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BUILDING A MODEL OF HEATING AN OIL TANK UNDER THE THERMAL INFLUENCE OF A SPILL FIRE

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The object of this study is the process of liquid combustion in a spill, and the subject of research is the distribution of temperature along the wall and roof of a vertical steel tank that is heated under the thermal influence of a spill fire. The heat balance equation for the wall and roof of the tank with oil product was constructed. The assumption of a small thickness of the wall and roof of the tank relative to its linear dimensions makes it possible to move to two-dimensional differential equations of the parabolic type. The equations take into account the radiative heat exchange with the flame, the environment, the internal space of the tank, as well as the convective heat exchange with the surrounding air, the vapor-air mixture, and the liquid inside the tank.

Using the methods of similarity theory, estimates of the coefficients of convection heat exchange of the outer surface of the tank with the surrounding air and the inner surface with the vapor-air mixture and liquid in the tank in the conditions of free convection were constructed. The application of the finite difference method for solving the heat balance equations has made it possible to derive the temperature distribution on the surface of the tank at an arbitrary moment in time. It is shown that the value of the coefficient of convection heat exchange of the liquid exceeds the corresponding value for the air-vapor mixture by (1÷2) orders of magnitude. As a result, the part of the wall located below the oil product level is heated to a temperature of (80÷230) °C depending on the viscosity of the liquid. This occurs despite the fact that the value of the mutual radiation coefficient reaches its maximum value at the lower part of the wall. From a practical point of view, this means that the part of the wall above the level of the oil product in the tank can reach dangerous temperature values, and it should be cooled first. The constructed model of tank heating also enables determining the limit time for the start of cooling of the tank

Keywords: flammable liquid spill, spill fire, tank heating, heat flow

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1. Introduction

Tank farms are the main place for storing oil and related products in the process of their processing. A significant number of emergencies begin with an emergency spill of flammable liquid [1]. The appearance of an ignition source in the form of an open flame or electrostatic discharge leads to the ignition of liquid vapors. As a result of the accumulation of significant volumes of flammable liquids in a small area, there is a danger of cascading fire spread to nearby technological objects [2] and natural landscapes [3]. According to [4], about 44 % of large-scale fires, in which the "domino" effect was observed, started with a tank fire or a spill fire. According to study [5], about half of the fires in tank farms started due to the appearance of open flames. Another 10 % were due to electrostatic discharges and lightning, others due to operator errors or equipment failure.

Vertical steel tanks are the main way of storing oil and oil products. The danger of thermal effects of fire on them is associated with depressurization of flange connections, loss of strength of steel structures [6], and heating of individual parts of the tank to the self-ignition temperature of liquid vapors stored in it. Heating steel structures to the self-ignition temperature of the liquid turns them into a source of ignition. This can lead to an explosion of the vapor-air mixture in the gas space of the tank or to the occurrence of combustion of vapors at the outlet of breathing devices. Another negative consequence of large-scale fires is the emission of harmful substances into the atmosphere [7]. Spreading over long distances, they significantly affect the state of the air and create danger for the population [8]. Spills of oil and related products, which are not accompanied by fires, lead to soil, underground, and river water pollution [9].

Therefore, it is a relevant task to carry out studies on devising methods for preventing the cascading spread of fire in tank farms.

2. Literature review and problem statement

In [10], global data on 595 fires in tank farms with oil and oil products were studied. Statistically, the probability of the fire spreading to neighboring tanks, the expected number of injured and dead were determined. But this approach does not provide an answer as to whether there is a danger of fire spreading to neighboring tanks under specific conditions and how this danger can be reduced. Paper [11] provides an overview of large-scale accidents associated with the spillage of flammable liquids. Empirical and semi-empirical models of liquid spreading and burning on a solid surface or on a water surface are considered. It is concluded that the rate of burning of spills is lower compared to burning in tanks, but this conclusion is made for spills mainly on smooth surfaces, for example, glass, concrete, etc. In [12], a model of liquid spreading and burning on a horizontal surface was built. The model takes into account the gravitational spreading of the liquid and its burning and makes it possible to determine the dynamics of change in the diameter of the spill. But the influence of the slope of the surface on the shape of the spill remains neglected. In [13], the influence of the angle of inclination of the surface on the process of liquid spreading and burning was experimentally studied. But glass is used as a surface, as a result of which the influence of surface irregularities and impregnation deep into it is ignored. This does not make it possible to use the model for practical calculations.

In [14], there is an overview of pool fire models – situations where the spillage of flammable liquid is limited by obstacles, for example, a collapse around the tank. It is noted that fires of this type account for about 42 % of all fires in reservoir parks. Flame pulsations, specific mass burning rate, flame height, heat transfer by radiation are considered, but the heating of neighboring objects under the thermal influence of fire is neglected.

In [15], the parameters of combustion centers with a diameter of (0.1÷0.4) m were experimentally investigated: mass loss rate, flame height, degree of its blackness, heat radiation density. The disadvantage of this approach is the impossibility of extrapolating the results to larger spills, which are typical for accidents in tank farms. As a result, the application of the results to the study of real fires is complicated. In [16], the random component of the heat flow by radiation from a combustible liquid fire was experimentally investigated. It is shown that the cross-sectional area of the torch is described by the normal distribution law. The mathematical expectation and correlation function of the random process corresponding to the density of the heat flux from the torch were determined. At the same time, the consequences of the thermal effect of the fire on the neighboring tank are not taken into account.

In [17], using computational fluid dynamics (CFD) methods, a fire in several basins was considered simultaneously. Due to the coalescence of the flames, this leads to an increase in the rate of burning of the liquid, an increase in the height of the flame and heat radiation from the fire. But the heating of neighboring tanks under the thermal influence of fire is not considered. In [18], CFD methods were used to calculate safe distances from a flammable liquid spill fire under windy conditions, but the effects of the fire's thermal effects on nearby objects were neglected.

An estimate of the convection component of the heat flux by radiation from a spill fire is reported in [19]. The proposed approach is based on the system of Navier-Stokes equations and takes into account the shape of the spill and the temperature of the flame. However, the consequences of the thermal effects of the fire on the neighboring tanks were not considered. In [20], the thermal effect on steel structures – their loss of strength depending on the temperature – was considered. But the characteristics of the combustion chamber, which lead to a given amount of heat flow, are left out of consideration. In [21], a model of the thermal effect of a fire in a collapse on the part of the tank wall, which is above the level of the liquid poured into it, was built. The temperature distribution along this part of the tank wall and its change over time were found. The model makes it possible to determine which parts of the wall heat up to dangerous temperatures and in what time. The model takes into account radiation and convection heat exchange with the fire and the surrounding environment. At the same time, heat exchange with other parts of the wall and roof remains unaccounted for, both due to heat conduction and due to radiation. The heating of the part of the wall below the level of the oil product was also neglected.

Our review reveals that the prevention of the cascading spread of the fire to the neighboring tanks with oil products requires the determination of the tanks whose surface temperature reaches dangerous values, as well as the limit time for the start of their cooling.

All this gives reason to assert that it is expedient to conduct research aimed at building a model of heating a tank with a petroleum product under the thermal influence of a spilled flammable liquid fire.

3. The aim and objectives of the study

The purpose of our work is to build a model of heating a tank with an oil product under the thermal influence of a fire spilling a flammable liquid, the feature of which is taking into account the radiation and convection heat exchange of the surface of the tank with the fire and the surrounding environment. In practice, this opens up opportunities to identify areas on the wall and roof of the tank that are heated to dangerous temperature values and, as a result, require cooling.

To achieve this goal, the following tasks must be solved:

– to build the heat balance equation for the surface of the tank under fire conditions;

– to determine the convection heat exchange coefficients of the tank wall with gas and liquid media;

– to solve the heat balance equation.

4. The study materials and methods

The object of our study is the process of liquid combustion in spillage, and the subject of research is the temperature distribution along the wall and roof of a vertical steel tank when it is heated under the thermal influence of a fire. The main hypothesis assumes that the temperature distribution on the surface of the tank can be described by a two-dimensional heat conduction equation, which takes into account the radiation and convection heat exchange with the fire and the environment. The main assumptions are uniform distribution of temperature along the thickness of the wall, thermal insulation of the lower edge of the wall, and absence of wind. The heating of a tank with a flammable liquid (petrol A-95) and a combustible liquid (fuel oil) was studied.

To determine the temperature distribution on the surface of the tank, the two-dimensional heat conduction equation was used. Radial and convective components of heat transfer were calculated using the theory of heat transfer. Coefficients of convection heat exchange were estimated by methods of similarity theory. The finite difference method was used to solve the system of partial differential equations. The method was implemented in the Delphi 11 (USA) programming environment.

5. Results of building a model of heating a tank with an oil product under the thermal influence of a fire

5. 1. Construction of heat balance equations for the tank surface under fire conditions

Typical sizes of vertical steel tanks for storage of oil and oil products with a capacity of (0.7÷20) thousand m^3 are (10÷40) m in diameter and (9÷18) m in height. At the same time, the thickness of the wall and roof does not exceed 10 mm, that is, it is 3 orders of magnitude smaller compared to the height and diameter. This makes it possible to consider the temperature to be the same throughout the entire thickness of the roof of the tank. Then the temperature distribution along the tank wall will be described by the two-dimensional heat conduction equation, which in cylindrical coordinates takes the following form:

$$\frac{\partial T_w}{\partial t}=a\left(\frac{\partial^2 T_w}{\partial z^2}+\frac{1}{R^2}\frac{\partial^2 T_w}{\partial \phi^2}\right)+\frac{q_w}{c_s \rho_s \delta_w};$$

$$0<z<H;\ \ 0<\phi<2\pi, \tag{1}$$

where $T_w(\varphi, z, t)$ is the temperature of the tank wall at the point with coordinates (φ, z) at time t; a – coefficient of thermal conductivity of steel; R, H – radius and height of the tank (Fig. 1); c_s, ρ_s – specific heat capacity and specific density of steel; δ_w – thickness of the tank wall; $q_w=q_w(\varphi, z, t)$ – the heat flow density at a given point on the wall of the tank due to heat exchange with the fire, the environment, and the internal space of the tank.

The temperature distribution over the roof of the tank is described by the equation:

$$\frac{\partial T_r}{\partial t}=a\left(\frac{\partial^2 T_r}{\partial r^2}+\frac{1}{r}\frac{\partial T_r}{\partial r}+\frac{1}{r^2}\frac{\partial^2 T_r}{\partial \phi^2}\right)+\frac{q_r}{c_s \rho_s \delta_r};$$

$$0<r<R;\ \ 0<\phi<2\pi, \tag{2}$$

where T_r (r, φ, t) is the temperature of the tank roof at the point with coordinates (r, φ) at time t; δ_r is the thickness of the tank roof; $q_r=q_r$ (r, φ, t) is the heat flow density at a given point on the roof of the tank due to heat exchange with the fire, the environment, and the internal space of the tank.

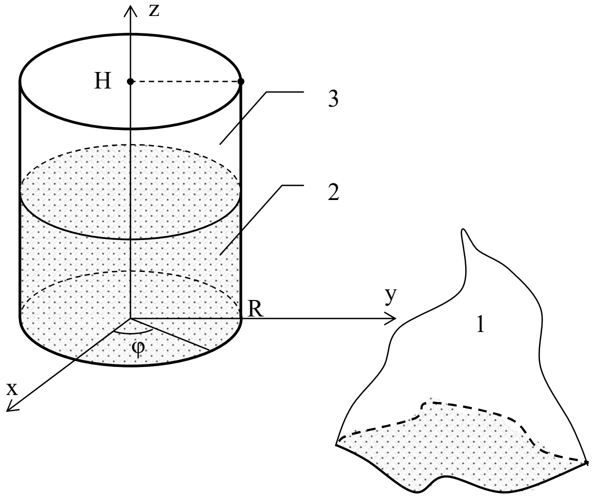

Fig. 1. Choosing the coordinate system when modeling the heating of the tank wall and roof: 1 — flame; 2 — the lower part of the wall in contact with the liquid in the tank; 3 — the upper part of the wall that is not in contact with the liquid in the tank

The condition of equality of wall temperature and roof temperature along their connection determines the boundary condition:

$$T_w(\phi, H, t)=T_r(R, \phi, t). \tag{3}$$

The assumption of thermal insulation of the lower edge of the wall corresponds to the boundary condition of the second kind:

$$\left.\frac{\partial T_w(\phi, z, t)}{\partial z}\right|_{z=0}=0. \tag{4}$$

As an initial condition, it was assumed that the temperatures of the wall and roof of the tank are equal to the ambient temperature:

$$T_w(\phi, z, 0)=T_0;\ \ T_r(r, \phi, 0)=T_0. \tag{5}$$

The density of heat flux to the roof and the upper part of the wall, which is not in contact with the oil product in the tank, is represented by the sum:

$$q=q_1+q_2+q_3+q_4+q_5+q_6+q_7, \tag{6}$$

where q_1 – heat flux density by radiation from the fire; q_2 – heat flow density by radiation into the environment; q_3 – heat flow density by convection heat exchange with the surrounding air and combustion products; q_4 – heat flux density by radiation from the inner surface of the wall through the gas space of the tank; q_5 – heat flux density by radiation from the inner surface of the tank roof; q_6 – heat flux density by radiation from the surface of the liquid; q_7 – heat flow density due to convection heat exchange with the steam-air mixture in the gas space of the tank.

The heat flux density to the lower part of the wall in contact with the oil product in the tank is represented by the sum:

$$q = q_1 + q_2 + q_3 + q_8, \tag{7}$$

where q_8 is the heat flow density due to convection heat exchange with the liquid in the tank.

The heat flux density by radiation from a fire is described by the Stefan-Boltzmann law:

$$q_1 = c_0 \varepsilon_f \varepsilon_w \left(\left(\frac{T_f}{100} \right)^4 - \left(\frac{T}{100} \right)^4 \right) \psi_f, \tag{8}$$

where c_0=5.67 W/($m^2 \cdot K^4$) is constant; ε_f, ε_w are the degrees of blackness of the radiating surface of the flame and the wall of the tank, respectively; T_f is the flame surface temperature; T is the temperature of the tank surface at a given point; ψ_f is the mutual radiation coefficient between the flame and this point on the surface of the tank.

The density of heat flux by radiation into the environment is described by the expression:

$$q_2 = c_0 \varepsilon_w \left(\left(\frac{T_0}{100} \right)^4 - \left(\frac{T}{100} \right)^4 \right) \left(1 - \psi_f \right), \tag{9}$$

where T_0 is the ambient temperature.

The heat flow density due to convection heat exchange with the surrounding air is determined by Newton's law:

$$q_3 = \alpha_3 \left(T_a - T \right), \tag{10}$$

where T_a is the temperature of air masses. These, in particular, can be heated combustion products. α_3 is the coefficient of convection heat exchange with the surrounding air masses.

The heat flux density by radiation from the inner surface of the tank wall is described by the Stefan-Boltzmann law:

$$q_4 = \frac{c_0 \varepsilon_w^2}{\pi} \iint_{S_w} \frac{\cos\phi_w \cos\phi}{r^2} \left(\left(\frac{T_w}{100} \right)^4 - \left(\frac{T}{100} \right)^4 \right) \mathrm{d}S_w, \tag{11}$$

where φ is the angle between the normal vector at a given point on the surface of the tank and the direction vector to the point A_w on the inner surface of the tank wall S_w; φ_w is the angle between the direction vector and the normal vector to the tank wall at the point A_w; T_w is the temperature of the wall at the point A_w.

The heat flux density by radiation from the roof of the tank is described by the expression:

$$q_5 = \frac{c_0 \varepsilon_w^2}{\pi} \iint_{S_r} \frac{\cos\phi_r \cos\phi}{r^2} \left(\left(\frac{T_r}{100} \right)^4 - \left(\frac{T}{100} \right)^4 \right) \mathrm{d}S_r, \tag{12}$$

where φ is the angle between the normal vector at a given point on the surface of the tank and the direction vector to the point B_r on the inner surface of the roof of the tank S_r; φ_r is the angle between the directional vector and the normal vector to the roof of the tank at point B_r; T_r is the temperature of the roof at point B_r.

Heat flux density by radiation from the liquid surface:

$$q_6 = c_0 \varepsilon_f \varepsilon_\ell \left(\left(\frac{T_\ell}{100} \right)^4 - \left(\frac{T}{100} \right)^4 \right) \psi_\ell, \tag{13}$$

where T_ℓ is the temperature of the liquid in the tank; ψ_ℓ is the mutual radiation coefficient between a point on the inner surface of the tank wall and the surface of the liquid. The temperature of the liquid is considered constant throughout its volume due to the leveling of its irregularities due to mixing by convection currents.

Heat flow density due to convection heat exchange with the steam-air mixture in the gas space of the tank:

$$q_7 = \alpha_7 \left(T_m - T \right), \tag{14}$$

where α_7 is the coefficient of convection heat exchange of the inner surface of the wall with the steam-air mixture; T_m is the temperature of the steam-air mixture. The temperature of the mixture is assumed to be the same throughout the entire volume of the gas space since the unevenness of its heating is leveled out by the resulting convection movements.

The heat flux density due to the convection heat exchange of the inner surface of the wall and the liquid in the tank takes the following form:

$$q_8 = \alpha_8 \left(T_\ell - T \right), \tag{15}$$

where α_8 is the coefficient of convection heat exchange between the wall and the liquid in the tank with which it is in contact.

5. 2. Determining convection heat exchange coefficients of the tank wall with gas and liquid media

Convection heat exchange of a vertical wall under conditions of free convection in a gas or liquid environment can be described by the criterion equation:

$$\mathrm{Nu} = C \left(Gr \cdot \mathrm{Pr} \right)^n, \tag{16}$$

where C, n – constants; Nu, Gr, Pr – Nusselt, Grasgoff and Prandtl numbers, respectively:

$$\mathrm{Nu} = \frac{\alpha L}{\lambda}; \tag{17}$$

$$\mathrm{Gr} = \frac{g L^3 \beta \left| T - T_{env} \right|}{\nu^2}; \tag{18}$$

$$\mathrm{Pr} = \frac{\nu}{a}; \tag{19}$$

L – characteristic size; α – coefficient of convection heat exchange; λ, a – coefficients of thermal conductivity and thermal conductivity of a gaseous or liquid medium; g – acceleration of free fall; β – the volumetric expansion coefficient; ν – coefficient of kinematic viscosity, m^2/s; T – temperature of the tank wall; T_{env} – the temperature of the medium in contact with the wall. The values of the constants C and n are taken as:

$$C = 0{,}135; \quad n = 1/3. \tag{20}$$

Substituting expressions (17) to (20) into formula (16) yields:

$$\frac{\alpha L}{\lambda}=0.135\left(\frac{gL^3\beta\left|T-T_{env}\right|}{\nu^2}\cdot\frac{\nu}{a}\right)^{1/3};$$

$$\alpha=0.135\lambda\left(\frac{g\beta\left|T-T_{env}\right|}{\nu a}\right)^{1/3}. \qquad (21)$$

The dependence of the convection heat transfer coefficient on the tank wall temperature under free convection conditions is shown in Fig. 2. The temperature of the gas medium is taken as T_{env}=20 °C.

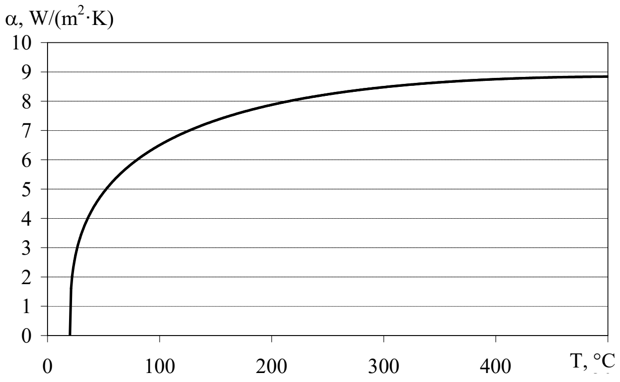

Fig. 2. Value of the coefficient of convection heat exchange of the tank wall with a gas medium under free convection conditions depending on the temperature of the tank wall

The specified dependence holds for the heat exchange of the inner surface of the wall and the roof with the vapor-air mixture in the gas space of the tank, as well as the outer surface of the wall and the roof with the surrounding air in the case of a calm wind.

Fig. 3 shows the dependence of the convection heat exchange coefficient on the temperature difference between the wall and the liquid using the example of gasoline and fuel oil, provided that the liquid temperature is 20 °C.

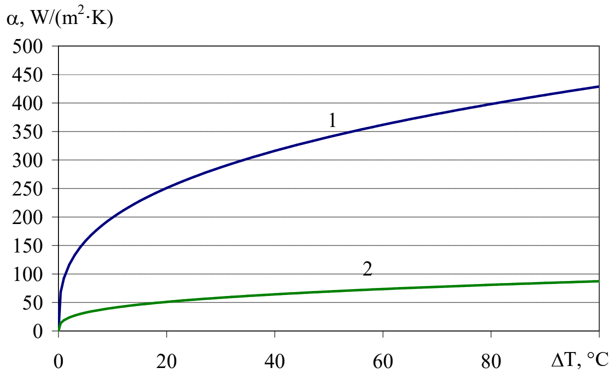

Fig. 3. Dependence of the convection heat exchange coefficient on the temperature difference between the wall and liquid: 1 – gasoline; 2 – fuel oil

A comparison of the values of the convection heat exchange coefficient of the tank wall with liquid (Fig. 3) and gas (Fig. 2) media shows that for gasoline this value is almost 2 orders of magnitude higher than the corresponding value for air.

5. 3. Solving the heat balance equations

As an example, the burning of spilled diesel fuel flowing with a volume velocity of 2 l/s onto a surface with an angle of inclination of 3° is considered. In this case, the direction of inclination coincides with the X axis, and the origin of coordinates is located at the point of liquid leakage (Fig. 4). The RVS-5000 reservoir (height H=12 m, diameter D=23 m) is located at a distance of 6 m from the liquid leakage point in the direction of the Y axis. There is no wind. To determine the shape and size of the spill, the liquid spreading and impregnation model [22] was used with the following parameter values: hydraulic conductivity of wetted soil $1.68\cdot10^{-7}$ m/s; capillarity index (suction head) 0.95 m; soil porosity 0.31; the average depth of irregularities is 1.7 cm.

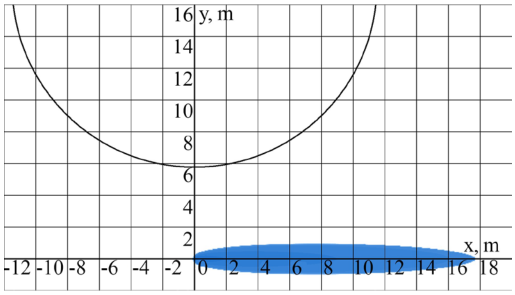

Fig. 4. Mutual location of the diesel fuel filling station and the RVS-5000 tank

Due to change in the size of the spill, the values of the mutual radiation coefficient between the points on the surface of the tank and the flame are not constant and also change over time. Fig. 5 illustrates the distribution of the values of the mutual radiation coefficient along the tank wall at the maximum spill area.

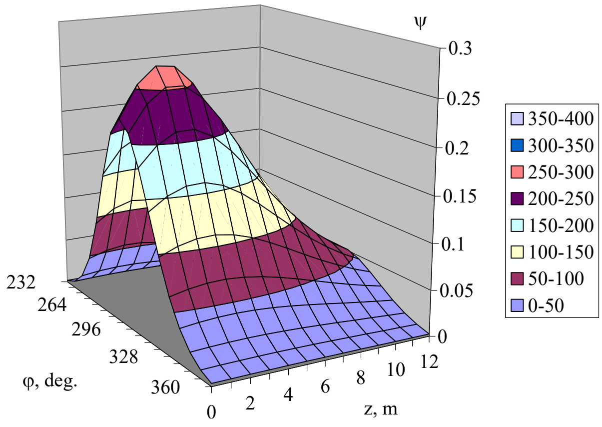

Fig. 5. Mutual location of the diesel fuel filling station and the RVS-5000 tank

Fig. 6 shows the temperature distribution along the wall of a tank filled with gasoline (kinematic viscosity coefficient ν=0.6 mm²/s) up to the level h=4 m, 20 minutes after the start of liquid leakage and fire.

The maximum temperature value of 350 °C is reached on the upper part of the tank, which is not in contact with the liquid, but with the vapor-air mixture in the gas space of the tank (Fig. 6). The part of the wall below the liquid level in the tank reaches a maximum temperature of about 80 °C (Fig. 6), which does not pose a threat of explosion or combustion of gasoline vapors. If the tank is filled with fuel oil (kinematic viscosity coefficient ν=80 mm²/s), then under the same conditions the temperature of the lower part of the wall reaches 230 °C (Fig. 7).

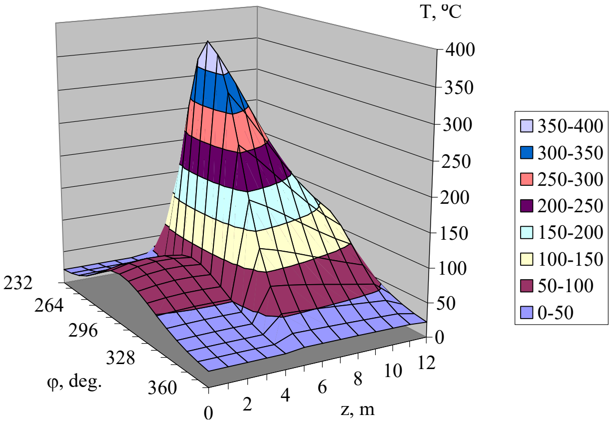

Fig. 6. Temperature distribution along the wall of the RVS-5000 tank filled with gasoline up to a level of 4 m, depending on the vertical coordinate z and the angular coordinate φ 20 minutes after the start of the fire

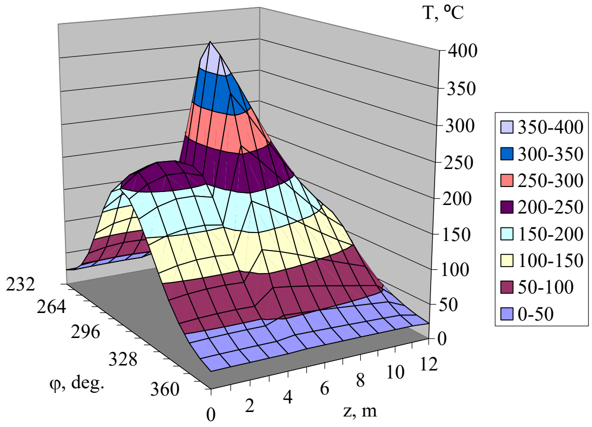

Fig. 7. Temperature distribution along the wall of the RVS-5000 tank filled with fuel oil up to a level of 4 m, depending on the vertical coordinate z and the angular coordinate φ 20 minutes after the start of the fire

The higher temperature value compared to the case of gasoline is due to the fact that the cooling effect of the petroleum product poured into the tank is determined by its viscosity (Fig. 3).

6. Discussion of results related to constructing a model of heating a tank with an oil product under the thermal influence of a fire

The small thickness of the walls and roof of the tank relative to its linear dimensions allows considering the two-dimensional equations of heat conduction on the wall (1) and roof (2) of the tank instead of the three-dimensional equation of the general form. These equations contain a term that takes into account the heat flow from the external environment to a point on the surface of the tank. This heat flow is due to:

– radiant heat exchange with a combustible liquid spill fire (8);

– radiant heat exchange with the environment (9);

– convection heat exchange with air masses (10);

– convection heat exchange of the lower part of the wall (Fig. 1) with the liquid (15);

– convection heat exchange of the upper part of the wall and roof with the steam-air mixture in the gas space of the tank (14);

– radiant heat exchange of the upper part of the wall and roof of the tank with the liquid surface (13), with the inner surface of the wall (11) and roof (12).

The radiation heat flux density is determined by the Stefan-Boltzmann law. At the same time, in (11), (12), in contrast to (8), (9), (13), the temperatures are under the sign of the integral since they take different values at different points of the wall and roof.

The heat flow density due to convection is determined by Newton's law. The value of the coefficient of convection heat exchange, which is included in formulas (10), (14), (15), was found using the criterion equation (16), which holds under the condition of free convection.

Analysis of the graphical dependence shown in Fig. 2 reveals that the value of the coefficient of convection heat exchange of the wall with the surrounding air or steam-air mixture in the gas space of the tank increases with the increase in the temperature of the wall. At the same time, the value of the convection heat exchange coefficient does not exceed 9 $W/(m^2 \cdot K)$.

Liquids with a lower viscosity are characterized by higher values of the convection heat transfer coefficient due to more intensive mixing of the liquid in the walls of the tank. Analysis of graphical dependences in Fig. 3 reveals that the value of the coefficient of convection heat exchange in the case of gasoline in the tank is 5 times higher than the value of the coefficient of convection heat exchange in the case of filling the tank with fuel oil.

The heat balance equations (1), (2) together with boundary conditions (3), (4) and initial conditions (5) describe the temperature distribution along the wall and roof of the tank at an arbitrary moment in time. The finite difference method was used to solve the system of equations, the essence of which is to replace infinitesimally small increments with finite ones and to move from differential equations to finite difference equations.

The comparison shows that a fire spilling a flammable liquid differs from a fire in a vertical steel tank [23, 24] in the following:

– in the presence of a slope of the surface, the spill has the shape not of a circle, but of an oval (Fig. 4), elongated along the direction of the slope;

– the dimensions of the spill change due to leakage, impregnation, and burnout of the liquid;

– the lower part of the tank (Fig. 5), and not the upper part, is exposed to the greatest thermal impact from a spill fire [23].

Analysis of the graphic dependence in Fig. 5 reveals that the highest value of the mutual radiation coefficient is reached at a height of z=2 m. However, due to the fact that the tank was filled with liquid to the level of 4 m, the lower part of the tank was cooled by it. At the same time, the lower part of the wall, which is in contact with the liquid in the tank, is heated to lower temperature values (Fig. 6), despite the higher value of the mutual radiation coefficient. This is due to a larger value of the coefficient of convection heat exchange between the wall and the liquid compared to the value of the similar coefficient for the wall and air. For gasoline, the coefficient of convection heat exchange exceeds the similar value for air by almost 50 times. Taking into account that the self-ignition temperature of gasoline vapors is about 300 °C, it can be concluded that the upper part of the wall, located above the liquid level in the tank, is heated to dangerous temperature values. This means its transformation into an ignition source for gasoline vapors in the tank and the possibility of an explosion of vapors in the gas space of the tank if the concentration of vapors is within the limits between the lower and upper concentration limits of flame propagation. A similar conclusion holds in the case of a fire in one of the tanks in the tank group [23]. Heating the upper part of the tank adjacent to what is burning can turn it into a source of ignition. But in the case of spill burning, the roof of the neighboring tanks is under safer conditions since the combustion center is located much lower compared to the combustion in the tank [25]. The result is also consistent with practical recommendations [24], in which the main attention is paid to cooling the upper part of the tank wall.

A smaller value of the convection heat exchange coefficient in the case of fuel oil leads to the fact that the temperature of the lower part of the wall reaches higher values than for gasoline – 230 °C (Fig. 7). At the same time, the temperature distribution along the upper part of the wall remains the same (Fig. 6, 7).

The advantage of the constructed model of tank heating under the thermal influence of a spill fire is that it allows finding the temperature distribution over the entire surface of a vertical steel tank. This makes it possible to identify areas that heat up to dangerous temperature values and are subject to cooling.

Limitations of the model include the condition of a calm wind, and the condition of a small thickness of the wall and roof of the tank compared to its linear dimensions.

The disadvantage of the constructed model is that the question of the intensity of water supply for cooling the wall and roof of the tank remains unanswered. Thus, the prospects for further research are related to taking into account the cooling effect of water supplied to the walls of the tank with the help of mobile equipment or stationary cooling systems.

The proposed model of the thermal effect of a fire spilling a flammable liquid on a tank with an oil product could be used both in the design of cooling systems [26] and under operational mode. From a practical point of view, the model built is the basis for constructing a decision support system for the head of fire extinguishing. Having an application that implements the specified model, the fire extinguishing manager, after conducting a reconnaissance, can make a decision about the cooling of the tanks and determine the cut-off time for the start of cooling. The results could also be used in the design of fire detectors [27].

7. Conclusions

1. Heat balance equations for the wall and roof of the tank, which are two-dimensional differential equations of the second order of the parabolic type, were constructed. The equations are based on the assumption of a small thickness of the wall and roof of the tank relative to its linear dimensions. The equations take into account radiative and convective heat exchange with the fire, the environment, the liquid, and the vapor-air mixture inside the tank.

2. Convection heat exchange coefficients of the tank wall with gas and liquid media were determined. It is shown that the coefficient of convection heat exchange of the wall with the surrounding air or steam-air mixture in the gas space of the tank increases with the increase in the temperature of the wall. At the same time, the value of the convection heat exchange coefficient does not exceed 9 W/(m^2·K). The coefficient of convection heat exchange of the wall with the liquid in the tank depends significantly on its kinematic viscosity and is about 90 W/(m^2K) for fuel oil at a temperature difference of 100 °C between the wall and the liquid, and for gasoline – about 430 W/(m^2K).

3. It is shown that the greatest amount of heat flow by radiation from the fire falls on the lower part of the tank wall: the value of the mutual radiation coefficient reaches 0.24. But due to the fact that this part of the wall is cooled by contact with the liquid in the tank, the maximum temperature value is reached on the upper part of the wall, which is above the liquid level. In particular, in the case of a tank with gasoline, the temperature of the lower part of the wall does not exceed 80 °C. At the same time, the wall temperature above the liquid level reaches 300 °C.

Conflicts of interest

The authors declare that they have no conflicts of interest in relation to the current study, including financial, personal, authorship, or any other, that could affect the study, as well as the results reported in this paper.

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Data availability

The data will be provided upon reasonable request.

Use of artificial intelligence

The authors confirm that they did not use artificial intelligence technologies when creating the current work.

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