.

Figure 1 presents the installation with a nitrogen circulation cycle at a pressure of 30 atm. The flow of air ( $1.08 \text{ m}^3/\text{m}^3\text{ta}$ )\* is compressed in turbocompressor 2 to a pressure of 6.5 atm and is directed into regenerators 1. Then the air is heated in heat exchanger-liquifier 5 from a temperature of  $101^\circ\text{K}$  to  $118^\circ\text{K}$  and, expanded in air turbine expansion engine 4 (turboexpander) from a pressure of 6 atm to a pressure of 1.4 atm, proceeds to fractionating column 3. Since the column is irrigated with liquid nitrogen, there is almost complete separation of the air, and 95-98% nitrogen emerges from the upper part of the column. The reverse flow of nitrogen, passing through the reflux supercooler 6 and oxygen supercooler 7, proceeds to heat exchanger-liquifier 5, where it divides into two flows. One flow,  $1.03 \text{ m}^3/\text{m}^3\text{ta}$  with a temperature of  $93^\circ\text{K}$  is directed into the regenerators, and the other flow,  $0.552 \text{ m}^3/\text{m}^3\text{ta}$  with a temperature of  $119^\circ\text{K}$  proceeds to basic nitrogen heat-exchanger 9.

The regenerators are kept from freezing by passing  $0.198 \text{ m}^3/\text{m}^3\text{ta}$  of circulation nitrogen through them together with the residual nitrogen. The circulation nitrogen is liquified in nitrogen turbocompressor

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\* $\text{m}^3\text{ta}$  is a cubic meter of treated air in the column under normal conditions.

11 to a pressure of 30 atm and is directed to preliminary nitrogen heat-exchanger 10, after which part amount of the nitrogen,  $A_{te} = 0.347 \text{ m}^3/\text{m}^3\text{ta}$  expands in complete expansion turboexpander 12 from a pressure of 29.5 to 1.4 atm. After the basic nitrogen heat-exchanger, part of the nitrogen  $A_{2e} = 0.538 \text{ m}^3/\text{m}^3\text{ta}$  is expanded in partial-expansion nitrogen turboexpander 8 from a pressure of 29 to 6.5 atm; the other remaining average-pressure nitrogen  $A_{30} = 0.212 \text{ m}^3/\text{m}^3\text{ta}$  is liquified in heat exchanger-liquifier 5 and is throttled. After throttling, the flows mix and are directed to the column condenser, from which the flow  $A_{cc} = 0.75 \text{ m}^3/\text{m}^3\text{ta}$  passes through reflux supercooler 6 and the throttle, and irrigates the upper part of the column. Liquid oxygen is withdrawn from the column evaporator through oxygen supercooler 7 to the consumer.

We should note that the given basic parameters of the system will vary depending on the pressure of the circulating nitrogen. With increasing pressure, the temperature of the nitrogen passing into turboexpanders 8 and 12 will increase; correspondingly, with decreasing pressure the temperature will decrease. Thus, bearing in mind that the flow pressure of technological air is constant, a redistribution of flows and introduction of supplementary heat-exchange devices are needed in the basic system.

However, for any pressure of the circulating nitrogen, it is essential to determine the optimum temperature of the nitrogen  $T_1, \text{K}$  ahead of the complete-expansion turboexpander and to determine the optimum amount of expander nitrogen  $A_{te}$ , expanding in the turboexpander. Only when these conditions are adhered to is reliable comparison of the energy characteristics of the cycle possible.

In subsequent calculations the following initial values were used: the loss of air in regenerators 8%; the temperature

difference at the hot end of the regenerators  $-5^{\circ}\text{C}$ ; the temperature difference at the hot end of the preliminary nitrogen heat-exchanger  $-5^{\circ}\text{C}$ ; the heat influx from the environment  $1.2 \text{ Kcal/m}^3\text{ta}$ ; the reflux ratio in the upper part of the rectification column is 0.45, which corresponds to the minimum, 4%, disparity with respect to nitrogen; the adiabatic efficiency of the air turboexpander 0.8; the adiabatic efficiency of the partial-expansion nitrogen turboexpander 0.75; the adiabatic efficiency of the complete-expansion nitrogen turboexpander\* of the first stage 0.75, of the second stage 0.78; the isothermal efficiency of the air turbocompressor of 0.58; the isothermal efficiency of the nitrogen circulation turbocompressor of the first stage 0.58, of the second stage 0.56; the temperature difference in the condenser-evaporator  $3.8^{\circ}\text{C}$ ; the concentration of oxygen 99.2%; the supercooling of oxygen  $5^{\circ}\text{C}$ ; the concentration of nitrogen 95%; the pressure of the treated air 6.5 atm.

The air pressure is selected to insure that the regenerators will not freeze. In all of the calculations the pressure decrease along the flow line was taken into account, beginning with a practical estimate of the hydraulic resistances of the devices. Calculations were carried out for one normal cubic meter of treated air. The temperature of the start of expansion of the air in the partial-expansion air and nitrogen turboexpander is determined by the necessity of attaining, for the discharge of gas from these machines, a state close to that of saturated vapor. The nitrogen temperature  $T_1$  at the entrance into the complete-expansion turboexpander can vary

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\*In this case the arbitrary adiabatic efficiency of the drop turboexpander, equal to the ratio of the actual heat drop to a complete adiabatic heat, is 0.81.

within a wide range.\* An increase of this temperature results in increased heat drop in the expander and decreases somewhat the specific power consumption in the production of the liquid product. However, during this the temperature differences in the heat-exchangers decrease, and their dimensions significantly increase. With extreme decrease of the observed temperature, in connection to the increased amount of turboexpander nitrogen the dimensions of the heat-exchangers also increase.

When determining of optimum temperature  $T_1$ , the temperature its values given and the following magnitudes determined:  $A_{ta}$  is the amount of expander air;  $N_e^*$  is the specific power consumption taking into account the energy recovered by the turboexpanders; and the factor  $\frac{q}{\Delta t_{pb}}$  is for the preliminary and basic nitrogen heat-exchangers (here  $q$  is the thermal load;  $\Delta t_{pb}$  is the average logarithmic temperature difference). The factor  $\frac{q}{\Delta t_{pb}}$  characterizes the relative values of the heat-exchanger surfaces, and consequently the dimensions and weight of the heat-exchangers. Increasing the size of the heat-exchanger, apart from raising the cost, results in a certain power-consumption increase because of increased resistance in the reverse flow. Considering these details, we considered that temperature ahead of the exchanger for which the product  $N_e \frac{q}{\Delta t_{pb}}$  will be minimum is the optimum temperature.

The results of the calculations are given graphically in Fig. 2.

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\*A complete-expansion average-pressure turboexpander was developed and tested at the Deep Cold Laboratory of the Bauman Moscow Higher Technical School under the supervision of Engineer M. S. Babichev. During the expansion in one stage from 30 to 1.5 atm and at 90,000 revolutions per minute, the adiabatic efficiency of the turboexpander was 65%.

Similar calculations to determine the optimum values of  $T_1$  were carried out for the other values of the pressure of the circulating nitrogen (Fig. 3). As a result, conditions were found for which the characteristics of the examined cycle could be compared.

The cycle with circulating nitrogen was calculated for pressures of circulating nitrogen of 7, 15, 30, and 200 atm and for the above-given efficiency values of the machines and devices. The magnitude of the conditional adiabatic efficiency of the piston expander in a high-pressure cycle ( $p = 200$  atm) is 0.75. As is obvious from Fig. 4, the discontinuity in the basic characteristics of the cycles practically disappear at pressures higher than 30 atm. This is explained, on the one hand, by the fact that the adiabatic efficiency of the complete-expansion turboexpander is 0.81 with stage 1 efficiencies of 0.75 and 0.78, whereas the conditional adiabatic efficiency of the piston expander is 0.75; on the other hand, the magnitudes of the adiabatic heat drop and optimum temperature change but slightly at pressures greater than 30 atm, as can be seen in Fig. 3. Moreover, when calculating the high-pressure circulation cycle the final expansion pressure in the piston expander is 3.5 atm.\* Consequently, the power characteristics of the cycles at a pressure higher than 30 atm differ insignificantly the power consumption per 1 kg of liquid oxygen in the high-pressure cycle ( $p = 200$  atm) is about 1.17 - 1.18 kw·hr while in the average pressure cycle ( $p = 30$  atm) it is about 1.2 kw·hr.

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\*Expansion to a lower pressure in the piston expander is difficult because of the great increase in its size. Moreover, in this case for protection against oil seepage there must be a remote condenser apparatus.

With a pressure decrease of the circulating nitrogen to below 20 atm the variances in the power consumption increase.

At  $p = 1.5$  atm the power consumption is about 1.3 kw·hr per kilogram of liquid  $O_2$ . Moreover, in this case, a more substantial variance in the size of the required surface of the heat-exchanger is observed (see Fig. 4).

The practical expedient choice of pressure of this circulating nitrogen is characterized by the curve  $A_{cn}$  (Fig. 4).

When  $A_{cn}$  is close to the value  $1 - 1.1 \text{ m}^3/\text{m}^3\text{ta}$ , we can use for the preliminary compression of nitrogen (in the first stage), the same turbocompressor as for the compression of technological air; in this case, the construction of the complete-compression nitrogen turbocompressor (second stage) is simplified.

If we designate the theoretical work of air liquefaction as  $L$  Carnot, and that of air separation as  $L_0$ , the thermodynamic efficiencies of the examined cycles are established by the following ratio:

$$\eta_r = \frac{L_{\text{Carnot}} + L_0}{N_0}$$

For air,  $L$  Carnot = 0.305 kw·hr/kg of liquid air;  $L_0 = 0.055$  kw·hr/kg of liquid  $O_2$ . Figure 5 shows the curve which characterizes the change of the thermodynamic efficiency of the examined cycle with nitrogen circulation.

The characteristics obtained allow selection of the optimum pressure in the circulation cycle. Bearing in mind that at pressures  $p = 19 - 24$  atm the isothermal efficiency of the turbocompressor may be from 4-6% higher than that of the turbocompressor with a degree of compression 28 - 32 (this fact is not considered in the calculation and is not shown in Fig. 4), we can assume that for the range of pressures from 19-20 atm to 30-32 atm there is no real

difference in the specific power consumption. Thus, the relative increase in the weight and dimensions of the heat-exchanger is 20-25% in comparison with the high-pressure-cycle apparatus.

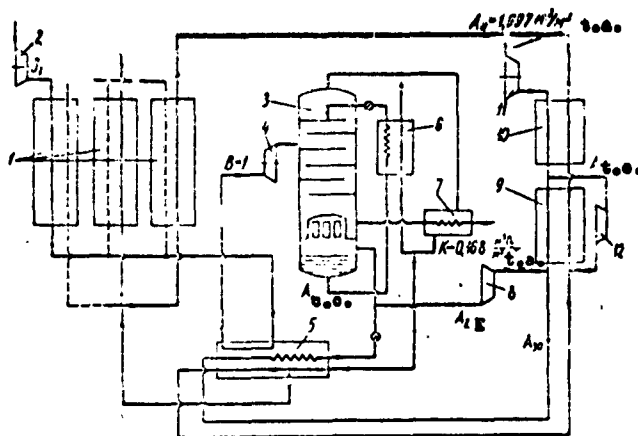


Fig. 1. Diagram of an oxygen installation with an average-pressure nitrogen circulation cycle.

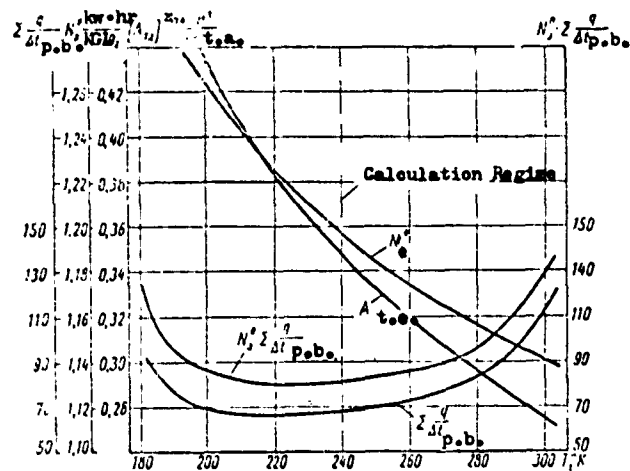


Fig. 2. The influence of nitrogen temperature  $T_1$  ahead of the complete-expansion turboexpander on the parameters  $N_e^0$ ,  $A_{t.e.}$

and on the product  $N_e^0 \sum \frac{q}{\Delta t p_{o.b.}}$

at a nitrogen pressure  $p = 30$  atm.

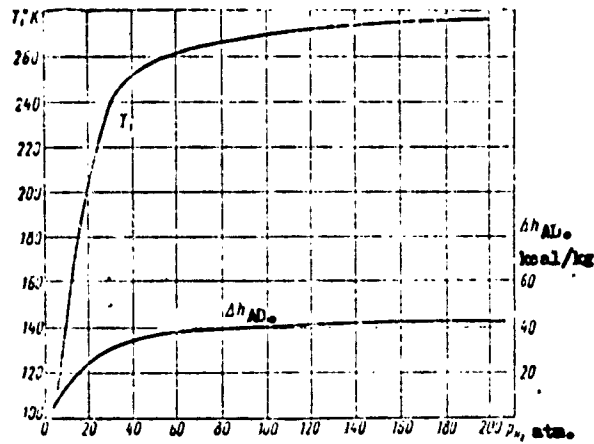


Fig. 3. Change of the optimum temperature  $T_1$  of nitrogen ahead of the turbo-expander and of the complete adiabatic heat drop  $\Delta h_{p.b.}$  with optimum initial temperature  $T_1$  (with respect to air) vs. the pressure of the circulation on cycle.

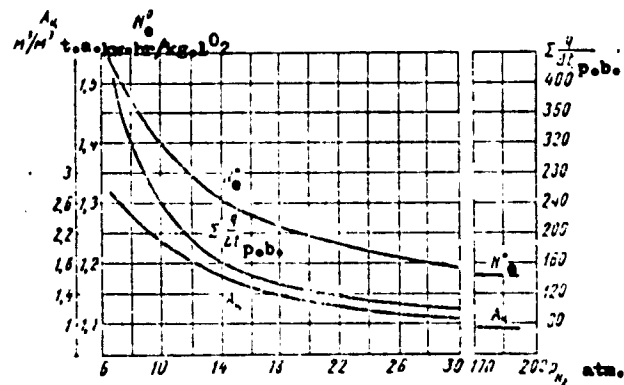


Fig. 4. Characteristics of cycles with nitrogen circulation.

Considering the expediency of using a turbocompressor for compressing the technological air, instead of the nitrogen turbocompressor of the first stage, we can consider that the optimum value of the maximum pressure of nitrogen in the circulation cycle

is 30 - 32 atm. Soviet turbomachine construction has confirmed the possibility of creating a turbocompressor designed for such a pressure with an isothermal efficiency of 50-55%.

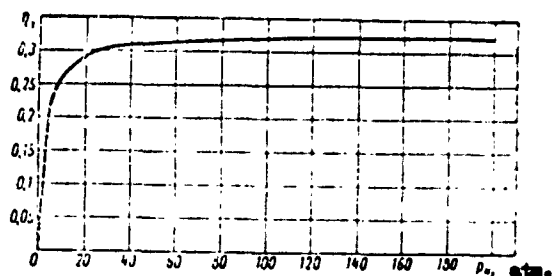


Fig. 5. Change of the thermodynamic efficiency of the cycle with nitrogen circulation.

From the preceding it follows that for large-scale installations the use of high-pressure ( $p = 200$  atm) in the cycles with nitrogen circulation does not substantially increase the efficiency. Such a measure is also not justified from the standpoint of complexity, high cost, and the large dimensions of the equipment — the piston compressors and expanders. The high-pressure cycle is somewhat more advantageous with respect to the heat exchanger. As the results of refined calculations indicate, the heat-transfer coefficient in the basic and preliminary high-pressure-cycle heat exchangers is approximate twice that in the average-pressure-cycle ( $p = 30$  atm) heat exchangers. The curve  $\eta \sim \frac{1}{\Delta t_{p_0 b_0}}$  in Fig. 4 does not take into account the change of the heat-transfer coefficient since for the wide range of average pressure this change is insignificant; for high pressures, however (170 to 200 atm), the curve is purposely not drawn.

Considering the increased thickness of the walls of the heat-transfer surfaces in the high-pressure-cycle heat-exchangers, their total general weight is decreased by a factor of approximately 1.5 compared to the weight of the average-pressure-cycle heat-exchanger, while their overall size is approximately half as large. The preceding is significant also with respect to the decreased heat influx from the environment. This is why the use of high pressure is more rational in the small-scale deep cooling installations, when the use of a turbomachine is impossible.

The concentration of nitrogen discharged from the column and circulating in the cooling cycle influences the power characteristics of the cycle. When the purity of the nitrogen is increased and the output remains constant, there is a decrease in the power used to compress the air and to compensate for losses due to insufficient recuperation into the environment as well (per unit of the obtained product). On the other hand it is necessary to increase the cold output of the circulation cycle; moreover, with increased concentration of nitrogen from the column, the power expended on separation increases. These facts are reflected in the increased amount of circulation nitrogen  $A_{cn}$ . Considering the possible use of an air turbocompressor to compress the nitrogen in the first stage, it was established that for a cycle with a pressure of 30 atm, with optimum temperature  $T_I = 220^\circ - 240^\circ K$  of the nitrogen ahead of the turbo-expander, the most suitable is  $Y_{N_2} = 95-96\% N_2$ , for which  $A_{cn} = 1.097 m^3/m^3ta.$

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