

THERMOPHYSICAL PROCESSES IN CRYOSPHERE
**ON THE CHOICE OF REFRIGATING FLUID FOR
TYPE “GET” SYSTEMS FOR SEASONAL COOLING**

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The paper provides a description of “GET” systems operation designed for soils seasonal cooling using different kinds of working fluids, and offers a key criterion for evaluating the performance efficiency of “GET” systems, when working on some specific types of refrigerants. It has been shown that the “GET” system runs well if such refrigerating fluids, as carbon dioxide, ammonia, or Freon 22 are used. However, the “GET” system is more likely to fail to work if methylene chloride, acetone, Freon 113 and methanol are chosen as working fluids.

Cooling systems, refrigerating fluid (refrigerant), temperature, pressure

Russian “GET”, seasonal (genuine thermal energy-based) cooling systems with horizontal pipes are described in detail by [Dolgikh et al., 2008]. Fig. 1 presents a flow diagram for the “GET” system working cycle, the equations to substantiate the efficiency of horizontal closed-loop systems for soils cooling provided in [Anikin, 2009]. Mathematical modeling of such systems with ammonia as a working fluid is discussed in detail in [Anikin et al., 2011]. However, the types of specific refrigerants that can be used in such systems are yet to be investigated. The lack of these presents a problem that we are attempting to address. As it was shown in [Anikin et al.,

2011], the maximum temperature in the “GET”-system evaporator is defined by expression:

$$t_{\max} = t_k + \frac{\rho_L g H - \Delta p}{(dp_{\text{nas}}/dt)}, \quad (1)$$

where t_k – is condensing unit temperature; ρ_L – is refrigerant liquid density; g – is acceleration due to gravity; H – is height of condensing unit above evaporator; $\frac{dp_{\text{nas}}}{dt}$ – is derivative from the saturated

vapor pressure versus temperature; Δp – pressure difference required to overcome frictional resistance encountered in the interval between condenser and the point of evaporator where temperature reaches its maximum – t_{\max} [Anikin et al., 2011]. Numerical modeling of the “GET” system operation with the use of a simulation method described in [Anikin et al., 2011] has shown that the following condition is always fulfilled:

$$\Delta p \ll \rho_L g H, \quad (2)$$

which allows to present t_{\max} as follows:

$$t_{\max} = t_k + \frac{\rho_L g H}{(dp_{\text{nas}}/dt)}. \quad (3)$$

It is obvious that, for heat to be transferred from evaporator to the ambient air, in accordance with Fourier's law the inequalities given below should be satisfied:

$$t_a < t_k < t_{\max}, \quad (4)$$

where t_a – is atmospheric temperature. It follows from (3) and (4) that:

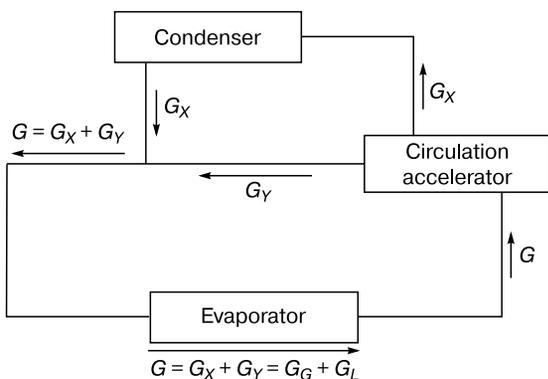


Fig. 1. Flow diagram for “GET” systems.

G_x – liquid mass, flowing from condenser; G_y – liquid mass, flowing from circulation accelerator; G_l – liquid mass flow; G_g – gas mass flow.

$$t_{\max} > t_a + \frac{\rho_L g H}{(dp_{\text{nas}}/dt)}. \quad (5)$$

For the soils to freeze to subzero temperatures it is essential, that at any point along the evaporator tube, the refrigerant temperature t_i is negative, i.e. when the inequality (6) is satisfied

$$t_i < t_{\max} \leq 0 \text{ }^\circ\text{C}. \quad (6)$$

As follows from relations (5) and (6), for the "GET" system operations it is required that the following condition is satisfied:

$$t_a < -\Delta t H, \Delta t = \frac{\rho_L g}{(dp_{\text{nas}}/dt)}. \quad (7)$$

The value of $\Delta t H$, as a part of the expression (7) represent by itself a maximum difference between the condenser and evaporator temperatures, and Δt – is analogous value at $H = 1$ m. As is seen from (7), Δt depends only on the refrigerant characteristics. To assess the impact of the refrigerant on the "GET" system's performance, we replace the derivative in equation (7) with the difference, to obtain:

$$\Delta t = \frac{0.5(\rho_{L1} + \rho_{L2})g(t_2 - t_1)}{(p_{\text{nas}2} - p_{\text{nas}1})}. \quad (8)$$

Here ρ_{L1}, ρ_{L2} – are densities of the refrigerating fluid at t_1 and t_2 , respectively; $p_{\text{nas}1}, p_{\text{nas}2}$ – are saturated vapors pressures at temperatures t_1 and t_2 . Calculated by formula (8), Δt values at $t_1 = -20$ °C and $t_2 = -0$ °C are provided in Table 1.

To analyze the "GET" system performance using different refrigerants, we investigated the temperature conditions at the Vankorskoye oil and gas field. The air temperature data from the *Igarka* weather station were reported every two hours, and averaged monthly at the time interval from the beginning of June 2008 through the end of March 2012. The average monthly temperatures within this time span are shown in Table 2.

We calculated a number of months in the considered period for each refrigerant given the condition (7) was satisfied in each, and was denoted this number by n . The total number of months with negative temperatures within this period is denoted by m . The values obtained are shown in Table 3, with the condenser elevation relative to evaporator, as $H = 2, 3$, and 1 m.

As is seen from Table 3, the refrigerants, allowing for "GET" systems operations for most of the winter season, are carbon dioxide, ammonia, and Freon 22. The refrigerants, the "GET" system is not designed to use as working fluids, are methylene chloride, ace-

Table 1. Δt values for different refrigerants

Refrigerating fluid	$p_{\text{nas}1}, 10^5$ Pa	$p_{\text{nas}2}, 10^5$ Pa	$\rho_{L1}, \text{kg/m}^3$	$\rho_{L2}, \text{kg/m}^3$	$\Delta t, \text{ }^\circ\text{C}$
Carbon dioxide	19.7	34.9	1030	924.8	0.13
Ammonia	1.90	4.3	665.1	638.6	0.53
Freon 22	2.46	4.98	1351	1285	1.02
Freon 12	1.53	3.13	1457	1395	1.75
Freon 142	0.65	1.47	1218	1170	2.85
Freon 21	0.29	0.71	1470	1380	6.59
Freon 11	0.16	0.40	1580	1534	12.47
Methylene chloride	0.06	0.18	1359	1328	22.45
Acetone	0.03	0.10	845	812	23.22
Freon 113	0.05	0.15	1664	1626	32.93
Methanol	0.01	0.04	828.5	810.4	53.06

Table 2. Average monthly temperatures from Igarka weather station

Date	$t_a, \text{ }^\circ\text{C}$										
06.2008	10.19	07.2008	15.51	08.2008	12.13	09.2008	7.82	10.2008	-4.19	11.2008	-18.27
12.2008	-23.80	01.2009	-23.61	02.2009	-30.48	03.2009	-20.97	04.2009	-6.82	05.2009	-3.15
06.2009	9.11	07.2009	17.41	08.2009	13.19	09.2009	7.19	10.2009	-4.65	11.2009	-22.02
12.2009	-34.55	01.2010	-29.03	02.2010	-33.88	03.2010	-16.21	04.2010	-6.97	05.2010	-1.54
06.2010	8.04	07.2010	12.97	08.2010	10.00	09.2010	2.97	10.2010	-2.93	11.2010	-20.91
12.2010	-31.66	01.2011	-24.78	02.2011	-26.59	03.2011	-11.85	04.2011	-3.37	05.2011	4.46
06.2011	13.93	07.2011	12.07	08.2011	10.98	09.2011	8.10	10.2011	-2.01	11.2011	-17.75
12.2011	-16.28	01.2012	-27.17	02.2012	-18.17	03.2012	-15.36	-	-	-	-

Table 3. n, m – parameters and their ratio at various values H

Refrigerating fluid	$H = 2$ m					$H = 3$ m					$H = 1$ m				
	$\Delta t, ^\circ\text{C}$	$-2\Delta t, ^\circ\text{C}$	n	m	n/m	$\Delta t, ^\circ\text{C}$	$-3\Delta t, ^\circ\text{C}$	n	m	n/m	$\Delta t, ^\circ\text{C}$	$-\Delta t, ^\circ\text{C}$	n	m	n/m
Carbon dioxide	0.13	-0.25	29	29	1	0.13	-0.38	29	29	1	0.13	-0.13	29	29	1
Ammonia	0.53	-1.06	29	29	1	0.53	-1.60	28	29	0.97	0.53	-0.53	29	29	1
Freon 22	1.02	-2.05	27	29	0.93	1.02	-3.07	26	29	0.89	1.02	-1.02	29	29	1
Freon 12	1.75	-3.50	23	29	0.79	1.75	-5.25	22	29	0.76	1.75	-1.75	28	29	0.97
Freon 142	2.85	-5.70	22	29	0.76	2.85	-8.54	20	29	0.69	2.85	-2.85	27	29	0.93
Freon 21	6.60	-13.20	19	29	0.66	6.60	-19.80	13	29	0.45	6.60	-6.60	22	29	0.76
Freon 11	12.46	-24.92	7	29	0.24	12.46	-37.38	0	29	0	12.46	-12.46	19	29	0.66
Methylene chloride	22.44	-44.88	0	29	0	22.44	-67.32	0	29	0	22.44	-22.44	11	29	0.38
Acetone	23.22	-46.44	0	29	0	23.22	-69.66	0	29	0	23.22	-23.22	10	29	0.34
Freon 113	32.94	-65.88	0	29	0	32.94	-98.82	0	29	0	32.94	-32.94	2	29	0.07
Methanol	53.06	-106.12	0	29	0	53.06	-159.18	0	29	0	53.06	-53.06	0	29	0

Note. n – is number of months, in which condition (7) was satisfied; m – is number of months with below-zero atmospheric temperature.

tone, Freon 113, and methanol; whereas other refrigerants are subsumed into interim class. In [Gorelik and Gorelik, 2011], however, it was suggested that the system can operate using acetone, when horizontal evaporator is employed.

The contradiction between the results and conclusions in this paper [Gorelik and Gorelik, 2011] is actually superficial, as the efficiency of the system performance depends on the elevation of condenser relative to evaporator. So, if the condenser is placed 1 m above evaporator, then, as follows from Table 3, the system operates a third of the winter using acetone. However, this appears to be not practical, since the evaporator pipe is typically buried at a depth of 1 m below the surface, and the condenser is supposed to be positioned at least 1.5 m above ground level, otherwise it may be snowed under. Thus, the value of H should not be less than 2.5 m (in actual systems, H ranges from 2.5 to 6 m).

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