Dynamic simulation of force loading of drives of mobile power facilities with variable external resistance

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Abstract. The article describes dynamical simulation of the transfer of power load parameters from the engine to the final mechanism on a mobile power source under external variable loads. Using the dynamic model developed by the authors the kinematic and power characteristics of each intermediate mechanism can be found.

1 Introduction

Recently, in mobile power vehicles (tractors and cars) and technological machines, combined high-load drives are increasingly used, the main purpose of which is to increase the productivity and efficiency of machine operation by harmonizing the components of the transformable power, which is achieved by changing the speed of the machine's executive body in the stationary mode of operation of the drive engine.

The main parameters characterizing the variables of external load disturbances are: - coefficient of adhesion of the wheel of the mobile power vehicle with the supporting surface; wheel rolling resistance coefficient; the force of external resistance from the working units; longitudinal and transverse slope of the supporting surface, height (depth) of road irregularities. To determine them as informational variables, it is advisable to use the wheel speed:

- variable speed of movement of mobile power; changes in the value of the torque supplied to the front wheels and the traction force developed by them; uneven weight on the front wheels; engine torque jump.

Besides, variable external resistance occurs when there is a kinematic mismatch, it worsens the traction performance of the tractor, and if the front and rear wheels operate with different slippage, then the grip of the lagging wheels is used to a lesser extent than the grip of the running wheels. The greater the kinematic discrepancy, the more unevenly the grip qualities of the wheels of both axles are used, in this case only the running wheels remain the

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driving wheels, because lagging wheels roll with a skid, therefore, they become driven. In works [1,2], the author considered the rectilinear motion of a 4K4 tractor, in which the rear wheels run, and the front wheels roll with a skid (Figure 1). In this case, the front wheels are affected by a negative tangential traction force $P_{K1}$ created by the reactions of the supporting surface and directed against the movement and is a brake, forming a torque $M_{T1}$, which is transmitted through the front wheel drive to the transfer case.

![Fig. 1. Rice 1-Scheme of distribution of load torques in the transmission of the 4K4 tractor.](image)

The methods of kinematic analysis and force calculation have been used to study the cyclic process of dynamic loading of the working equipment of the pumping complex. The methodology of system analysis was used, as well as the method of redistribution of the moments of inertia forces and the power balance of the power drive. The mathematical apparatus is effectively used to establish the dependences of the change in power characteristics on the performance indicators and dynamic loads of the pumping complex. The method of critical analysis established the priorities for the development of geometric parameters, and the use of the laws of mechanics made it possible to substantiate the structural elements of kinematics that require improvement. A new scientific idea has been developed and substantiated for implementing the principles of energy recovery and optimal redistribution of power when performing the most energy-intensive work processes of a ground-based production pump complex, which has enriched the scientific concept of increasing the energy efficiency of production complexes and expanded the boundaries of applicability of the results obtained in hybrid drives [3].

Loading cycle (high-cycle fatigue) and high-frequency cyclic loading with torsional-flexural vibrations of the blades (high-cycle fatigue) of a compressor disk of an aircraft gas turbine engine, which is important for applications, is considered. An estimate of the durability of the considered element of aircraft structures is given. An example of calculating fatigue failure under high-frequency torsional vibrations of an experimental sample of a certain shape is also considered, in which it was possible to reproduce the observed effect of a sharp change in the direction of growth and type of quasi-crack during cyclic loading [4].

It is worth noting that recently, abroad, ground-based mobile power facilities have been created in accordance with new requirements. But still they do not meet the requirements of promising machines. Therefore, there is a need to develop their designs, for which it is
necessary to create appropriate theories. This work is devoted to the development of dynamic modeling of force loading of drives with variable external resistance, with new scientific approaches

2 Research methods

We use the method of analytical mechanics to write equations for the transfer of force loading of the drive motor and driven mechanisms in generalized coordinates with indefinite factors [5]:

- for drive motor

\[
\frac{d}{dt} \left( \frac{\partial \tau}{\partial \phi_1} \right) - \frac{\partial \tau}{\partial \phi_1} = Q_1 + \lambda, \tag{1}
\]

- for driven mechanisms

\[
\frac{d}{dt} \left( \frac{\partial \tau}{\partial \phi_2} \right) - \frac{\partial \tau}{\partial \phi_2} = Q_2 + \lambda i. \tag{2}
\]

where \(Q_1, Q_2\) are the generalized forces of the drive motor and driven mechanisms; \(T\) is the kinetic energy of the rotating masses; \(\lambda\)- indefinite multiplier;

The kinetic energy equation of the system:

\[
T = \frac{1}{2} (J_1 \dot{\phi}_1^2 + J_2 \dot{\phi}_2^2)
\]

where \(J_1\) is the moment of inertia of the elements kinematically connected with the rotor of the drive motor and the masses brought to it (the rotor itself, intermediate transmission elements, the leading part of the continuously variable transmission); \(J_2\) is the moment of inertia of the elements kinematically connected with the driven part of the continuously variable controlled transmission, reduced to the latter (its driven part, intermediate elements and the executive body of the machine).

3 Research results

The desired generalized forces \(Q_1\) and \(Q_2\) can be found as coefficients for the corresponding variations of the generalized coordinates. If in the left parts of equations (1) and (2) we perform the necessary operations of differentiation and substitute the values of the generalized forces, then these equations will take the following form:

\[
\begin{align*}
J_1 \ddot{\phi}_1 &= (A_1 - B_1 \ddot{\phi}_1) + \lambda, \\
J_2 \ddot{\phi}_2 &= M_i - \lambda i, i = a + bm
\end{align*}
\]

where \(A\) and \(B\) are the parameters of the straight line, with the help of which the linear part of the mechanical characteristic of the engine is approximated, and \(A \equiv 0.03d, M_{in}^{max}\) and \(B\) is the tangent of the slope of the linear part of the characteristic, \(I\) is the gear ratio, and \(a\) and \(b\) are some constants determined by the design data of the controlled transmission, \(b = \frac{1}{l_{gea}}\); the control action of the built-in control circuit on the stepless transmission is characterized by the value of the controlled coordinate \(m\), which has a functional dependence \(m = f_0(t)\). In fact, this function denotes the regularity of the connection of the driven links [6], \(m = \ddot{f}_0(t) = st\).

An indefinite factor can be found by the expressions [7,8]

\[
\begin{align*}
\lambda &= J_1 (b \dot{m} \ddot{\phi}_2 + (a + bm) \ddot{\phi}_2) - (A - B (a + bm) \ddot{\phi}_2) \\
J_2 \ddot{\phi}_2 &= M_{c3} + ((J_1 (b \dot{m} \ddot{\phi}_2 + (a + bm) \ddot{\phi}_2) - (A_2 - B_2 (a + bm) \ddot{\phi}_2)) i, \\
J_2 \ddot{\phi}_2 i_2^{-1} - J_1 (a + bm) \ddot{\phi}_2 - J_1 (a + bm) \ddot{\phi}_2 - A_2 + B_2 (a + bm) \ddot{\phi}_2 &= M_{c3} i_2^{-1} \\
[2 \ddot{i}_2^{-1} - J_1 (a + bm) \ddot{\phi}_2 - J_1 (b \dot{m} - B_2 (a + bm)) \ddot{\phi}_2 - A_2 = M_{c3} i_2^{-1} \\
[2 \ddot{i}_2^{-1} - J_1 i_2^{-1}] \ddot{\phi}_2 - [J_1 b_1 s_2 \ddot{\phi}_2 - B_2 i_2^{-1}] \ddot{\phi}_2 - A_2 = M_{c3} i_2^{-1}\end{align*}
\]

3
\[ (J_2 - J_1) i_2^{-1} \dot{\phi}_2 - [J_1 b_1 s_2 t_2 - B_2 i_2^{-1}] \dot{\phi}_2 - A_2 = M_{c3} i_2^{-1} \]
\[ M_{c3} = M_{c2} + k_3 t_2 \]

We divide this equation by \( i_2^{-1} \) and get

\[ (J_2 - J_1) \dot{\phi}_2 - [J_1 b_1 s_2 t_2 - B_2] \dot{\phi}_2 - A_2 i_2 = M_{c2} + k_3 t_2 \]

\( k \) - the value of the overshoot resistive load.

For the entire \( n \times q \) mechanical system of the machine, we write

\[ \begin{align*}
(J_{n-1} - J_n) \dot{\phi}_{n-1} - [J_{n-2} b_{n-2} s_{n-1} t_{n-1} i_{n-1} - B_{n-1}] \dot{\phi}_{n-1} - A_{n-1} i_{n-1} \\
= M_{c_{n-1}} + k_{n-1} t_{n-1} \\
\end{align*} \]

\[ \begin{align*}
(J_n - J_{n-1}) \dot{\phi}_{n-1} - [J_{n-1} b_{n-1} s_n t_n i_n - B_n] \dot{\phi}_n - A_n i_n = M_{c_n} + k_n t_n \quad (3) \\
\end{align*} \]

With the solution of the above systems of equations, we can find the kinematic parameters of intermediate drive mechanisms with variable external loads in the form of a moment of resistance.

4 Conclusion

Thus, the force loading of drives under external variable loads is dynamically modeled. This dynamic model is used to calculate the kinematic and power characteristics, taking into account the dynamic loading of each mechanism, mass, dimensions, power resistance drop.

References

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