Mathematical model of loading a semi-trailer bearing system with an autonomous diesel power plant on turn

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Abstract. Road trains consisting of a tow truck and a semi-trailer are widely used nowadays in Russia for cargo transportation, including in different territories. However, in mountainous operating conditions compared to plain ones, the reliability of the load-bearing system of semitrailers of road trains is significantly reduced, as a result of which there is a mismatch between the flow of semitrailer failures and scheduled repairs, which significantly increases the costs of their operation. Different approaches can be used to reduce these costs. Many domestic and foreign scientists have dealt with the issues of reliability of auto transport systems. Their works contain a lot of valuable information of applied significance, but the reliability of the load-bearing system of semitrailers with an autonomous diesel power plant of road trains and its improvement have not received sufficient attention.

Key words: mathematical model, road train, loading, power plant.

1 Introduction

When organising transport, various ways of placing the load in the body are used. [1-3] To facilitate loading and unloading operations, the load is often placed mainly in the rear of the semi-trailer with the required load distribution on the axles of the combination (Fig. 1a) or with maximum displacement to the rear (Fig. 1b) [4].

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Fig. 1. Schemes of load placement in a semi-trailer to facilitate loading and unloading operations: a - with observance of the required load distribution on the axles of the road train; b - without observance of the required load distribution on the axles of the road train.

Fig. 2. Structural diagram of the road train with indication of the main parameters.

When the load is evenly distributed in the body (Fig. 2), the total centre of mass of the road train (excluding the mass of axles and wheels) is at the point C.M.\(_{rt}\). [5]

2 Methods

We assume that the part of the load above the saddle and the related part of the van with the centre of mass at the point C.M.\(_{0}\) is rigidly connected to the tow truck, the centre of mass of which is located at the point C.M.\(_{t}\). Thus, the tow truck with the saddle and the part of the cargo and van located above the saddle are represented as one rigid body (the front part of the road train) [6] with the centre of mass at the point C.M.\(_{1}\). A part of the load and the van with the frame above the semi-trailer suspension is also represented as one rigid body (the rear part of the combination) with centre of mass at the point C.M.\(_{2}\). Distances \(h_1\) and \(h_2\) from the load centres to the roll centres, respectively, of the front and rear parts of the road train are the arms of the moments of centrifugal forces that arise at corners and cause angular deformations of the front and rear parts of the road train [7]. If these angular deformations differ from each other, then there is an angular deformation of the frame, causing increased stresses in its elements. To simplify the model, at the first stage of modelling, the moments from the weight component of the roll component of the road train parts are not taken into account due to their smallness.

Fig. 3. C.M.\(_{t}\) - centre of mass of the truck; C.M.\(_{1}\) and C.M.\(_{2}\) - centres of mass of the front and rear parts of the road train

Based on the assumptions made, a calculation scheme was developed to determine the torsion angle of the road train frame at a turn, shown in Fig. 3.
In this calculation scheme, the rod $C_r$ (torsion) simulates the frame of the semi-trailer when it twists due to the moments of centrifugal forces of inertia. $F_1$ and $F_2$ have different magnitudes [8-9]. The axis of the rod is the roll axis of the road train. It connects the roll centres of the tow truck and semi-trailer. The roll centre is the instantaneous centre of displacement, which remains at rest with transverse body rolls or with different in sign but equal in magnitude wheel displacements. Methods for determining the roll centre position and angular stiffness for dependent and independent suspensions are discussed in the literature. Many tow trucks and semi-trailers have a dependent leaf spring suspension on semi-elliptic leaf springs. The roll centre of such suspension lies in the plane under the spring lugs, and the angular stiffnesses of the front and rear suspensions are determined, respectively, according to the formulas:

\[
C_{fr,ss} = 0.5c_{fr,ss}b_{fr,ss}^2
\]

\[
C_{r,ss} = 0.5c_{r,ss}b_{r,ss}^2
\]

here $b_{fr,ss}$ - distance between front suspension springs, $b_{r,ss}$ - distance between rear suspension springs, $c_{fr,ss}$ and $c_{r,ss}$ are stiffnesses of the elastic elements of the front and rear suspensions, respectively [10].

The angular stiffness of the tyres can be determined the same way:

\[
C_{fr,t} = 0.5c_{fr,t}b_{fr,t}^2
\]

\[
C_{r,t} = 0.5c_{r,t}b_{r,t}^2
\]

here $b_{fr,t}$ - front wheel track, $b_{r,t}$ - rear wheel track, $c_{fr,t}$ - stiffness of front tyre; $c_{r,t}$ - stiffness of rear tyre (in case of twin wheels and several rear axles - total stiffness of all tyres of one side).

Then angular stiffness of front and rear suspension taking into account tyres can be determined by the following formula:

\[
C_{fr} = \frac{C_{fr,ss}+C_{fr,t}}{C_{fr,ss}+C_{fr,t}}
\]
To increase the angular stiffness of the suspension in order to reduce body roll on a corner, a transverse stability stabiliser is installed. It is made, as a rule, in the form of a torsion arm with bent ends installed in rubber bushings and at vertical vibrations of the body simply rotates in the bushings, not creating forces. Therefore, it does not increase the vertical stiffness of the suspension, but allows to increase its angular stiffness \[ C_t = \frac{C_{rs} + C_{rt}}{C_{rs} + C_{rt}} \] (6) 

Thus, the obtained formulas allow us to determine the angular stiffness of tow trucks with different co-wheel formulas [14]. Angular stiffness of truck suspension \( C_1 \) is equal to the sum of the angular stiffnesses of the front and rear suspensions: 

\[ C_1 = C_{fr} + C_r \] (8)

Having determined the angular stiffness of the suspension, the roll angle of the sprung mass can be calculated by the formula:

\[ \varphi_1 = \frac{M}{C_1} \] (9)

The maximum value of the roll angle is regulated by normative documents: its value must not exceed 6-7° at a centrifugal force equal to 40 % of the vehicle weight.

Torsion angle of the frame is:

\[ \Delta \varphi = \frac{l_1h_1 + \frac{C_{s}h_2}{C_s^2} + \frac{C_{s}h_2}{C_s^2} + \frac{l_2h_2}{C_p^2}}{C_1 + \left(\frac{C_1}{C_2} + 1\right)C_p} \] (10)

3 Results

The results are summarised in Fig. 5.
4 Discussion

Fig. 4 shows that qualitatively the calculated graph (points are connected by straight lines) and the experimental graph (points are connected by curved lines) fully correspond to each other and have a good quantitative convergence. The analysis of the obtained dependence shows that when increasing the transverse stiffness of the rear suspension $C_2$ (in our case of the semi-trailer suspension) and reducing the angular stiffness of the front suspension $C_1$ (in our case of the tow truck suspension), the frame torsion angle decreases and tends to zero. The quantitative discrepancy between the results of calculation and experiment is due to the inaccuracy of the mathematical model, due to the failure to take into account some influencing factors [15-36]. The calculation scheme and the mathematical model were further refined [37-40].

5 Conclusions

Rational aggregation of tow truck and semi-trailer according to suspension parameters consists in their selection, at which the deformation of the semi-trailer frame on a turn was equal to zero or had a minimum value.

To find the ratio of angular stiffnesses of the tow truck and semi-trailer, at which the specific angle of twisting of the frame when overcoming a turn will be equal to zero, we can use the dependence, which includes the parameters of cargo placement [41-43].
Fig. 6. Dependence of the specific torsion angle of the frame on the relative transverse stiffness of the road train suspension (a) at $h_1 = 1.5 \text{ m}$, $h_2 = 1.8 \text{ m}$ and the relation $l_2/l_1$ equal to: 0.4 - curve 1; 0.5 - curve 2; 0.6 - curve 3; 0.7 - curve 4.

The vertical dashed lines in Fig. 6 show the values of the relative transverse stiffness of the road train suspension at which the specific torsional angle of the frame on bends is zero, i.e., the frame is not loaded with torque. Analysis of the graphs in Fig. 5 shows that if it is not possible to select a tow truck and semitrailer with an optimum ratio of angular stiffness, the specific torsion angle of the frame on corners increases almost proportionally to the deviation from the optimum ratio, i.e., the frame is loaded.

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