Investigation of dynamic loading of the traveling system of track-type tractor

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Abstract. The article presents a technique for studying dynamic loading of the tracked vehicle undercarriage system. It includes developing a dynamic model of the undercarriage system for loads simulation in typical loading modes and finite element models of the undercarriage component parts to determine the stress-strain state and the safety factor of the parts for each computational case. The undercarriage system is described as a set of solid bodies connected by elastic-damping force elements among themselves, and with a ground contact area for a numerical solution of equations of motion. The time domain loads on the undercarriage system components obtained according to such a model in typical loading modes are used to identify the most dangerous, from the point of view of strength, computational cases for each element. Loads for each computational case are transferred to finite element models of the undercarriage parts, which are used to evaluate the strength. The technique is displayed in typical modes “tilted down on the idler when lifting the blade” and “single rail crossing”. The problems of load transfer from a dynamic model to component finite element models are considered. The described technique improves the reliability of undercarriage strength estimate in typical operation modes and allows you to reasonably choose configuration parameters by considering a large number of complex spatial loading modes, taking into account the kinematics and elastic characteristics of the undercarriage system.

1. Introduction
One of the most important tasks at the crawler tractor development stage is to ensure the required strength parameters of the undercarriage system structural elements. This includes determination of the maximum loads acting upon components and units in typical loading conditions. Currently, to solve this problem, the finite element method is widely used [1, 2]. In a first approximation, loads can be determined according to simplified models by solving a static problem. To register load dynamic components, a spatial dynamic model is used, taking into account the kinematics and elastic characteristics of the undercarriage system elements. For commercial crawler tractors, the number of structural elements can reach several dozens, which significantly complicates the analytical derivation of the equations of motion. Modern tools to solve these tasks are software packages for automated analysis of the multibody dynamics [3, 4, 5, 6, 7]. Equations of motion are formed in such complexes according to the description of the system in the form of a set of solid or deformable bodies, hinges and force interactions from the library of typical elements, and there are built-in tools for their numerical solution. On the basis of such complexes, specialized applications have been developed, they significantly simplify tracked vehicle models development, taking into account the dynamics and...
kinematics of the undercarriage system elements [8, 9, 10, 11]. An additional advantage of dynamic models application for load assessment is the ability to consider a large number of computational cases, it allows, firstly, for assessing the strength with high reliability over the entire range of operating loads, and, secondly, for solving optimization and weight reduction problems effectively [12,13]. This paper discusses the technique of strength analysis of the tracked vehicle undercarriage system as an example of a commercial crawler tractor, drawbar category 75, with load calculation using a dynamic model created in the UM Tracked Vehicle dynamics analysis module of the “Universal Mechanism” Russian program package [8].

2. Main part
The study object is a commercial crawler tractor of drawbar category 75 with an independent bogie suspension of the dual-roller micro-sprung suspension.

The dynamic model of a suspension-mounted system consists of a mass-inertial model of the body frame, as well as a mass-inertial model of the attached equipment that is hinged to the body frame and restricted with geometric constraints. The track roller frame, bogie suspension elements and tensioner are modeled as a set of solids connected by linear elastic damping load-carrying elements. To obtain the reaction distribution in the bearings of the bogie and balancer axles, the bearing units are modeled by linear elastic-damping force constraints in the directions of load bearing area. Track chains are modeled taking into account track spatial dynamics as a set of rigid bodies connected by hinges. The interaction of tracks with the drive wheel, road wheels and track support rollers, as well as with a hard ground contact area is carried out using linear contact interaction models built into the UM Tracked Vehicle module.

Load analysis of the undercarriage system involves selection of loading modes, which are the most dangerous in terms of strength. These modes are determined by standard operating conditions of commercial crawler tractors. The loading modes of the undercarriage system of the commercial tractor under consideration are shown in Table 1. In each mode, the coefficient of traction between the tracks and the road is assumed to be 1. The “equilibrium” mode is introduced as the basis to estimate the level of loads in other modes.

Table 1. Loading Modes

| №  | Name and diagram | Description |
|----|------------------|-------------|
| 0. | Equilibrium      | The blade and ripper are in a transport position. Driving speed is 0 km/h. |
| 1. | Tilted down on the idler when lifting the blade | The blade is in a working position; the ripper is in a transport position. The moment on the driving wheels corresponds to the clutch limit. The resistance to the blade movement corresponds to the ultimate force of track adhesion to the ground. The force in the blade hydraulic lift cylinders corresponds to the maximum pressure. |
| 2. | Tilted down on the rear road wheels when lifting the ripper | The blade is in a transport position; the ripper is in a working position. The moment on the driving wheels corresponds to the clutch limit. The resistance to movement of the ripper corresponds to the ultimate adhesive force of the tracks to the ground. The force in the ripper hydraulic lift cylinders corresponds to the maximum load on the rear bogie. |
| 3. | Single rail crossing | The blade and ripper are in a transport position. Movement speed is minimum steady. |
| 4. | Angular restraint | The blade and ripper are in a transport position. The moments on the driving wheels of the different sides correspond to the clutch limit and are directed in the opposite directions. The right idler and rear road roller abut against the stoppers at the height of their centers. |
For each of such modes according to the dynamic model, time domain loads on the undercarriage system elements are obtained. In each case for each element, the moments of maximum loading are distinguished, and they are taken as calculated cases in the static strength analysis. Below, there is an example of the analysis of the loads on the bogies for the “tilted down on the idlers when lifting the blade” mode. The most loaded unit is the front bogie of the undercarriage system.

From the diagram analysis of the vertical reactions in Figure 1, it can be concluded that in this mode, the highest vertical loads on the front bogie occur at time $I$. As the tractor continues to be tilted down, the loads are redistributed from the bogie to the idler. In this case, it is obvious that the loads on the bogie at the time $I$ determine the computational case for the bogie strength analysis.

The number of computational cases may exceed the number of computational modes. Thus, in the “single rail crossing” mode from the loading cases analysis, it is impossible to unambiguously single out a single computational case for strength analysis, even if we consider only the loads on the bogie from the road wheel axles. The graphs in Figure 2 shows that first (point I) a large negative longitudinal force appears at the front roller, and then as the bogie drives onto the rail (point II), on the contrary, it is positive, and, further on, it is a negative longitudinal force at the rear bogie road roller (point III). The vertical forces directed to these rollers also behave differently: first, a large vertical force acts on the first roller, then almost an entire vertical reaction moves to the rear roller. Thus, to analyze the bogie strength in this mode, it is necessary to consider at least three computational cases corresponding to different combinations of magnitudes and directions of loads on the bogie. If to take into account the remaining loads (forces and moments) acting on the bogie, then the number of computational cases may be even greater.

**Figure 1.** Vertical reactions to the front bogie in the “tilted down on the idlers when lifting the blade” mode: 1, 2 - from the front and rear sections of the shock absorber; 3, 4 - from the bearings of the front roller axle; 5, 6 - from the bearings of the rear roller axis; 7, 8 - from the upper stoppers

**Figure 2.** Longitudinal ($F_x$) and vertical ($F_z$) reactions of the axles of the middle bogie road wheels in the “single rail crossing” mode: 1 - front roller axles; 2 - rear roller axle
The reactions obtained by the dynamic model for each computational case are then transferred to finite element models of the analyzed components. Most software packages of finite element analysis allow for importing loads through a special format text file. This file contains coordinates of the reaction points and coordinates of the reactions vectors (forces and moments) at each point. To carry out strength analysis, it is possible to consider the details of the tractor undercarriage in the equilibrium position at each moment of time according to the D'Alembert's principle. To do this, the transmitted reactions in the hinges of at read shoe are complemented by inertial forces distributed over the component volume by the inertia relief method:

\[ (F^a) + \int_V (a) \rho dV = (0) \]  
\[ (M^e) + \int_V (r) \times [(\varepsilon) \times (r)] \rho dV = (0) \]  

where:

- \((F^a), (M^e)\) are the system of external, with respect to a component, forces and moments acting in the nodes of the finite element model that coincide with the centers of the hinges;
- \((a)\) is linear accelerations of a component;
- \((\varepsilon)\) is rotational accelerations of a component;
- \((r)\) is the radius vector of component body points;
- \(\rho, V\) are the component density and volume.

In this case, when calculating the stress-strain state by the finite element method, there is no need to set boundary conditions for each computational case, which allows for automating the process of loads transfer into the finite element model and for considering a very large number of computational cases without a significant increase in the workload [14].

One of the problems in transferring loads into a finite element model by the inertia relief method is the requirement that the mass inertia characteristics of a component should exactly match each other both in the dynamic model and in the finite element model, especially with regard to mass moments of inertia of a component. It is also required to ensure coincidence between the mass centers of the dynamic and component finite element models. Therefore, no significant simplifications of models deflected from the original geometry of calculated components are allowed when developing finite element models. Also, if possible, it is necessary to use computational cases where inertial loads are absent or small if compared with external force factors.

Another problem is the fact that, in the dynamic model, loads are lumped, and, in the finite element, they are distributed. The transition to distributed loads in the software packages of finite element analysis is carried out by introducing specific constraints between the degrees of freedom of the load application point and the nodes of the component part model. For example, for this purpose the NASTRAN software package uses the elements of type RBE2 or RBE3.

The example of loading the tractor undercarriage bogie shown in Figure 3 requires application of a nonlinear finite element iterative equation solver to correctly determine the contact forces of interaction of the axle and bearing rings. The model also takes into account the pretensioned bogie axles; the models of welds and other contact elements in the interacting bogie components are entered. The calculation was carried out in a nonlinear geometric formulation, taking into account the elastoplastic properties of materials for steel components according to the bilinear diagram.

The stress-strain state computation results for the computational case "tilted down on the idler when lifting the blade" are shown in Figure 4 and 5.
When analyzing the stress-strain state of the body considering a large number of computational cases, it is convenient to use the stress distribution diagram for all computational cases. If there is a classification of calculated cases, for example, they can be divided into regular operational, extreme and emergency, then it makes sense to calculate the stress distribution diagram for each group of loads separately, since for them, you can assign different requirements for safety factors or permissible acting stresses.

3. Conclusions
The technique of strength analysis of the crawler tractor undercarriage system, based on sequential application of a dynamic model and finite element models with possible automated data transfer, is demonstrated. The spatial nonlinear dynamic model of the undercarriage system allows for identifying the most dangerous, from the point of view of the load, strength and of their combinations acting on its components, taking into account the elastic properties of load-bearing elements and kinematics of guide elements. Spatial finite element models provide for the stress-strain state of the undercarriage system components. Application of the inertia relief method for transferring loads from a dynamic to a finite element model automates the process of exchanging data between the models and allows for increasing the number of calculated cases under consideration significantly without a significant increase in workload. Such an approach increases reliability of undercarriage strength evaluation in typical operating conditions and allows for choosing reasonable configuration parameters of its elements.
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