PECULIARITIES OF THE LIQUID CARGO DYNAMICS MODELING FOR THE CASE OF ROAD TANK TRANSIENT MOVEMENT MODES

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Abstract. *The paper deals with computer simulations of liquid cargo sloshing in reservoirs of road tanks. The estimation of turbulent and viscous liquid energy dissipation was done, the dependences of liquid energy dissipation for Newtonian and non-Newtonian models were obtained. Influence of liquid cargo density and viscosity on liquid energy dissipation and hydrodynamic pressures was analyzed. The maximal change in temperature during the first second after the start of road tank braking was determined. The obtained results allowed to make recommendations for computer modeling of a single road tank transient movement.*

1 INTRODUCTION

A great amount of liquids is transported by road tanks. A moving road tanker with liquid cargo is a complex dynamic system and special attention should be paid to the relative displacement of liquid cargo, which can lead to loss of stability and controllability of the car. To provide more safe and ecologically not dangerous transportation process there is a need to improve road tank constructions which can be realized by computer modeling and depends on liquid cargo viscosity. The full account of the phenomena occurring during transient driving modes of the road tank requires to use the model of cargo as a continuous liquid.

Usually, road tanks transport similar liquids during their operation life. Liquid cargo oscillations in road tanks at transient movement modes may cause an increase in liquid temperature due to internal friction forces (energy dissipation of liquid cargo). A substantial liquid temperature change can lead to its viscosity change.

The main purpose of the work is to analyze the value of liquid cargo viscous dissipation and the temperature discontinuity calculation to make recommendations for computer modeling of a single road tank transient movement.

2 DYNAMIC EQUATIONS FOR LIQUID CARGO MOVEMENT

To take into account the phenomena occurring during road tank transient movement modes it is necessary to use the model of transported liquid as a continuous medium. All further results were obtained considering liquid cargo as incompressible liquids due to not significant pressure values.

General relations characterizing the liquid cargo oscillations in the tank reservoir are valid for both Newtonian and non-Newtonian fluids and include such dynamic equations per mass unit, as the equation of mass and energy conservation. They have the following form [1, 2]:

$$\begin{cases} \frac{\partial u}{\partial t}+u\frac{\partial u}{\partial x}+v\frac{\partial u}{\partial y}+w\frac{\partial u}{\partial z}=F_x+\frac{1}{\rho}(-\frac{\partial p}{\partial x}+\frac{\partial \tau_{xx}}{\partial x}+\frac{\partial \tau_{xy}}{\partial y}+\frac{\partial \tau_{xz}}{\partial z}); \\ \frac{\partial v}{\partial t}+u\frac{\partial v}{\partial x}+v\frac{\partial v}{\partial y}+w\frac{\partial v}{\partial z}=F_y+\frac{1}{\rho}(-\frac{\partial p}{\partial y}+\frac{\partial \tau_{yx}}{\partial x}+\frac{\partial \tau_{yy}}{\partial y}+\frac{\partial \tau_{yz}}{\partial z}); \\ \frac{\partial w}{\partial t}+u\frac{\partial w}{\partial x}+v\frac{\partial w}{\partial y}+w\frac{\partial w}{\partial z}=F_z+\frac{1}{\rho}(-\frac{\partial p}{\partial z}+\frac{\partial \tau_{zx}}{\partial x}+\frac{\partial \tau_{zy}}{\partial y}+\frac{\partial \tau_{zz}}{\partial z}); \\ \frac{\partial u}{\partial x}+\frac{\partial v}{\partial y}+\frac{\partial w}{\partial z}=0; \\ \rho C_p\left(\frac{\partial T}{\partial t}+u\frac{\partial T}{\partial x}+v\frac{\partial T}{\partial y}+w\frac{\partial T}{\partial z}\right)=\frac{\partial Q}{\partial t}+k_T\left(\frac{\partial^2 T}{\partial x^2}+\frac{\partial^2 T}{\partial y^2}+\frac{\partial^2 T}{\partial z^2}\right)+\Phi_d, \end{cases} \tag{1}$$

where u, v, w – projections of the fluid particle velocity vector on x, y, z axes, m/s^2;

F_x, F_y, F_z – projections of the external volume (mass) forces, N/m^3;

ρ – liquid density, kg/m^3;

p – isotropic stress (pressure), Pa;

τ_{ij} – components of the stress tensor, Pa.

C_p – specific heat at constant liquid pressure, J/(kg·K);

T – liquid temperature, K;

$\frac{\partial Q}{\partial t}$ – specific rate of external heat sources, J/(kg·s);

k_T – thermal conductivity, W/(m·K);

Φ_d – part of the mechanical energy turning into heat per time unit, J/(kg·s):

$$\Phi_d=\mu\left[2\left(\frac{\partial u}{\partial x}\right)^2+2\left(\frac{\partial v}{\partial y}\right)^2+2\left(\frac{\partial w}{\partial z}\right)^2+\left(\frac{\partial v}{\partial z}+\frac{\partial w}{\partial y}\right)^2+\left(\frac{\partial w}{\partial x}+\frac{\partial u}{\partial z}\right)^2+\left(\frac{\partial u}{\partial y}+\frac{\partial v}{\partial x}\right)^2\right], \tag{2}$$

μ – dynamic viscosity coefficient, Pa·s.

To determine the motion parameters for a liquid with constant viscosity it is enough to solve the system of the first four equations of the system (1). Wherein the temperature field can be calculated from the fifth equation of the system using on the obtained velocity field.

Shear adhesion forces between particles of real liquids at their movement are characterized by liquid cargo internal friction or viscosity. Earlier investigations on the tanks dynamics were performed only for the Newtonian liquids [3, 4]. Shear stresses between the particles of the Newtonian liquid are directly proportional to the relative velocity of its layers movement and depend on the liquid type.

For the non-Newtonian liquids this dependence is not directly proportional. Accordingly, the liquid viscosity coefficient is not constant and can depend on the temperature, pressure and shear rate, its duration and other factors. In the books [5, 6, 7] there are presented classifications of various liquid media and it is mentioned that existing semi-empirical and empirical rheological models can be divided into two wide-spread basic types: pseudoplastic ("purely viscous") and visco-plastic environments.

The system of equations (1) requires to specify boundary conditions corresponding to the analyzed object. In the considered problems the conditions of non- penetration through solid walls $\upsilon_n = 0$ and particles trapping of a viscous fluid $\upsilon_\tau = 0$ [8] were taken into account. The peculiarity of created models is the presence of a liquid-air interface.

At vehicle movement along the road its vertical oscillations always appear and they become a constant source of eddies or artificial turbulizers [9, 10]. In addition, at such transitional movement modes as emergency braking or acceleration, there is a hydroimpact acting the tank wall and top or damping devices. Therefore, all simulations of liquid cargo sloshing at non-stationary movement modes of tanks were carried out taking into account the presence of turbulent stresses, depending on the flow rate oscillations. This led to the appearance of six new turbulent stress unknowns in the first four equations of the system (1). To determine these unknowns it is necessary to find the connection between pulsation and averaged rate in a turbulent flow [1]:

$$
\begin{cases}
\frac{\partial \bar{u}}{\partial t} + \bar{u}\frac{\partial \bar{u}}{\partial x} + \bar{v}\frac{\partial \bar{u}}{\partial y} + \bar{w}\frac{\partial \bar{u}}{\partial z} = F_x + \frac{1}{\rho}\left[-\frac{\partial \bar{p}}{\partial x} + \frac{\partial}{\partial x}\left(\bar{\tau}_{xx} - \rho\overline{u'u'}\right) + \frac{\partial}{\partial y}\left(\bar{\tau}_{xy} - \rho\overline{v'u'}\right) + \frac{\partial}{\partial z}\left(\bar{\tau}_{xz} - \rho\overline{w'u'}\right)\right]; \\
\frac{\partial \bar{v}}{\partial t} + \bar{u}\frac{\partial \bar{v}}{\partial x} + \bar{v}\frac{\partial \bar{v}}{\partial y} + \bar{w}\frac{\partial \bar{v}}{\partial z} = F_y + \frac{1}{\rho}\left[-\frac{\partial \bar{p}}{\partial y} + \frac{\partial}{\partial x}\left(\bar{\tau}_{yx} - \rho\overline{u'v'}\right) + \frac{\partial}{\partial y}\left(\bar{\tau}_{yy} - \rho\overline{v'v'}\right) + \frac{\partial}{\partial z}\left(\bar{\tau}_{yz} - \rho\overline{w'v'}\right)\right]; \\
\frac{\partial \bar{w}}{\partial t} + \bar{u}\frac{\partial \bar{w}}{\partial x} + \bar{v}\frac{\partial \bar{w}}{\partial y} + \bar{w}\frac{\partial \bar{w}}{\partial z} = F_z + \frac{1}{\rho}\left[-\frac{\partial \bar{p}}{\partial z} + \frac{\partial}{\partial x}\left(\bar{\tau}_{zx} - \rho\overline{u'w'}\right) + \frac{\partial}{\partial y}\left(\bar{\tau}_{zy} - \rho\overline{v'w'}\right) + \frac{\partial}{\partial z}\left(\bar{\tau}_{zz} - \rho\overline{w'w'}\right)\right], \\
\frac{\partial \bar{u}}{\partial x} + \frac{\partial \bar{v}}{\partial y} + \frac{\partial \bar{w}}{\partial z} = 0; \\
\frac{\partial u'}{\partial x} + \frac{\partial v'}{\partial y} + \frac{\partial w'}{\partial z} = 0,
\end{cases}
\tag{3}
$$

where $\bar{u}, \bar{v}, \bar{w}$ – projections of the liquid particles averaged rate on x, y, z axes respectively, m/s^2;

u', v', w' – projections of the liquid particles pulse rate on x, y, z axes respectively, m/s^2;

$\bar{\tau}_{ij}$ – averaged stress tensor components, Pa.

This can be provided by turbulence models. The analysis in [11] shows that for the solution of our problem the k-ε turbulence model [11, 12] is more preferable to use. It involves solving of two additional equations for the turbulent kinetic energy k and turbulence energy dissipation rate ε for each finite element of liquid at each time step [11, 12]:

$$\frac{\partial(\rho k)}{\partial t} + \frac{\partial(\rho \bar{v}_j k)}{\partial x_j} = \frac{\partial}{\partial x_j}\left[(\mu + \frac{\mu_t}{\sigma_k})\frac{\partial k}{\partial x_j}\right] - \rho\overline{v_i' v_j'}\frac{\partial \bar{v}_i}{\partial x_j} - \rho\varepsilon; \tag{4}$$

$$\frac{\partial(\rho \varepsilon)}{\partial t} + \frac{\partial(\rho \bar{v}_j \varepsilon)}{\partial x_j} = \frac{\partial}{\partial x_j}\left[(\mu + \frac{\mu_t}{\sigma_\varepsilon})\frac{\partial \varepsilon}{\partial x_j}\right] + \frac{\varepsilon}{k}\left[C_{\varepsilon 1}\left(-\rho\overline{v_i' v_j'}\frac{\partial \bar{v}_i}{\partial x_j}\right) - C_{\varepsilon 2}\rho\varepsilon\right], \tag{5}$$

where k – specific (per unit mass) liquid turbulent kinetic energy, m^2/s^2;

$\bar{v}$ – averaged velocity, m/s;

σ_ε, σ_k, C_μ, $C_{\varepsilon 1}$, $C_{\varepsilon 2}$ – some dimensionless empirical constants;

ε – specific (per unit mass) dissipation rate of liquid turbulent kinetic energy, m^2/s^3;

μ_t – turbulence eddy viscosity, Pa·s:

$$\mu_t = C_\mu \rho \frac{k^2}{\varepsilon}. \tag{6}$$

Analytical solution of equations (3) has not been received yet for the problems of the liquid flows for random initial and boundary conditions, so their analysis is usually done using numerical methods. We've used ANSYS CFX software as a tool for solving the above-mentioned equations.

In ANSYS it is possible to determine only the specific (per unit mass) values of liquid energy dissipation rate. In general case it is necessary to take into consideration the unevenness of finite element mesh, as well as the presence of liquid or air. Therefore, we suggested the method of total turbulent energy dissipation calculation for liquid sloshing in reservoir which includes following steps:

1) finite element modeling of the transported liquid cargo oscillations in the tank for a certain time period;

2) finding the turbulent energy dissipation rate, corresponding to a particular timestep, as the total mass of each finite element and its energy dissipation rate multiplication;

3) determining a cumulative total of liquid energy dissipation per cycle of oscillations as the sum of the energy dissipation rate and the timestep multiplication.

So, all of the presented computational results for liquid cargo energy dissipation were obtained using the mentioned technique.

3 ANALYSIS OF THE ADEQUACY OF COMPUTER SIMULATION RESULTS TO THE EXPERIMENTAL DATA

To analyze the adequacy of the real processes to the modeling techniques used in the ANSYS Workbench environment it was created a finite element model of the water-filled tank moving with acceleration in two axes (lateral and transverse) or which was decelerating at turn. The geometrical dimensions of the tank are shown in Figure 1.

Inside the prototype road tank there were installed three transverse partitions of spherical shape with a hole in the middle [13, 14]. In experimental studies, described in detail in [14], measurements and calculations were carried out for different filling levels of the tank. Also, calculations were performed for the reservoir without partitions and partitions with flat and convex shape, some of which had holes.

The created computer model including more than 110 thousand of finite elements was used for the comparison with experimental data described in [13]. In the calculations, the linear acceleration along the longitudinal axis of the tank is equal to 0.3g, along the transverse axis – 0.25g, the filling level of the tank was equal to 40 %.

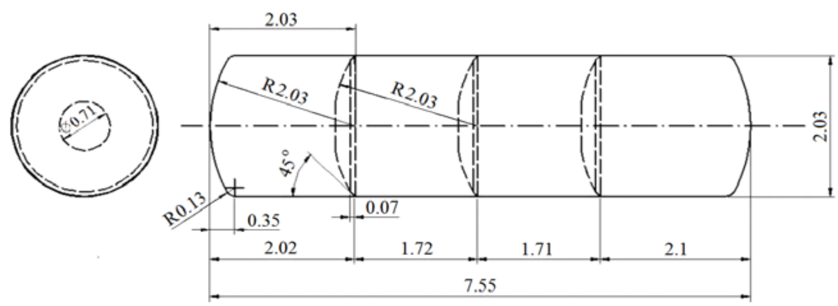

Figure 1: Geometric parameters of the tank with three partitions [13]

The comparison of obtained values of liquid pressure forces corresponding to the longitudinal and lateral forces acting the tank, with the experimental and theoretical results presented in [13], has shown that they do not differ by more than 13.2 %. This confirms the validity of the methodology used in the performed simulations.

4 ANALYSIS OF ENERGY EQUATION NUMERICAL SOLUTION FOR LIQUID CARGO OSCILLATIONS IN ROAD TANK RESERVOIRS

Liquid cargo oscillations in the road tank at its transient movement modes may be accompanied by a liquid temperature increase caused by the internal friction forces (liquid cargo energy dissipation). A substantial change in the temperature of the transported cargo can lead to its viscosity change. The energy dissipation of liquid is divided in two parts: viscous and turbulent. Turbulent energy dissipation of liquid cargo depends on the degree of flow vorticity and it is a part of the vortices mechanical energy, converted into heat. To estimate the effects of liquid turbulent and viscous dissipation at road tank braking there were carried out the simulations of liquid cargo oscillations in the 4 m length reservoir of the rectangular cross-section with dimensions 1.5×1 m^2 for liquids of 0.001–300 Pa·s dynamic viscosity. The reservoir was partially-filled, its filling level was 50 %.

The obtained computational results showed that the value of liquid turbulent energy dissipation component increases for the 80–120 Pa·s liquid dynamic viscosity and then decreases (Figure 2, a). The values of viscous energy dissipation at liquid low viscosities (up to 1 Pa·s) increase in direct proportion to the value of viscosity, but then it depends nonlinearly. When the dynamic viscosity increases in the range of 1–100 Pa·s the viscous energy dissipation increases considerably, then it remains almost constant (in the range of 1200–1400 J), and then decreases gradually, due to a very small relative velocity of the liquid particles (Figure 2, b). Generally, the liquid viscous energy dissipation is 0.0003–2.5 % of the total liquid viscous energy dissipation for different values of the dynamic viscosity. Therefore, it can be neglected in comparison with the turbulent component at computer modeling for cases of road tank braking.

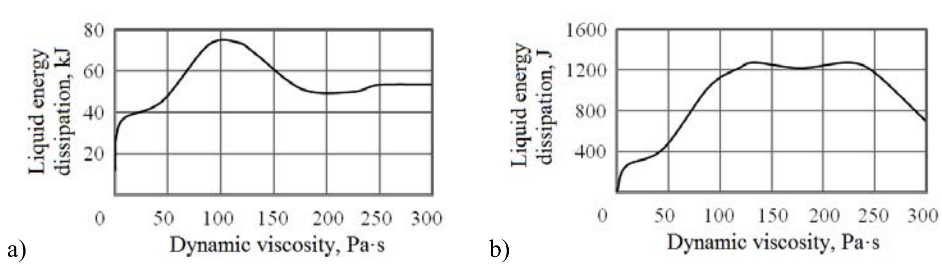

Figure 2: Turbulent (a) and viscous (b) liquid (with water density) energy dissipation at braking of tank with

rectangular cross-section

To analyze the influence of liquid cargo rheological properties on its sloshing in the road tank with baffles and on the values of hydrodynamic pressures and liquid energy dissipation there were performed the simulations for braking (initial velocity – 15 m/s, deceleration – 0.6 g) of road tank (4 m length, 2 m diameter) with perforated baffle (seven holes of 20 cm diameter) partially filled with liquid cargo of 967 kg / m^3 density and μ_0 = 0.1 Pa initial viscosity.

The computations were done for three rheological models of liquid cargo behavior: Newtonian, de Waele and Bingham-Shvedov. Results showed that the total liquid energy dissipation is maximal for Bingham model and it is twice more than the values obtained for other models. At the same time, the results for Newtonian and de Waele models differ less than 10% per one cycle of liquid cargo oscillations [15].

At the same time, the hydrodynamic pressure in the road tank reservoir is largely dependent on the liquid density and this dependence is almost directly proportional. The coefficients of pressure-density proportionality for different liquids in comparison with water are given in the table 1.

Type of liquid cargo	Liquid cargo density		Maximal values of hydrodynamic pressures	
	kg/m^3	relative to water density	kPa	relative to water pressure
Nitric acid	1512.6	1.51	8.29	1.46
Gasoline	734.5	0.74	4.20	0.74
Water	998.2	1.00	5.69	1.00
Kerosene	815.0	0.82	4.63	0.81
Xylol	980.0	0.98	5.41	0.95
Treacle	1440.0	1.40	8.02	1.41
Sunflower oil	920.0	0.92	5.25	0.92

Table 1 : Comparison of liquids density and pressure values with the corresponding values for water

In order to analyze the effect of the internal friction on the liquid temperature change, there were performed the computations of liquid cargo sloshing in road tank for liquids of 0.001–300 Pa·s dynamic viscosity. Based on our previous calculations [16], the vehicle of the considered configuration stops in 4.2 seconds at emergency braking. Therefore, simulations were performed specifically for this period of time.

Dependence of the liquid average temperature maximum increase on its viscosity is shown in Figure 3. As can be seen from the figure, when the cargo viscosity increases to 1 Pa·s it can be observed the maximal average temperature increase, and then, with a viscosity decrease the temperature changes decrease either due to lower relative velocities of liquid cargo particles.

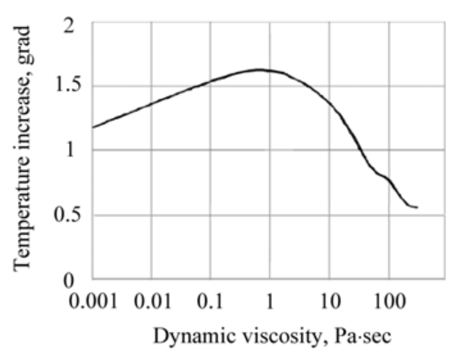

Figure 3: The dependence of average temperature increase on the liquid cargo viscosity

In general, the maximum temperature deviation from the initial value was 1,62 °C. So, if there is a need of computational analysis for road tanks with liquid cargo for one cycle of tank transient movement the temperature changes can be neglected and the process can be considered to be isothermal.

The obtained results have shown that the maximal change in temperature takes place during the first second after the start of braking. Computational results confirmed the obvious fact that the movement of liquid in the tank becomes more gradually when liquid viscosity increases (Figure 4). It can be explained by the fact that the relative velocities of liquid particles decrease at liquid dynamic viscosity increase.

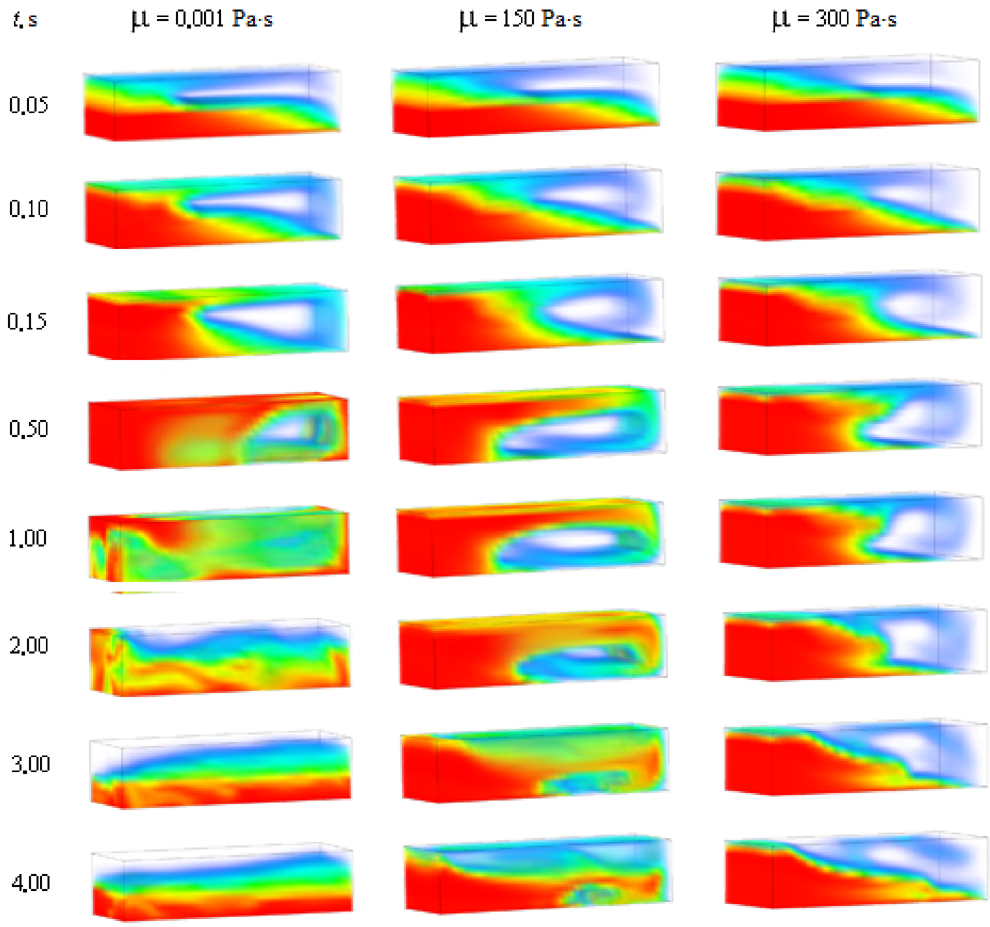

Figure 4: Free surface position for liquid with different dynamic viscosity at road tank braking with 0.6*g* acceleration

5 CONCLUSIONS

1 The comparison of our values of liquid pressure forces corresponding to the longitudinal and lateral forces acting on the tank, with the experimental and theoretical results presented in [13], has shown that they do not differ by more than 13.2 % . So the methodology used in the performed simulations is valid.

2 Computational results demonstrated that liquid viscous energy dissipation is 0.0003–2.5 % of the total liquid viscous energy dissipation for different values of the dynamic viscosity. Therefore, it can be neglected in comparison with the turbulent component at computer modeling for cases of road tank braking.

3 The results showed that the hydrodynamic pressure in the road tank reservoir is largely dependent on the liquid density and this dependence is almost directly proportional. There were calculated the coefficients of pressure-density proportionality for different liquids in comparison with water.

4 The obtained results have shown that the maximal change in temperature takes place during the first second after the start of braking. In general, the maximum temperature deviation from the initial value was 1.62 °C. So, for computational analysis of road tanks with liquid cargo for one cycle of tank transient movement the process can be considered to be isothermal

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