Year-round thermal stabilization of permafrost soils during road construction in the northern climatic zone of Russia

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Abstract. The device for year-round cooling of permafrost soils is designed to ensure the construction and operation of roads and railroads in the tundra zone. It includes pipes buried in the ground near the roadbed, a wind turbine powering an electric current generator, an absorption-type cooling unit, and an exhaust fan ensuring the flow of atmospheric air through the pipes. During the long and cold winter, the pipes laid in the ground provide deep freezing of the ground beneath a thick layer of snow. In spring and autumn, at temperatures close to 0°C, the air in the pipes is forcibly cooled by the cooling unit. In summer, at air temperatures exceeding 10°C, air movement through the pipes is blocked. In conditions of short summer, low levels of the sun above the horizon, the high albedo of snow and ice, and their low thermal conductivity, snow deposits remain on the ground throughout the summer period, and in the subsequent winters, accompanied by snowfalls, they intensify. Around a roadway built on permafrost, a protective snow layer is formed, preventing the flow of heat from the atmosphere and solar radiation into the ground. The protective layer can exist for decades, even if the Earth’s climate changes globally. The article presents hydraulic and thermal calculations, explaining the operation of the device in question. Keywords: permafrost soil, ground thawing, thermowell, wind power plant, buried cooling pipes, coolant, cooling unit.

1 Introduction

The development of territories in the North and East of Russia associated with the construction of roads and railroads, as well as various engineering facilities and structures encounters many difficulties. They are associated with the almost ubiquitous spread of permafrost, the freezing of waterlogged soils in winter, and their deep thawing in summer. Numerous studies have shown that deformations of, for example, railroad tracks or roadbeds built on permafrost are caused by thawing and subsidence of permafrost soils at the base of the bed [1-9]. Thawing usually occurs under high and wide embankments, which is facilitated by the low albedo (8...12%) of solar radiation of soils, granites, and pebbles at the

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base of the roadbed. When the depth of seasonal ground freezing is low, talks several meters thick and tens or hundreds of meters long are formed under the embankment [10,11].

There is a very important task of maintaining sub-zero temperatures of soils under road or railroad tracks, under buildings for industrial and civil purposes, etc. [12].

This problem is solved by intensive cooling of soils, during the whole winter and springtime, so that a layer of snow and ice is preserved on the ground surface as long as possible (ideally during the whole summer period). The albedo for ice and snow is up to 60...80%, they have low thermal conductivity ($\lambda_{\text{ice}}=0.35$-$2.2$ W/m$^2$K) and a very high specific heat of melting ($c=330$ kJ/kg). This allows snow-ice deposits to be used as a barrier to summer heat fluxes from the atmosphere into the ground. At present, the freezing of weakened soils is ensured by the creation of a pile field made of thermopiles installed in the foundations of buildings, under railroad and freeway beds, etc. [12].

Thermos hafts are made in the form of pipes plugged in the bottom part, filled with a liquid coolant, such as kerosene, which has a low freezing point, a high coefficient of volumetric thermal expansion, and low viscosity. The upper part of the thermowell is in contact with atmospheric air, while the lower part is buried in the ground. When winter frosts, the density of the heat transfer fluid in the upper part of the thermowell increases, becoming higher than in its lower part. In this case, there is free convection of the coolant in the pipe, which contributes to the removal of heat from the ground to the atmosphere.

The practice of operation of thermos piles has shown that as the temperature difference between the atmosphere and the cooled ground decreases the equivalent thermal conductivity provided by the coolant becomes negligibly small. Thermal pile freezes the ground only at the beginning of winter and only at very strong frosts. In summer, when there is the weakening of frozen soils, thermos piles do not function according to their intended purpose, as the liquid in the pipe enters a stable stratified state, in which its density in the lower part of the pipe becomes higher than in the upper part, heated by solar radiation and warm atmospheric air. The motion of the coolant, causing cooling of the ground, stops.

Even with a pile field, for example, in Yamal, the ground remains warm enough by October, when snowfalls begin, and the fallen snow immediately begins to play the role of thermal insulation, preventing the ground from freezing in the winter. The seasonal freezing of soils turns out to be insufficient.

In the work under consideration it is proposed to turn the powerful snow cover, formed near the roadbed "from an enemy into a friend", i.e. to use it to strengthen the freezing of soils, and on large areas with minimal expenditure of resources and without involving service personnel.

The proposed device is intended for use in the Northern climatic zone of Russia, in particular on the Yamal peninsula, where oil and gas fields are discovered and intensively developed, where construction of new enterprises and cities is carried out accompanied by population growth, where intensive development of railway and road network is required. In northern regions of Russia, the duration of wintertime reaches 9...10 months a year and is accompanied by severe frosts, winds, and snowstorms. But with the global change of the Earth's climate, the short summer period is getting warmer, and sometimes even hotter. Increasing seasonal thawing of permafrost soils makes it very difficult to build large objects such as roads and railroads. Figure 1 shows a version of a device for continuous, almost year-round, freezing of weakened soils, in the area of construction or operation of roads, buildings, and structures.
Let in a ground, on depth h from its surface the pipes 2 having diameter 2R≈0,3m and length l=1≈500...1000m large enough for overlapping of the powerful talk of length 300...500m are buried parallel to a roadbed. If necessary, instead of one pipe a battery of pipes can be laid in the ground, where the pipes are connected using a collector. In the battery pipes are laid parallel to each other at a distance of S. Cold atmospheric air with an average temperature of t_a≈-20°C is pumped through the pipes during the whole winter.

A distinctive feature of atmospheric conditions in the Northern climatic zone of Russia is stable windy weather. Strong winds are observed both in the spring-summer period when they have speed up to u_1=10...20 m/s, and weaker in winter, when they have speed u_2=5...10m/s. The direction of winds is predominantly meridional, from north to south in summer and from south to north in winter. This peculiarity is caused by the proximity of large land areas on the Eurasian continent and the Arctic Ocean, as well as by a sharp difference in the physical characteristics of soils and water.

The named climatic features of the North of Russia allow the use of wind power plants (WPP) for forced pumping of cold atmospheric air through the cooling pipes 2, providing cooling and freezing of the ground 1.

Let's assume that the cooling system of weakened soils is operated autonomously, based only on natural energy sources located in the Northern climatic zone of Russia (primarily the Yamal Peninsula), and includes a coolant supply unit in the cooling tube 2 (Fig. 2) and its cooling unit.

Hydraulic and thermal calculations are performed for one cooling pipe, buried in the ground near the roadbed. The heat carrier, pumped through the pipe, 2 is the atmospheric air, cold in winter and forcedly cooled at atmospheric temperatures close to 0°C (i.e., in early spring and the pre-winter period). Let's assume that the energy source, providing the forced pumping of atmospheric air through tube 2 and its cooling is a wind power plant (WPP). In Yamal, other energy sources can also be used, such as, for example, associated gases currently burned at the oil refinery.

2 Solution methods

The unit includes the following units: an inlet water and snow breaker (IWB) designed to prevent ice and snow deposits inside the pipes cooling the ground, a wind power unit (WPU), a battery of pipes buried in the ground parallel to the roadbed, a wind flow converter (WFC) supplying cold air into the pipes, operation of an electric power generator and a cooling unit cooling air before the pipes during the spring and summer. All the equipment operates autonomously, with minimum use of service personnel, involved only for routine repair works 1...2 times a year. Let's list the named units.
The peculiarity of climatic conditions of the North is the high-water saturation of atmospheric air. In winter atmospheric water is in a solid state, in the form of ice crystals, forming large and small snowflakes, in summer-in a liquid state, forming fog and rain drops. Specific water content in the atmosphere is defined by the term “water content”, which we will denote by \( W \), g/m\(^3\). The most unfavorable operating conditions of cooling pipes in terms of the water content of the air stream correspond to the temperature range from -10\(^\circ\)C to +5\(^\circ\)C. The water content of air can reach \( W \approx 1...2 \) g/m\(^3\), and the water has a low temperature. Fine rain and drizzle, getting into the pipe, freeze in it, forming ice, and snow, can accumulate in the elbows and constrictions of the pipe up to complete overlapping of its flowing section. At higher air temperatures below -10\(^\circ\)C atmospheric water content decreases dozens of times, and air temperatures exceeding +3\(^\circ\)C already meet the conditions of summertime when ice difficulties come to naught.

In its simplest form, the HSS is a steel vessel of a large volume with transverse partitions installed in it and the floor having a slope. It can be made, for example, from the boiler shell of a tank wagon, which is not in circulation due to non-compliance with the standards.

In the left part of the boiler, the upper filler hatch is retained, and in its bottom, a through hole is cut through, in which the entrance to the cooling tube is mounted. Inside the boiler, partitions are welded to form a serpentine corridor on the walls of which the water contained in the air (liquid water in summer-autumn and snow in winter) is deposited.

When atmospheric air enters the boiler, its flow rate is sharply reduced, while the water contained in the air due to inertial forces precipitate on the transverse partitions and accumulate on the floor of the boiler. The capacity of the boiler part should be sufficient at least 10m\(^3\) to contain the entire mass of water entering it for the entire autumn and winter period. In summer, at positive air temperatures, all the accumulated mass of snow and ice should melt and drain by gravity outside.

Oxidized smooth steel (if the boiler is left unpainted) has a high degree of blackness \( \varepsilon = 0.82 \), paint black, matte (if the boiler is painted) has a degree of blackness even greater \( \varepsilon = 0.92...0.96 \).

Specific heat flux from solar radiation in the Yamal latitude is about 600W/m\(^2\). With the area of the boiler part of 50m\(^2\) for heating and melting of ice and snow and evaporation of residual water, there is a heat power of about 30 kW, and the time of its activities during the polar day can reach 500 hours.

This should be more than enough to completely dry out the VCO for the next winter.

The pressure drops outcomeing from the local resistance of all channels of the VSO, shown in Fig. 1, can be estimated by the Weisbach formula:

\[
\Delta p = \sum_{k=1}^{n} \sum_{k} \frac{\rho g u_k^2}{2},
\]

Where: \( \rho_g = 1.2 \)–air density, kg/m\(^3\); \( k \)–the number of channels in the VSO (Figure 1 shows \( k = 4 \) channels); \( u \approx 4 \) m/s average air velocity in the channel; \( \zeta \approx 1.2 \) local resistance coefficient. Calculation according to formula (1) shows that the pressure drops created by HSO do not exceed \( \Delta p = 5 \) Pa, which is a small value.

Let's assume that the node of the cold air supply to cooling tube 2 is a wind flow converter (WFC). We assume that this product, manufactured in factory conditions, is mounted on the frame of a freight wagon with \( H \times W \times L \) dimensions. The PVP unit is made based on a standard freight container 1CC, by GOST R 51891 satisfying the conditions of transportation by railway, river, and road transport. Characteristics of 1CC container are given in Table 1.
Four wind wheels, generators of electrical energy with a total capacity of not less than 10 kW, generating an electric current up to I=75A, are installed inside the room PVP. There are also two inverters with a capacity of 5 kW each and a block of batteries to ensure the operation of the control electrical appliances. Inverters are intended for adaptation of the power supply of all elements of the PVP to autonomous working conditions. Parameters of the inverters are given in Table 2.

The entrance to the PVP room is made through a lockable hatch installed on its roof. There are partitions inside the CWP, forming boxes, in which the wind wheels are installed. Wind wheels are installed inside the wind turbine unit, on its side walls, each in its box by the direction of the wind rose. Adjustment of the airflow rate to the wind wheel must be provided by the gate installed on the air intake hatch of the box. The slide gate is designed to protect the impeller of the wind wheel from snow and dripping ice, most likely at air temperatures of +3°C...-10°C. The shape of the boxes provides a narrowing of incoming air flows, created by the wind, to increase their speed in front of the wind wheels. This provides an increase in the planned capacity of each electric generator, fed by a wind wheel up to 8 kW.

### Table 1. Characteristics of the 1CC container.

<table>
<thead>
<tr>
<th>Container</th>
<th>Height H, mm</th>
<th>Width W, mm</th>
<th>Length L, mm</th>
<th>Area LW, m²</th>
<th>Area HW, m²</th>
<th>Area LH, m²</th>
<th>Weight Gross R, kg</th>
</tr>
</thead>
<tbody>
<tr>
<td>1CC</td>
<td>2591</td>
<td>2438</td>
<td>6058</td>
<td>14.77</td>
<td>6.34</td>
<td>15.75</td>
<td>3050</td>
</tr>
</tbody>
</table>

The practice of operation of wind turbines for various purposes has shown that their power depends on the air speed V, m/s, and the radius R of the wind wheel, m. All other factors—the numbers of blades on the impeller, their weight, area, and profile do not give significant errors. At the permissible error of calculation, equal to the power of the wind turbine in kilowatts is estimated by the empirical formula:

\[
P = \frac{n^2V^3}{7000}
\]  

Research of different models of wind turbines, operated for the power supply of buildings and constructions, showed that horizontal-axis units have significantly better characteristics in comparison with vertical-axis units and are more often used in practice. Below we consider the horizontal-axis wind turbine with four wind wheels.

### Table 2. Parameters of powerful inverters (site www.outbacpower.com).

<table>
<thead>
<tr>
<th>Inverter model</th>
<th>Technical parameters</th>
</tr>
</thead>
<tbody>
<tr>
<td>FX2348ET</td>
<td>2.3 kW/48V DC/230V AC/50 Hz</td>
</tr>
<tr>
<td>VFX3024E</td>
<td>3 kW/24V DC/230V AC/50 Hz</td>
</tr>
<tr>
<td>MPI -LCD</td>
<td>48 V - 4.5 kW</td>
</tr>
<tr>
<td></td>
<td>48 V - 6 kW</td>
</tr>
</tbody>
</table>
Geometric characteristics of the duct: duct height: \( b_1 = 1500 \text{ mm} \), \( b_2 = 500 \text{ mm} \), canal thickness \( h = 400 \text{ mm} \), draft angle \( \alpha = 20^\circ \). 

The layout of the equipment of one box is shown in Fig. 2. Each box 1 is made of a steel sheet with thickness \( \delta = 1.5 \text{ mm} \), and forms two identical ducts with thickness \( h = 400 \text{ mm} \). Inside box 1, the wind wheel 2 and two identical ducts 3 and 4 are mounted. Wind wheel 2, diameter \( D = 1150 \text{ mm} \), has a rotational axis at the midline of the walls of box 1. It is planned that each wind wheel provides the rotation of a standard alternator VG-5 (28)/114-300-02G.

Ducts 3 and 4 have a variable along the flow height \( b_1 = 1500 \text{ mm}, b_2 = 500 \text{ mm} \) forming the confusion and diffusion sections with cross sections \( S_1 = b_1 h \) and \( S_2 = b_2 h \) respectively.

Convex sections are oriented on the opposite walls of the PVP room, the diffuser sections are oriented toward its center.

The wind airflow enters the PVP through the confusing section of the duct, increases its speed at the diffuser, loses kinetic energy, providing rotation of the wind wheel, and leaves the PVP, with a lower speed, passing first through the diffuser, and then through the confusing of the second duct. The lines of airflow are marked with arrows in Fig. 2. Such a layout of the equipment allows for the simultaneous operation of two wind wheels and an electric generator with a capacity almost twice as high as the capacity of one wind wheel. At the same time, the wind turbine operates regardless of the change of wind direction to the opposite one. From the continuity equation (Bernoulli) for incompressible fluids, it follows that with the reduction of channel cross-sections the speed of fluid flow through it grows proportionally

\[
s_1 V_I = s_{II} V_{II} = \text{const}.
\]

Narrowing of the channel in \( b_1/b_2 = 2 \) times accordingly increases the speed of the air jet coming to the impeller of the wind wheel. At height of the channel \( b \), its flow cross-section is equal to \( s = b h \). Reduction of channel cross-section at the confuse is achieved by decreasing the height from \( b_1 = 1.5 \text{ m} \) at the inlet to \( b_2 = 0.8 \text{ m} \) at the outlet. Then the degree of narrowing of the confuse (respectively the degree of expansion of the diffuser) is equal:

\[
n = \frac{s_{II}}{s_I} = \frac{b_2}{b_1} = 1.875.
\]

From equality (3) it follows that the flow velocity coming to the wind wheel is equal:

\[
V_{II} = \frac{b_1}{b_2} V_I = 1.875 V_I.
\]
Calculation by formula (2) of the power of the wind turbine, without taking into account losses on local resistances in the duct, gives the outcomes presented in Table 3. It also shows the values of volumetric air flow rate at the outlet section of the confused:

\[ Q = Vbh. \]  

(4)

The cone and the diffuser have the same length l and the same flow contraction (expansion) angles, denoted by \( \alpha \).

**Table 3.** Calculated power of wind turbine, without taking into account losses in ducts.

<table>
<thead>
<tr>
<th>The Wind</th>
<th>Speed of undisturbed flow, ( V_1 ), m/s</th>
<th>Flow velocity on the wind wheel, ( V_{Ip} ), m/s</th>
<th>Volume flow rate, Q, m³/s</th>
<th>The capacity of one unit, P, kW</th>
</tr>
</thead>
<tbody>
<tr>
<td>Weak</td>
<td>5</td>
<td>9.735</td>
<td>1.947</td>
<td>0.70</td>
</tr>
<tr>
<td>Moderate</td>
<td>10</td>
<td>28.12</td>
<td>5.841</td>
<td>16.835</td>
</tr>
<tr>
<td>Strong</td>
<td>15</td>
<td>37.5</td>
<td>9.735</td>
<td>39.92</td>
</tr>
</tbody>
</table>

At the length of the confuse \( l=2 \)m, the angle of flow contraction in it is \( \alpha=20^\circ \), in which case \( \sin \frac{\alpha}{2} = 0.165 \)

Convexity angles larger than \( \alpha=8^\circ \) outcome in flow detachment from the internal walls of the diffuser, which leads to a sharp increase in the duct resistance coefficient. This value for the confuse and diffuser is calculated by the formulas:

\[ \xi_{kf} = \frac{\chi}{8 \sin \frac{\alpha}{2}} \frac{n^2-1}{n^2}. \]  

(5)

\[ \xi_{df} = \frac{\chi}{8 \sin \frac{\alpha}{2}} \frac{n^2-1}{n^2} + \sin \alpha \cdot \frac{(n-1)^2}{n}. \]  

(6)

Here \( \chi \) is the dimensionless coefficient of hydraulic friction of the duct (Darcy coefficient), expressed through the Reynolds number \( Re \):

\[ Re = \frac{4Vbh}{\nu(b+2h)} \]  

(7)

\( \nu \) is the kinematic viscosity of air - a value depending on its temperature, m²/s. For the temperature range of -20...-30°C we can assume that \( \nu = 1.16 \times 10^{-5} \) m²/s.

For laminar and turbulent flow regimes, the Darcy coefficient \( \lambda \) is expressed as equalities, respectively:

\[ \chi = \frac{64}{Re}. \]  

\( \frac{0.3164}{Re^{0.25}} \)  

(8)

Based on the geometric parameters of the duct \( b=1 \)m, \( h=0.25 \)m, the flow velocity \( V=15 \)m / s, and the kinematic viscosity of air \( \nu=1.6 \times 10^{-5} \)m² / s, we find by formula (7), that the Reynolds number \( Re = 625000 \).

Such large values of Reynolds number corresponding to the turbulent flow regime, therefore, to find the Darcy coefficient, we must use the second of the formulas (8):

\[ \chi = \frac{0.3164}{Re^{0.25}} = \frac{0.3164}{28.1} = 0.0112. \]  

(9)

Substituting \( \chi=0.0112 \) and \( n =1.875 \) into equations (5) and (6) we obtain
The total local loss of velocity head $\Delta p$ (Pa) at the confuse and diffuser of the duct is as follows:

$$\Delta p = (\xi_{kf} + \xi_{df}) \frac{\rho g V^2}{2}.$$  

The power of loss $N$, W, in the duct is equal:

$$N = Q \Delta p = VS \Delta p,$$  

Where: $Q$-volume flow rate of air passing through the duct, m$^3$/s.

The outcomes of calculations of local losses in the duct, are presented in Table 4.

<table>
<thead>
<tr>
<th>Flow rate $V_{lp}$, $m^3/s$</th>
<th>Rate of head loss $\Delta p$, Pa</th>
<th>Volume flow rate $Q$, $m^3/s$</th>
<th>Loss of power $N$, W</th>
<th>Relative power losses on $N$</th>
</tr>
</thead>
<tbody>
<tr>
<td>9.735</td>
<td>13.93</td>
<td>1.947</td>
<td>0.027</td>
<td>3.5%</td>
</tr>
<tr>
<td>25.12</td>
<td>92.76</td>
<td>5.841</td>
<td>0.542</td>
<td>4%</td>
</tr>
<tr>
<td>37.5</td>
<td>206.72</td>
<td>9.735</td>
<td>2.01</td>
<td>5%</td>
</tr>
</tbody>
</table>

From the last columns of Table 4 and Table 3, we can see that in the duct with confusion and diffusor sections, having a taper angle $\alpha=2.0$ degrees, the hydraulic resistance to the airflow, reduces the efficiency of the duct by 4...5%. Consequently, power from 16.3 kW to 37.5 kW is achievable.

Let's assume a priori that the electric power of the electric generator of WPP is 10 kW and half of this power feeds the fan installed at the outlet of cooling tube 2, forcibly creating in it the air current. In wintertime, this air has a temperature $t_g \approx -20^\circ C$, which is enough for ground 1. In the spring and summer periods, the air should be additionally cooled. For this purpose, it is proposed to use a small-sized refrigeration unit of absorption type, which has no moving elements and can work autonomously.

The refrigeration unit is also mounted on the standard platform of the freight car together with the VCO and the PVP. The unit contains a heat accumulator filled with thermal storage material (TAM), a heat exchanger filled with cold storage material (HLW), absorption cooling unit (AHU), which provides cooling for the HLW and TAM heating systems.

As TAM, technical lignite wax (melting point+89$^\circ$C), or Locke's mixture is preferred ($CH_3COONa\cdot3H_2O$ (50%)$+Na_2S_2O_3\cdot5H_2O$ (45%)$+glycerin$ (3.5%)$+CaCl_2$ (1,5%),

having high specific heat of melting (3120 kJ/kg, at melting temperature$+60^\circ$C) By absorbing heat these substances melt, and coming back to the solid state release the absorbed heat. The task of continuous maintenance of the negative temperature of XLAS in the exchanger regardless of the ambient temperature is solved by the fact that the heat accumulator with TAM and the exchanger with XLAS have a thermal connection to each other through the cooling elements designed as an absorption cooling unit (AHU). The unit generates cold without the use of moving mechanical elements and can operate autonomously without the involvement of service personnel.

The working substance in an AXA can be, for example, a 30...45% lithium bromide (LiBr) solution. The cooling element consists of a metal barrel equipped with a sealed heat pipe. It is filled with a two-phase working substance, which is a mixture of two components, with sharply differing boiling points. The component boiling at the lower temperature is the refrigerant, the second component, called the absorbent, can absorb the refrigerant. The
The principle of AXA operation is based on obtaining negative temperatures (down to -5...-7°C) in the refrigerant evaporation chamber [15,16].

In the cooling element, the evaporation chamber is fixed in a heat accumulator and, depending on the choice of TAM, can be maintained at temperatures close to plus 90°C. The working substance is contained in it at high pressure, maintained by continuous evaporation of water from the solution and its transition to a gaseous state. The AXA absorber chamber is placed in a heat exchanger filled with HLAS. In it, a lithium bromide solution (a very strong water absorbent) continuously absorbs steam and maintains a low pressure (about 2 m bar) inside the chamber. The evaporation of water takes place practically in a vacuum and is accompanied by cooling of the absorber chamber walls, which causes cooling of the XLAS. The solution of lithium bromide, absorbing water, turns into a diluted solution or rises to the chamber-evaporator on a wick, where it evaporates on the hot walls of the chamber or drains by gravity depending on the mutual arrangement of elements. Concentrated lithium bromide solution through a tube with a hydraulic gate, returns to the chamber-absorber, ensuring the continuity of the process.

In spring and summer, the TAM is heated by the heat from the wind turbine. At the same time all cooling elements start to generate cold continuously, providing a negative temperature of HLAS placed in the cooling tube buried in the ground (Fig.2). At the same time, the system generates a gravity flow of liquid coolant into the vessel filled with TAM. At the same time, the flow of coolant through the heat exchanger is accompanied by its cooling to negative temperatures. The achieved positive effect is reduced to the year-round maintenance of permafrost soils 1 in the area of deepening of the cooling tube 2. For a mathematical description of the thermal process in the cooled ground 1, let us consider, first of all, the flow of cold air through pipe 2.

In the device two air flows are considered, the first flow, which has a high-velocity \( V \), goes to the wind wheel, and provides its rotation, together with the rotor of the electric generator. Unit of wind wheel and electric generator is marked with item 8 in fig.1. The second flow, having low velocity \( u \), is created by extraction unit 6 (fans, powered by the electric generator 8), passes through the pipes 2 and cools the ground 1. Small values of velocity \( u \) allow cooling of the air in the cooling unit.

Let's assume that the x-axis is directed along the flow along the pipe axis, the kinematic viscosity of air \( \nu \) remains constant, and the flow occurs in laminar mode with an average velocity \( \bar{u} \). It is due to the difference of pressures \( p_1-p_2 \) at the pipe inlet and outlet. For the convenience of calculations, the temperature reference is made from the atmospheric air temperature \( t_g \) and the excess temperature of the pipe wall \( \theta=t_{st}-t_g \) is introduced. The distribution of air temperature along the tube radius changes with distance from the tube inlet and is described by the function

\[
\theta(x,r) = t_{st} - t_g
\]

The radial distribution of velocities in a flow moving in a circular tube of radius \( R \) is described by the Gagain-Poiseuille equation:

\[
u(r) = 2\bar{u} \left[ 1 - \left( \frac{r}{R} \right)^2 \right] . \tag{12}
\]

That said:

\[
\bar{u} = \frac{(p_1-p_2)R^2}{8\nu \rho \mu} \tag{13}
\]
In the steady-state flow regime, when the average air velocity in the pipe remains constant, the differential equation of heat conduction with cylindrical symmetry of the problem is written as follows:

\[
\frac{\partial^2 \theta}{\partial r^2} + \frac{1}{r} \frac{\partial \theta}{\partial r} = \frac{2 \hat{u}}{a_g} \left( 1 - \left( \frac{r}{R} \right)^2 \right) \frac{\partial \theta}{\partial x}
\]  

(14)

Here \( a_g \) is the thermal conductivity of the coolant (atmospheric air)

\[ a_g = \frac{\lambda_g}{c_g \rho_g} \approx (1.74\ldots1.38) \cdot 10^{-5} \text{m}^2/\text{s} \]  

(15)

It is determined by the coefficients of its thermal conductivity \( \lambda_g \), specific heat capacity \( C_g \), and density \( \rho_g \). The value indicated in (15) corresponds to the temperature interval \( t_g=-10^\circ\text{C}....-40^\circ\text{C} \). Equation (14) is considered under two boundary conditions: on the pipe wall and in the initial cross-section of the pipe:

\[
\theta|_{r=0, x>0} = 0; \; \theta|_{r=R, x=0} = 0.
\]  

(16)

The solution of equation (14) under boundary conditions (16) will be a series including the Bessel functions of the first order of zero and first order \( J_0(\mu_i \cdot r/R) \) and \( J_1(\mu_i) \) [8]:

\[
\theta(x,r) = \theta_0 \sum_{i=1}^{\infty} \frac{1}{\mu_i J_1(\mu_i)} J_0 \left( \mu_i \frac{r}{R} \right) \exp \left\{ -\frac{x}{2R} \sqrt{\text{Pe}^2 \left[ 1 - \left( \frac{r}{R} \right)^2 \right]^2 + 4\mu_i^2} - \text{Pe} \left[ 1 - \left( \frac{r}{R} \right)^2 \right] \right\}.
\]  

(17)

Here \( \theta_0 \) is the air temperature at the inlet of pipe 2 (Fig. 1), i.e., after its passage through the cooling unit.

\( \text{Re} \) is the dimensionless Peclet criterion determined by the average flow velocity:

\[ \text{Pe} = \frac{2R \hat{u}}{a_g}. \]  

(18)

\( J_0(\mu_i \cdot r/R) \) Bessel function of the first kind of zero order, \( J_1(\mu_i) \) the derivative of this function

\[ J_1(x) = -J'_0(x) \]

Numerical values of functions are presented in tables [21], both functions have innumerable rapidly increasing roots \( \mu_i \). In calculations we usually use the first three to four roots and extremums of functions shown in Table 5.
When \( r=0 \), the Bessel function of order zero is equal to one \( J_0(\mu_i r/R) = 1 \) and expression (17) will be simplified:

\[
\theta(x, r) = \theta_0 \sum_{i=1}^{\infty} \frac{2}{\mu_i J_1(\mu_i)} \cdot \exp \left\{ -\frac{x}{4R} \sqrt{1 + \left(\frac{4\mu_i}{Pe}\right)^2} - 1 \right\}.
\]  (19)

Pickle criterion (18) due to small air thermal conductivity (15), quite large in value (\( Re \approx 5000 \)), so the second term under the radical sign inequality (17) can be neglected. In this case, equality (19) will become even simpler:

\[
\frac{\theta(x, R)}{\theta_0} \approx \sum_{i=1}^{\infty} \frac{2}{\mu_i J_1(\mu_i)} \cdot e^{-\frac{x}{4R}}.
\]  (20)

The expression (20) shows the relative temperature change, in fractions of the initial difference between the temperatures of the pipe wall and the coolant flowing inside the pipe. As \( x \) increases, the wall cools down, and the coolant heats up. The sections of the pipe close to the beginning of the pipe will cool down to the temperature of the pumped air, and then more and more distant sections will cool down.

Pipe 2 is considered a heat sink with a capacity of \( q_l \), W/m. This value is determined by the known formula used to calculate the heat transfer of pipes in the ground:

\[
q_l = \frac{2\pi \lambda_{gr}(t_{tr}-t_{gr})}{\ln \left[ \frac{h}{R} \left( \sqrt{\frac{k}{R}} \right)^{-1} \right]}.
\]  (21)

Here \( \lambda_{gr} \) is the ground thermal conductivity coefficient, W/m°C; \( t_{gr} \) is the ground temperature near the cooling pipe, at a depth of \( h \), then is the ground temperature at the ground surface °C; \( h \) is the distance from the ground surface to the pipe axis, m. At \( R = 0.15 \text{m} \), and \( h = 1.5 \text{m} \) \( \left( \frac{h}{R} \right)^2 = 100 >= 1 \) so neglecting the one in the denominator, we simplify the last equality,

\[
q_l = \frac{2\pi \lambda_{gr}(t_{gr}-t_0)}{h n \left[ \frac{2h}{R} \right]^{-1}},
\]  (22)

Setting the temperatures of the pipe walls and the ground surface \( t_r = -20°C \) and \( t_0 = 0°C \), with the given values of \( d \), \( h \) and \( \lambda \) we obtain \( q_l = 56...63 \) W/m. This cold performance is provided by the airflow moving along pipe 2. The speed of this flow is determined by the formula:

\[
u = \frac{q_l}{2\rho_{gr} c_{gr}(t_0-t_{tr})},
\]  (23)
Substituting the density values $\rho_g=1,2 \text{ kg/m}^3$ and specific heat capacity $C_g=10^3 \text{ J/kg} \cdot \text{°C}$, $T_0=-20^\circ \text{C}$, and the value of $q_l$ from equality (22), we obtain a sufficiently small required airflow velocity $V\approx0.1 \text{ m/s}$. This speed can be provided by the fan at the pipe outlet (Fig.1) even if its length exceeds 1000 m.

3 Analysis of outcomes

The temperature distribution in the ground in the vicinity of one cooling tube is described by the equality [13]:

$$\theta(x, y) = \frac{1}{2\pi \lambda_{gr}} q_l \ln \frac{x^2 + (y_0 + y)^2}{x^2 + (y_0 - y)^2},$$

where $x$ and $y$ are the distances from the pipe axis in the horizontal and vertical directions, $y_0$ is the distance from the pipe axis to the ground surface, m.

The thermal resistance of the soil mass surrounding the pipe system shown in Fig. 1 is determined by the well-known formula [14]:

$$\frac{h}{\lambda_{gr}} = \frac{1}{2\pi \lambda_{gr}} \ln \left\{ \left[ \frac{4h}{2R} \right] \cdot \left[ 1 + \left( \frac{2h}{s} \right)^2 \right] \cdot \left[ 1 + \frac{1}{4} \left( \frac{2h}{s} \right)^2 \right] \right\}. \tag{24}$$

where $s$ is the distance between the ages of cooling tubes 2 (Fig.1).

The depth of laying of pipes for reasons of operating conditions should not be less than 1-1.5 m, so for the calculations, it is advisable to use the three upper curves, for values $h/2R=3; 4$ and 5. The authors set the conditions $h/2R=5$ and $s/2R=10$. In this case, the thermal resistance of the groundmass to the cold flow is equal to $h/\lambda_{gr}=0.68^\circ \text{C/W \cdot m}$.

The first factor in square brackets of the right part of (22) corresponds to the thermal resistance of the massif to the cold flow created by one pipe A, the other two-to the change in the thermal resistance of the massif due to the presence of two symmetrically located pipes B and C in it. Fig. 3 shows the thermal resistance of the ground with a block of cooling pipes A, B, and C at ground thermal conductivity $\lambda_{gr}=1,123 \text{ W/m} \cdot ^\circ \text{C}$, calculated according to formula (2) for different $h/d$ and $s/2R$. It can be seen that it decreases with increasing distance between pipes $s/2R$ and increases with depth $h/2R$.

![Fig. 4. a). Variation of thermal resistance of the wet soil mass to the cold flow created by three parallel pipes. b) Temperature distribution in the cooled soil mass.](image-url)
The temperature distribution in the continuous soil mass in the vicinity of the cooling tube block A, B and C is determined by the equality [14]:

$$\frac{t(x,y)-t_0}{t_{tr}-t_0} = \frac{\ln \left[ \frac{x^2+(h+y)^2}{x^2+(h-y)^2} \cdot \left( \frac{(x-s)^2+(h+y)^2}{(x-s)^2+(h-y)^2} \cdot \frac{(x+s)^2+(h+y)^2}{(x+s)^2+(h-y)^2} \right) \right]}{2 \ln \left( \frac{2R}{s} \cdot \frac{1}{1+\left( \frac{x}{s} \right)^2} \right)}.$$  (25)

Here x and y are distances from the axis of the central pipe in the horizontal and vertical directions, respectively, s is the distance between the axes of the parallel laid cooling pipes, and h is the distance from the pipe axis to the ground surface (Fig.1). The outcomes of calculations according to this formula are presented in fig.4b.

The thermal conductivity coefficient of clay melted and frozen soils are in the range $\lambda_{gr}$=1,123-1,384 W/m·°C. Setting the values of pipe diameter $2R$=0.3m, depth of burial $h$=1.5m, and the distance between the axes of the pipes $s=2m$, we obtain $s/2R=9$ and $h/2R=5$. By formula (24) it is easy to find that the thermal resistance of the array $h/\lambda_{gr}$=0.641-0.520 °C/W.

Calculations according to formula (25) show that when cold air with temperature $t_g$=−20°C passes through tubes 2 during 60-90 days, the whole ground up to the surface of its interface with snow layer 6 is cooled to temperatures not higher than $t$=−17°C. Such low ground temperature, plus low thermal conductivity and high reflectivity of snow in layer 6 allow us to assume that the negative temperature of the frozen ground will last for at least 50-60 days, i.e. practically the whole summer.

4 Conclusion

The proposed method and device for thermal stabilization of permafrost soils in the Northern climatic zone of Russia imply their year-round operation. The method is designed for its implementation in the area of construction and operation of large engineering structures-railroads and highways, port facilities, etc., actively constructed on the Yamal Peninsula, where due to the global climate change of the Earth higher summer temperatures are registered. The permafrost is thawing to a great depth, making it very difficult to use road and construction equipment. The method of thermal stabilization of soils in the area of road construction and operation can only be implemented using natural energy sources, and the device based on it, operates autonomously, with a minimum staff of service personnel.

References

12. V.B. Gurevich et al, Port hydraulic structures (Transport, M., 1992)
15. V.S. Martynovsky, Thermodynamic characteristics of cycles of thermal and refrigeration machines (M.-L., 1952)
16. E.I. Mikulina et al, Low temperature technology (M., 1975)
17. P.P. Yushkov, Bessel functions and their applications to cylinder cooling problems (Publishing House of the Academy of Sciences of the BSSR, Minsk, 1962)