of the shaft between the bearings. The slide fits in the groove of the link which is made integral with the two plungers 8 located in the working chambers of the housings 7. Poppet suction and pressure valves are mounted in the housings. These valves have a common optimized design which reduces the weight of the moving parts to the minimum. The technical characteristics of the pump are as follows: output per rotation of the handwheel - 21 cm³; nominal pressure - 5.2 MPa; maximum pressure - 8 MPa; force required on the handle - 180 N (at nominal pressure); number of rotations of the handwheel required for opening and closing the ball valve - 476 and 439, respectively.

The pump design has the following advantages: concurrent suction and pressure strokes; uniform force on the handwheel; smooth movement of the plungers; improved ergonomic parameters because of replacement of the lever by a handwheel; reduced pulsation of oil flow; standardized parts; and compactness.

The pump parts are easy to manufacture. The housing can be machined in one setting which ensures locational accuracy of the mating surfaces. The weight of the block can be reduced by 25-30% if the housing parts are made from castings.

THEORETICAL AND EXPERIMENTAL DETERMINATION OF OPTIMUM PRESSURE FOR FILLING GAS CYLINDER SYSTEMS HAVING SELF-REGULATED MICROREFRIGERATORS

N. D. Merkel' and E. A. Fisher

The technical level of cylinder throttle microcryogenic systems (CTMCS) is generally characterized by the relative effective exergy \( \varepsilon_0 \) which is a ratio of exergy of heat to be removed from the object of cooling to the initial mass of the system [1]:

\[
\varepsilon_0 = \frac{\frac{\tau_e}{RT} \left[ \int_{p_1}^{p_2} \frac{\Delta t \eta_w d}{p} + \int_{p_2}^{p_3} \frac{\Delta t \eta_d d}{p} \right] + \rho_c \left[ \frac{4}{4 + 3h} \left( \frac{4 + Ap_1}{4 - Ap_1} \right)^3 + \frac{3h}{4 + 3h} \left( \frac{2 + Ap_1}{2 - Ap_1} \right)^2 - 1 \right] + \frac{1}{RT} \left( \frac{p_1}{z_1} \right) \right]}{\rho_c \left[ \frac{4}{4 + 3h} \left( \frac{4 + Ap_1}{4 - Ap_1} \right)^3 + \frac{3h}{4 + 3h} \left( \frac{2 + Ap_1}{2 - Ap_1} \right)^2 - 1 \right] + \frac{1}{RT} \left( \frac{p_1}{z_1} \right)},
\]

where \( \tau_e = (T - T_{cr})/T_{cr} \) is the exergic temperature function; \( T \) and \( T_{cr} \) are respectively the ambient and cryostating temperatures; \( R, \Delta i, z \) are respectively the gas constant, isothermal throttle (Joule-Thompson) effect, and cryoagent compressibility factor; \( p_1, p_2, \) and \( p_3 \) are respectively the pressure of the cryoagent at the moment of filling, start of operation under set conditions, and disruption of the set conditions; \( \eta_w \) and \( \eta_d \) are the factors of utilization of the available refrigerating power during the working and starting periods respectively; \( \rho_c \) is the density of the cylinder material; \( h = h/r \) is the relative height of the cylindrical part of the gas cylinder; \( h \) and \( r \) are respectively the height and inner radius of the cylinder and hemispheres; \( A = \omega / (\lambda \sigma_c) \) is the parameter characterizing the strength properties of

### Table 1

<table>
<thead>
<tr>
<th>Cryoagent</th>
<th>$P_{\text{lopt}}$, MPa</th>
<th>$\varepsilon_0$, kJ/kg at $P_2 = 35$ MPa</th>
<th>$\varepsilon_0$, kJ/kg at $P_{\text{lopt}}$</th>
<th>$\eta$, %</th>
<th>$m$</th>
</tr>
</thead>
<tbody>
<tr>
<td>Nitrogen</td>
<td>45; 50</td>
<td>16.56; 12.2</td>
<td>16.96; 13.12</td>
<td>2.8; 7.54</td>
<td>1.3; 1.51</td>
</tr>
<tr>
<td></td>
<td>50; 50</td>
<td>12.36; 10.84</td>
<td>13.53; 12.31</td>
<td>6.69; 13.56</td>
<td>1.5; 1.62</td>
</tr>
<tr>
<td>Air</td>
<td>45; 55</td>
<td>19.48; 14.21</td>
<td>20.18; 15.55</td>
<td>3.59; 9.43</td>
<td>1.33; 1.55</td>
</tr>
<tr>
<td></td>
<td>50; 55</td>
<td>15.37; 12.64</td>
<td>16.55; 14.77</td>
<td>8.36; 16.05</td>
<td>1.18; 1.22</td>
</tr>
<tr>
<td>Argon</td>
<td>55; 65.5</td>
<td>22.82; 18.45</td>
<td>25.41; 22.29</td>
<td>12.9; 20.5</td>
<td>1.71; 1.98</td>
</tr>
<tr>
<td>Mixture</td>
<td>35; 37.5</td>
<td>19.63; 17.68</td>
<td>22.22; 21.39</td>
<td>13.2; 19.63</td>
<td>8.73; 1.9</td>
</tr>
<tr>
<td></td>
<td>47.5</td>
<td>20.17</td>
<td>31.15</td>
<td>8.79</td>
<td>1.41</td>
</tr>
<tr>
<td></td>
<td>42.5; 43.9</td>
<td>33.22; 29.74</td>
<td>33.73; 31.24</td>
<td>4.03; 5.04</td>
<td>1.21; 1.29</td>
</tr>
</tbody>
</table>

**Note:**
1. Results of calculations respectively at $\eta_W = 1$, $\varepsilon_T = 1$, and $\eta_W = \bar{f}(p)$, $\varepsilon_T = 0.985$ (for a mixture only of the second variant) are put in the numerator and the results of analysis of experimental data respectively at $P_2 = P_1$ and $P_2 < P_1$ in the denominator.
2. The inversion pressure $P_1$ for nitrogen, air, argon, and the gas mixture are respectively 37.5, 42.5, 57.5, and 43 MPa.

The first term in the numerator of Eq. (1) characterizes the exergy of heat removed from the object of cooling under the cryostating conditions, the second, in the starting period. The factor of utilization of the available refrigerating power of the microrefrigerator depends on the ratio of the effective refrigerating power of the cycle $q$ to the available power $q_{\text{avail}}$

$$\eta = \frac{q}{q_{\text{avail}}} = 1 - \frac{c_p' \delta_{\text{hd}}}{T - T_{cr}}$$

and can be expressed through the thermal efficiency of the heat-exchanger of the microrefrigerator $\varepsilon_T$:

$$\varepsilon_T = \frac{q_{\text{hd}}}{T - T_{cr}}$$

$c_p'$ being the specific heat of the low-pressure stream at the microrefrigerator outlet and $\delta_{\text{hd}}$, the temperature head at the hot end of the microrefrigerator.

Equation (1) is valid for a cylindrical gas cylinder with hemispherical bottoms and a volume $V_C = \frac{1}{3}\pi r^2(4+h^3)$, which turns into a sphere at $h \to 0$ and an infinite cylinder at $h \to \infty$.

In determining the weight of the cylinder its shell was considered thick-walled and for calculating its strength use was made of the relations recommended in standard documents [2].

Analysis shows that Eq. (1) has an extremum, i.e., there is an optimum filling pressure $P_{\text{lopt}}$ at which $\varepsilon_0$ is maximum. As shown in [1], use of optimum nitrogen filling pressure ($P_{\text{lopt}} = 50$ MPa) in place of the traditional one ($P_1 = 35$ MPa) permits one to reduce the parameter $\varepsilon_0$ by 10% with a 1.5-fold increase in the duration of continuous operation of the CTMCS. But these data were obtained with reference only to a spherical cylinder and not confirmed experimentally, which does not allow practical recommendations on their basis.

It is of interest to make a deeper theoretical and experimental study for determining $P_{\text{lopt}}$ as a function of the type of the cryoagent, shape and material of the cylinder, degree of refinement of the self-regulated throttle microrefrigerator (STM), and relative cryoagent requirement for the start.

The calculations were performed for nitrogen, argon, oxygen, air, and a gas mixture containing nitrogen, Freon-14, Freon-13, and neon. The thermophysical properties of the pure gases and air were determined from tables [3, 4] and for the mixture they were calculated on the basis of the principle of the respective states with determination of the pseudocritical constants from the modified Prounitz and Gunn's rule [5]. For simplifying the analysis the