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NUMERICAL ANALYSIS OF HEAT LOSSES BY MAIN HEAT PIPELINES UNDER CONDITIONS OF COMPLETE OR PARTIAL FLOODING

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A two-dimensional problem of the nonstationary temperature field of a main heat pipeline and of its environment under conditions of complete or partial flooding has been solved numerically. At the boundary of the contact between the outer surface of the pipeline and water the presence of evaporation was taken into account. It has been revealed that under typical operating conditions of main heat-supply systems the flooding of the heat pipeline channel leads to a more than 5.5-fold increase in heat losses. It is shown that in order to estimate the heat losses one may use a one-dimensional model of the system considered. It has been established that for heat pipelines operating under conditions of flooding, the domain of solution of the problem can be bounded by the outer boundary of the pipeline.

Introduction. An analysis of the quantities of transport losses by heat pipelines under both regular and irregular operating conditions is a very urgent problem due to the necessity of obtaining reliable and accurate data on heat losses in heat-supply systems, since lately numerous inadequate estimates have appeared [1].

The current interest in the problem considered is due to a number of reasons, the basic of which are [1]: higher requirements on the efficiency of heat supply; activation of competition with decentralized means of heat supply; increase in heat energy supply and transportation rates; increase in the role of domestic appliances for keeping heat consumption records; the necessity of diagnosing the functioning state of pilepines, and carrying out works to increase the heat-supply system reliability.

Of separate interest is the development of the technique of determining heat losses by main pipelines operating under conditions of flooding.

The ways of water ingress into the channels of heat systems can be subdivided into two groups:

1) the flooding of heat pipeline routes is connected with the high permeability to water of ferroconcrete elements of the channel because of the nonhermetic sealing of the joints of its walls and covers [2];

2) water leakage [3], breakthrough of pipelines, as well as breakdowns in water-supply and water-removal systems — they all usually lead to the flooding of the channels of heat-supply systems.

The presence of water in the cavities available in the channels of heat-supply systems (Fig. 1) causes a change in the mechanism of heat exchange between the outer surface of a heat pipeline and environment and a substantial increase in heat-energy losses. As noted previously [1], the only technique used at the present time to determine heat losses [4] does not take into account the actual mechanisms of heat exchange between the systems considered.

The aim of the present work is mathematical simulation of heat transfer in the pipeline wall and a numerical analysis of the magnitude of heat losses from its surface under the conditions of flooding the heat pipeline channel with water.

Statement of the Problem. Two variants of the problem statement are being considered:

1. A "conductive" model. Here, the problem of heat conduction for a three-layered cylinder (pipe wall–insulation layer–water layer, Fig. 1) under inhomogeneous conditions of heat transfer on its outer contour is being solved. On the inner and outer boundaries of the system considered the boundary conditions of the first kind are specified. It is considered that on the outer boundary of the pipeline heat is transferred only by heat conduction, as well as in the layers of water and air. This assumption is made to reveal the scales of the influence of convective heat transfer on the intensity of losses of heat energy during its transportation.

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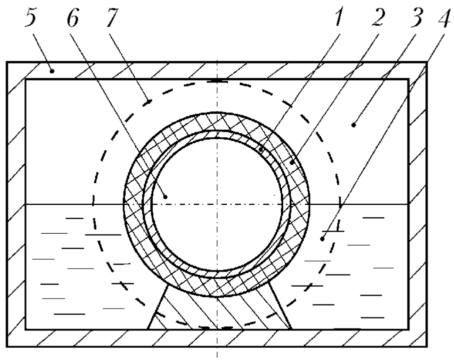

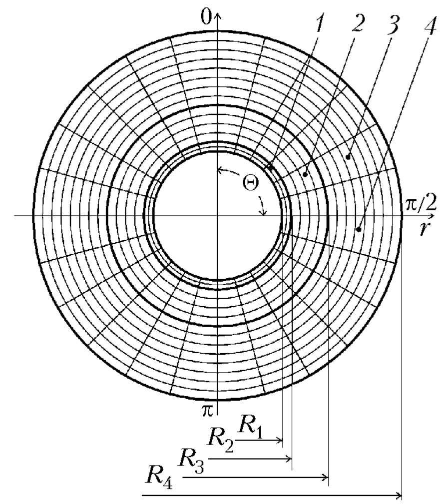

Fig. 1. Schematic of the cross section of the heat-supply system channel under the conditions of partial flooding and the boundary of the solution domain of the first problem: 1) pipe wall; 2) insulation; 3) air layer; 4) water layer; 5) channel casing; 6) heat-transfer agent; 7) outer boundary of the solution domain.

Fig. 2. Geometry of the solution domain: 1) pipe wall; 2) insulation; 3) air layer; 4) water layer.

2. A "convective" model. The problem of heat conduction for a two-layered cylinder (pipe wall–insulation layer) under inhomogeneous conditions of heat transfer on its outer contour is being solved. On the inner boundary of the system considered, boundary conditions of the first kind are specified, with boundary condition of the third kind being employed on the outer boundary.

In both the first and the second model, the process of evaporation on the boundary of contact of the heat pipeline surface with water is taken into account.

Figure 1 shows the scheme of the cross section of the heat pipeline channel under the conditions of partial flooding. The dashed line 7 (Fig. 1) shows the outer boundary of the solution domain. The problem has been solved with allowance for the following basic assumptions: 1) the conditions of perfect thermal contact are satisfied at the boundaries of the contact between the layers; 2) the thermophysical characteristics of the layers (pipe wall, insulation, water, air) are constant and known quantities; 3) the processes of water diffusion in the insulation layer as well as filtration of evaporation products in the insulation and water are not analyzed; 4) the process of heat transfer in the heat-transfer agent is not considered.

Mathematical Model. The geometry of the solution domain is given in Fig. 2. Both problems were solved in a cylindrical coordinate system, whose origin is at the symmetry axis of the pipe. In this connection, a circle was selected as the outer boundary of the solution domain of the first problem (see Fig. 1). The use of a rectangular boundary of the solution domain leads to unjustified complication of the realization of the algorithm of temperature field calculation for the first variant of the model of the system considered. Since both problems are axisymmetric, only half of the solution domain was considered in the range of change of the angular coordinate Θ: $0 \le \Theta \le \pi$.

The system of equations of heat conduction for the first ("conductive") model in the solution domain considered (Fig. 2) has the following form:

$$\frac{\partial T_{\mathrm{w.p}}}{\partial \tau} = a_{\mathrm{w.p}} \left(\frac{\partial^2 T_{\mathrm{w.p}}}{\partial r^2} + \frac{1}{r} \frac{\partial T_{\mathrm{w.p}}}{\partial r} + \frac{1}{r^2} \frac{\partial^2 T_{\mathrm{w.p}}}{\partial \Theta^2} \right), \quad R_1 \le r < R_2, \quad 0 \le \Theta \le \pi ; \tag{1}$$

$$\frac{\partial T_{\mathrm{in}}}{\partial \tau} = a_{\mathrm{in}} \left(\frac{\partial^2 T_{\mathrm{in}}}{\partial r^2} + \frac{1}{r} \frac{\partial T_{\mathrm{in}}}{\partial r} + \frac{1}{r^2} \frac{\partial^2 T_{\mathrm{in}}}{\partial \Theta^2} \right), \quad R_2 < r < R_3, \quad 0 \le \Theta \le \pi ; \tag{2}$$

$$\frac{\partial T_{\text{air}}}{\partial \tau} = a_{\text{air}} \left(\frac{\partial^2 T_{\text{air}}}{\partial r^2} + \frac{1}{r} \frac{\partial T_{\text{air}}}{\partial r} + \frac{1}{r^2} \frac{\partial^2 T_{\text{air}}}{\partial \Theta^2} \right), \quad R_3 < r \le R_4 , \quad 0 \le \Theta < \frac{\pi}{2} ; \tag{3}$$

$$\frac{\partial T_{\text{w}}}{\partial \tau} = a_{\text{w}} \left(\frac{\partial^2 T_{\text{w}}}{\partial r^2} + \frac{1}{r} \frac{\partial T_{\text{w}}}{\partial r} + \frac{1}{r^2} \frac{\partial^2 T_{\text{w}}}{\partial \Theta^2} \right), \quad R_3 < r \le R_4 , \quad \frac{\pi}{2} < \Theta \le \pi . \tag{4}$$

The initial conditions are

$$\tau = 0 , \quad R_1 \le r \le R_4 , \quad 0 \le \Theta \le \pi , \quad T_{\text{w.p}} = T_{\text{in}} = T_{\text{air}} = T_{\text{w}} = T_0 = \text{const} ; \tag{5}$$

the boundary conditions are

$$\tau > 0 , \quad r = R_1, \quad 0 \le \Theta \le \pi, \quad T_{\text{w.p}} = T_{\text{s1}} = \text{const} ; \tag{6}$$

$$\tau > 0 , \quad r = R_2 , \quad 0 \le \Theta \le \pi , \quad -\lambda_{\text{w.p}} \frac{\partial T_{\text{w.p}}}{\partial r} = -\lambda_{\text{in}} \frac{\partial T_{\text{in}}}{\partial r} , \quad T_{\text{w.p}} = T_{\text{in}} ; \tag{7}$$

$$\tau > 0 , \quad r = R_3 , \quad 0 \le \Theta < \frac{\pi}{2} , \quad -\lambda_{\text{in}} \frac{\partial T_{\text{in}}}{\partial r} = -\lambda_{\text{air}} \frac{\partial T_{\text{air}}}{\partial r} , \quad T_{\text{in}} = T_{\text{air}} ; \tag{8}$$

$$\tau > 0 , \quad r = R_3 , \quad \frac{\pi}{2} < \Theta \le \pi , \quad -\lambda_{\text{in}} \frac{\partial T_{\text{in}}}{\partial r} = -\lambda_{\text{w}} \frac{\partial T_{\text{w}}}{\partial r} - QW , \quad T_{\text{in}} = T_{\text{w}} ; \tag{9}$$

$$\tau > 0 , \quad r = R_4 , \quad 0 \le \Theta < \frac{\pi}{2} , \quad T_{\text{air}} = T_{\text{s2}} = \text{const} ; \tag{10}$$

$$\tau > 0 , \quad r = R_4 , \quad \frac{\pi}{2} < \Theta \le \pi , \quad T_{\text{w}} = T_{\text{s2}} = \text{const} ; \tag{11}$$

$$\tau > 0 , \quad R_1 \le r < R_2, \quad \Theta = 0, \quad \frac{\partial T_{\text{w.p}}}{\partial \Theta} = 0 ; \tag{12}$$

$$\tau > 0 , \quad R_1 \le r < R_2 , \quad \Theta = \pi , \quad \frac{\partial T_{\text{w.p}}}{\partial \Theta} = 0 ; \tag{13}$$

$$\tau > 0 , \quad R_2 < r < R_3 , \quad \Theta = 0 , \quad \frac{\partial T_{\text{in}}}{\partial \Theta} = 0 ; \tag{14}$$

$$\tau > 0 , \quad R_2 < r < R_3 , \quad \Theta = \pi , \quad \frac{\partial T_{\text{in}}}{\partial \Theta} = 0 ; \tag{15}$$

$$\tau > 0\,,\quad R_3 < r \le R_4\,,\quad \Theta = 0\,,\quad \frac{\partial T_{\rm air}}{\partial \Theta} = 0\,; \tag{16}$$

$$\tau > 0\,,\quad R_3 < r \le R_4\,,\quad \Theta = \pi\,,\quad \frac{\partial T_{\rm w}}{\partial \Theta} = 0\,. \tag{17}$$

In the second variant of the model, in view of the exclusion of domains 3 and 4 from consideration (Figs. 1 and 2), boundary conditions (16) and (17) and Eqs. (3) and (4) were also excluded, and boundary conditions (8) and (9) took the form of boundary conditions of the third kind:

$$\tau > 0\,,\quad r = R_3\,,\quad 0 \le \Theta < \frac{\pi}{2}\,,\quad -\lambda_{\rm in}\frac{\partial T_{\rm in}}{\partial r} = \alpha_{\rm air}\,[T_{\rm in}\,(\tau, R_3, \Theta) - T_{\rm sur}]\,; \tag{18}$$

$$\tau > 0\,,\quad r = R_3\,,\quad \frac{\pi}{2} < \Theta \le \pi\,,\quad -\lambda_{\rm in}\frac{\partial T_{\rm in}}{\partial r} = \alpha_{\rm w}\,[T_{\rm in}\,(\tau, R_3, \Theta) - T_{\rm sur}] - QW\,. \tag{19}$$

The mass rate of evaporation was calculated by the formula [5]

$$W = \frac{A\,(P_{\rm sat} - P_{\rm part})}{\sqrt{\dfrac{2\pi R_{\rm g}}{M}\,T_{\rm in}\,(\tau, R_3, \Theta)}}\,. \tag{20}$$

Method of Solution and Initial Data. The system of equations (1)–(20) was solved by the finite-difference method [6] using an iterational implicit difference scheme. The characteristic features of the problem solution were the discontinuity of the thermophysical characteristics at the metal–insulation, insulation–water (air), insulation–air interfaces and the presence of nonlinearity in boundary conditions (9) and (19).

Transition to a new time layer was realized with the aid of two fractional steps by the splitting scheme [7]. At the first of them, heat transfer over the coordinate r was calculated and at the second — by the angle Θ with the use of one-dimensional difference equations over the coordinates r and Θ, respectively. The system of one-dimensional difference equations was solved by the method of pivots [6].

The analysis was carried out for a pipeline with a diameter of nominal bore of 600 mm; the pipeline was manufactured from steel 10 (thickness 9 mm) with thermal insulation from glass wool (70 mm thick). The thickness of the layer of water (air), conditioned by the geometric parameters of standard channels [8], was taken equal to 246 mm. The value of temperature in the considered region at the initial instant was T_0 = 282 K. The temperature of the inner surface of the pipe was equal to $T_{\rm s1}$ = 373 K and of the outer boundary of the solution domain — to $T_{\rm s2}$ = 282 K. The partial pressure in Eq. (20) was determined by analogy with the process of surface evaporation [9]:

$$\psi = \frac{P_{\rm part}}{P_{\rm sat}} = \frac{m_{\rm w}}{m_{\rm w} + m_{\rm vap}}\,.$$

From physical considerations it is clear that $m_{\rm w}$ is much higher than $m_{\rm vap}$. Consequently, for the problem considered we may adopt that $\psi \approx$ 0.999–0.995.

The thermophysical properties of insulation on saturation with moisture were determined from the well-known expressions [10] and the effective coefficient of heat conduction $\lambda_{\rm ef}$ — by the formula

$$\lambda_{\rm ef} = \lambda_{\rm in}\varphi_{\rm in} + \lambda_{\rm w}\varphi_{\rm w}\,.$$

TABLE 1. Thermophysical Properties of Materials

Material	λ, W/(m·K)	C, kJ/(kg·K)	ρ, kg/m^3
Steel 10	57.7	0.466	7860
Glass wool	0.059	0.67	206
Water	0.571	4.2	1000
Air	0.025	1.005	1.27

Table 1 lists the thermophysical properties of: pipeline, insulation, water, and air that were used in the numerical analysis. The average coefficients of heat transfer α for the regimes of natural and forced convection were determined from the dimensionless equations [11]:
in the modes of natural convection

$$10^4 < \mathrm{Gr\,Pr} < 10^9\,,\quad \mathrm{Nu} = 0.47\,(\mathrm{Gr\,Pr})^{1/4}\,,\quad \mathrm{Gr\,Pr} > 10^9\,,\quad \mathrm{Nu} = 0.1\,(\mathrm{Gr\,Pr})^{1/3}\,,$$

in the modes of forced convection

$$5\cdot10^3 < \mathrm{Re} < 5\cdot10^4\,,\quad \mathrm{Nu} = 0.148\,\mathrm{Re}^{0.633}\,,\quad 5\cdot10^4 < \mathrm{Re} < 5\cdot10^5\,,\quad \mathrm{Nu} = 0.43 + 0.0208\,\mathrm{Re}^{0.814}\,\mathrm{Pr}^{0.31}\,.$$

Under the conditions of forced convection, moderate, actually possible speeds (up to 1 m/sec) of motion of the media surrounding the pipeline were considered. According to the estimations made in [12], the average temperature of the environment T_{sur} was taken equal to 296.3 K. Heat losses q_L related to 1 m of the pipeline length L were determined from the expressions:
in the one-dimensional variant of the model

$$q_L = \left(-\lambda_{\mathrm{w}}\frac{\partial T}{\partial r} + QW\right)\frac{2\pi R_3 l}{L}\,,$$

in the two-dimensional variant

$$q_L = \left[\int\limits_0^{\pi/2} -\lambda_{\mathrm{air}}\frac{\partial T_{\mathrm{air}}}{\partial r}\,d\Theta + \int\limits_{\pi/2}^{\pi} -\lambda_{\mathrm{w}}\frac{\partial T_{\mathrm{w}}}{\partial r}\,d\Theta + \int\limits_{\pi/2}^{\pi} QW\,d\Theta\right]\frac{2\pi R_3 l}{L}\,.$$

Results of Investigation. The results of the numerical analysis are listed in Tables 2–6. The analysis was carried out for the time period corresponding to the emergence of the process to a stationary regime. Preliminarily, numerical investigations were carried out using the one-dimensional model of the considered system, the mathematical statement of the problem for which is similar to that of Eqs. (1)–(20). A series of numerical experiments was carried out for typical conditions of system operation. The results of numerical simulation are listed in Tables 2 and 3.

Because of the absence of data on the temperature fields of pipelines operation under the conditions of flooding, estimation of the reliability of the results obtained was based on the compliance with the conditions of energy balance on the boundaries of the solution domain.

The error of the energy balance δ in all the variants of numerical analysis did not exceed 0.49%, which can be considered acceptable in estimating the heat losses from the main pipelines. During the numerical experiments performed emphasis was given to the analysis of the influence exerted by the fraction of moisture in the layer of porous thermal insulation on the intensity of heat losses, as well as to the extent of the influence of water evaporation occurring on the surface of thermal insulation on the intensity of heat removal.

In carrying out investigations, the regimes of both short-duration and long-duration floodings typical of evaporation of heat pipelines were analyzed. The former are characterized by relatively low volumetric fractions of moisture in the thermal insulation layer (0.05–0.2) and the latter — by rather high values of φ_{w} (up to 1.0). In the course of

TABLE 2. Results of Numerical Analysis with Respect to the Parameter ψ for the "Conductive" Model

Regime	One-dimensional model		Two-dimensional model		q_L^*	
	q_L, W/m	δ, %	q_L, W/m	δ, %	Design regime	Normative regime
Normative (Construction Specifications and Regulations 2.04.14–88 [3])	122.00	—	122.00	—	—	1
Design (pipeline is not flooded)	24.35	0.41	24.32	0.41	1	—
Flooded pipeline:						
ψ = 1 (evaporation is ignored)	133.67	0.41	133.15	0.14	5.49	1.10
$\psi = 0.999$	136.98	0.43	136.31	0.34	5.63	1.12
$\psi = 0.997$	143.66	0.46	142.96	0.36	5.90	1.18
$\psi = 0.995$	150.42	0.49	150.00	0.40	6.18	1.23

*Coincide for the one- and two-dimensional models.

TABLE 3. Results of Numerical Analysis with Respect to the Value of Moisture Content of Insulation for the One-Dimensional (δ = 0.41%) and Two-Dimensional (δ = 0.21%) "Conductive" Model

φ_w	q_L, W/m		Δq_L^*	
	One-dimensional model	Two-dimensional model	Design regime	Normative regime
0.05	176.13	175.93	7.23	1.44
0.1	212.21	212.00	8.71	1.74
0.2	270.24	269.81	11.10	2.21
0.4	350.28	349.87	14.38	2.87
0.6	402.86	402.01	17.21	3.30
0.8	440.05	439.87	18.10	3.60
1.0	467.75	466.97	19.20	3.83

*Coincide for the one- and two-dimensional models.

the numerical analysis we considered only stationary regimes of flooding, when the value of φ_w did not vary in time. The estimates show that for actual main heat pipelines more typical are rather long-duration periods of flooding (24 h and more). In such regimes the layer of porous insulation is saturated to high values of φ_w. It is seen from the results obtained that flooding of the heat pipeline channels will lead to an appreciable increase in heat losses, with the fraction of the latter due to evaporation amounting to 2.4–11.1%, and that the increase in the moisture content of the insulation leads to a corresponding increase in the intensity of heat removal from the outer surface of the pipeline.

A comparison of the results obtained by the one-dimensional and two-dimensional models shows that the latter does not yield considerably more accurate results (the difference is less than 0.2%). Consequently, for the analysis of the amounts of heat losses from the surface of main pipelines one may use the one-dimensional model of the system considered. Here it should be noted that when a pipeline is partially flooded (Fig. 1), intense heat removal from the "wetted" surface of the insulation does not cause any noticeable flow of heat over the angular coordinate from the zone of "dry" insulation to that saturated with moisture. This seems to be due to the fact that the thickness of the steel pipeline casing with high thermal conductivity is small enough, whereas the relatively "thick" insulation layer has low thermal conductivity. Correspondingly, even at temperature drops of several degrees over the angular coordinate the circular heat flux remains very low. At the same time, it would be of interest to analyze in future the processes of moisture diffusion and filtration in both radial and circular directions from zones with an elevated moisture content to those where moisture was absent at the initial time instant. Account for these factors may raise the accuracy of the analysis and of the estimates of the amounts of heat losses.

A comparison of the results of numerical simulation with standardized values of heat losses (Tables 2 and 3) shows that the normative values of heat losses [13] during operation of heat pipelines on the whole are very close in numerical value to the values of q_L obtained in the present work for the conditions of saturation of thermal insulation

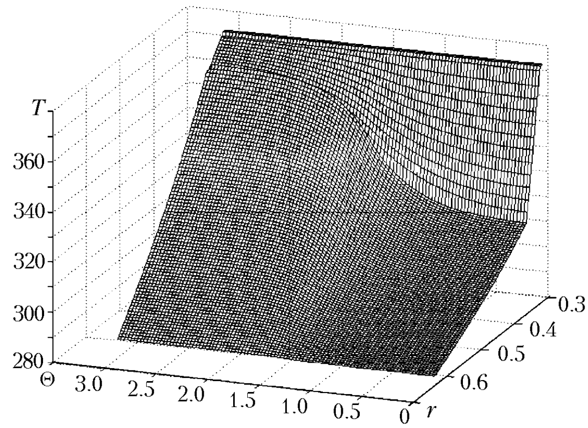

Fig. 3. Temperature field of the considered system of flooding of the pipeline by 50%. T, K; r, m.

TABLE 4. The Value of the Linear Heat Flux Density and Its Deviation from the Design Regime for the "Conductive" Model ($\delta = 0.41\%$)

Flooding, %	q_L, W/m	Δq_L
0 (design regime)	24.35	1
10	35.28	1.45
25	51.68	2.12
50	79.01	3.25
75	106.34	4.37
90	122.74	5.04
100	133.67	5.49

TABLE 5. Results of a Numerical Analysis of an Operating Heat Pipeline in Regime of Convective Heat Transfer ($\delta = 0.06\%$)

Heat transfer regime	Environment	V, m/sec	q_L, W/m	α, W/(m^2·K)
Natural convection	Air	—	110.7	2.75
	Water	—	140.8	222.98
Forced convection	Air	0.5	115.4	3.40
		0.75	120.4	4.40
		1.0	123.5	5.28
	Water	0.5	141.3	1090.2
		0.75	141.3	1516.4
		1.0	141.32	1916.5

with moisture with a relatively low level of saturation (up to 3–5%). With a further increase in the volumetric fraction of water in the layer of porous insulation the magnitudes of heat losses will increase and will increasingly differ from normative values.

Figure 3 shows a typical temperature field of the considered model in a stationary regime with 50% flooding of a heat pipeline.

Table 4 lists the results of numerical experiments depending on the degree of flooding of pipelines (without account for evaporation effect) for the "conductive" model. In the statement of the problem the flooding degree is manifested as a change in the value of the angular coordinate Θ at the air-water interface. Thus, for example, for 25% flooding of the heat supply system channel there corresponds the value $\Theta = 3\pi/4$, for 50% — $\Theta = \pi/2$, and for 75% — $\Theta = \pi/4$ (see Fig. 2). The data obtained allow us to state that the magnitude of heat losses is directly proportional

TABLE 6. Results of a Numerical Analysis with Respect to the Magnitude of the Moisture Content of Insulation for the "Convective" Model

Moisture content	Natural convection		Forced convection					
	q_L, W/m	δ, %	V = 0.5 m/sec		V = 0.75 m/sec		V = 1.0 m/sec	
			q_L, W/m	δ, %	q_L, W/m	δ, %	q_L, W/m	δ, %
0.05	205.71	0.061	206.53	0.061	206.58	0.061	206.62	0.061
0.1	270.34	0.061	271.75	0.061	271.86	0.061	271.91	0.061
0.2	398.97	0.061	402.06	0.061	402.28	0.061	402.40	0.061
0.3	526.44	0.060	532.15	0.060	532.54	0.060	532.76	0.060
0.4	653.71	0.059	662.04	0.059	662.65	0.059	662.98	0.059
0.5	779.83	0.054	791.72	0.056	792.54	0.056	793.06	0.056

to the degree of flooding of heat pipelines. In the limiting regime of flooding, linear heat losses exceed by 5.5 times those foreseen by design conditions.

Table 5 presents the results of a numerical analysis of a heat pipeline operating in the regimes of convective heat transfer. The relative difference between heat losses in the regime of natural convection of water and air amounts to 21.4%, whereas in the regime of forced convection of water and air the difference is 12.6–18.3%. From the data of the table it is seen that under the conditions of flooding the mode of heat transfer is immaterial in determining heat losses, and outer regions 3 and 4 can be excluded from consideration (see Figs. 1 and 2).

Table 6 presents the results of numerical experiments for the "convective" model of the system considered on the value of moisture content in thermal insulation. Here, just as in the first model (Table 3), moistening of thermal insulation leads to a sharp increase in heat losses. Moreover, the heat losses in the regimes of convective heat transfer exceed those observed in the "conductive" model by a factor of 1.17–2.07.

The large influence of the values of φ_w on q_L (Tables 3 and 6) shows that the refinement of the value of φ_w (both average ones and those distributed over the insulation thickness) may lead to a noticeable increase in q_L.

The results of numerical experiments show the advisability of a further analysis of the considered processes within the scope of a more complex problem that would take into account the dynamics of the process of saturation of the layer of porous insulation with water. Moreover, it is possible to take into account not only the flooding of main pipelines but also their dewatering. Here again both local stationary values of q and nonstationary q_L values can further be refined.

It should be noted that according to the results of numerical experiments, saturation of a layer of highly porous (usually the most effective) thermal insulation with water leads to great losses of heat energy as compared to low-porous insulation, which is less efficient in ordinary condition.

Based on the data obtained it can be said that the protection of heat pipelines from flooding with water seems to be one of the most effective measures of reducing the losses of heat energy in its conveyance to users.

CONCLUSIONS

1. The main factor of intensification of the process of heat energy loss on flooding of heat pipelines is the sharp increase in the insulation after it was saturated with moisture.

2. Under certain conditions the role of evaporation effect in determining heat losses from the surface of heat pipelines becomes important.

3. The use of the two-dimensional model substantially does not refine the one-dimensional model; therefore the latter can be used for estimating heat losses by main heat pipelines under the conditions of flooding.

4. When pipelines operate under flooding conditions, the solution domain of the problem may be limited by the outer boundary of the pipeline.

5. In the regimes of convective heat transfer, the heat losses are 1.17–2.07 times higher than those in the conductive model.

6. The results obtained point to the advantage of using the developed model and technique of numerical analysis for estimating the amounts of heat losses by main pipelines operating under flooding conditions.

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NOTATION

A, accommodation coefficient; *a*, thermal diffusivity, m^2/sec; *C*, heat capacity, kJ/(kg·K); Gr, Grashof number; *l*, pipeline length, m; $L = 1$ m; *M*, molecular mass of vapors, kg/mole; *m*, mass fraction; Nu, Nusselt number; *P*, pressure, Pa; Pr, Prandtl number; *Q*, phase transition heat, J/kg; q_L, linear density of a heat flux, W/m; Δq_L, relative deviation of linear density of a heat flux for various regimes; *R*, boundary of solution domain, m; Re, Reynolds number; R_g, gas constant, J/(mole·K); *r*, current radius, m; *T*, temperature, K; *V*, velocity of motion of the medium surrounding pipeline, m/sec; *W*, evaporation rate, $kg/(m^2 \cdot sec)$; α, heat transfer coefficient, $W/(m^2 \cdot K)$; δ, energy balance error, %; φ, moisture content; λ, thermal conductivity, W/(m·K); ρ, density, kg/m^3; Θ, current angle, rad; τ, time, sec; ψ, ratio of partial pressure to saturation pressure. Subscripts: 0, initial time instant; 1, 2, 3, 4, numbers of the boundaries of regions; w, water; air, air; in, insulation; sur, surrounding; s1, inner surface; s2, outer surface; vap, vapor; sat, saturation; part, partial; w.p, wall of a pipe; ef, effective; un, unit; g, gas.

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