Journal of Physics: Conference Series

PAPER • OPEN ACCESS

Dynamic method of controlling the traction force driving wheels of vehicle

To cite this article: A V Egorov *et al* 2019 *J. Phys.: Conf. Ser.* **1399** 055053

Dynamic method of controlling the traction force driving wheels of vehicle

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Abstract. The contemporary stage of development of methods and means of control traction force developed on the driving wheels of vehicles is used as one of the diagnostic parameters of the angular acceleration of the drive wheels and the angular acceleration of transmission units, however, scientific and technical studies of the methods take into account the law of generation of mechanical power during one revolution shaft engine vehicle. Also developed methods that are used to control the torque developed by the drive wheels of the vehicle, the acceleration time to a certain speed. At the contemporary stage of development of methods and means, there is an incomplete justification for the use of angular accelerations of transmission units, driving wheels for controlling traction force developed on the driving wheels of vehicles, and the non-use of linear accelerations of vehicles for controlling developed traction force. No disadvantages from the above is a dynamic method of controlling traction force the drive wheels of a vehicle, the scientific and technical justification of which is given in this article.

1. Introduction

At the present level of development of methods and means of controlling traction force developed on the driving wheels of vehicles, use as one of the diagnostic parameters the angular accelerations of the driving wheels and the angular accelerations of the transmission units [1-7], however, the scientific and technical substantiation of methods take into account the law of mechanical generation power for one revolution of the engine shaft of the vehicle.

Also developed methods that are used to control the torque developed by the drive wheels of the vehicle, the acceleration time to a certain speed [8].

Thus, at the present stage of development there is an incomplete justification for the use of angular accelerations of transmission units, driving wheels for controlling traction force developed on driving wheels of vehicles, as well as non-use of linear accelerations of vehicles for controlling developed traction force.

Journal of Physics: Conference Series **1399** (2019) 055053 doi:10.1088/1742-6596/1399/5/055053

2. Development of a dynamic method for controlling traction force the drive wheels of a wheeled vehicle

Consider the kinematic scheme of the rear-wheel drive 4 - wheel vehicle (figure 1).

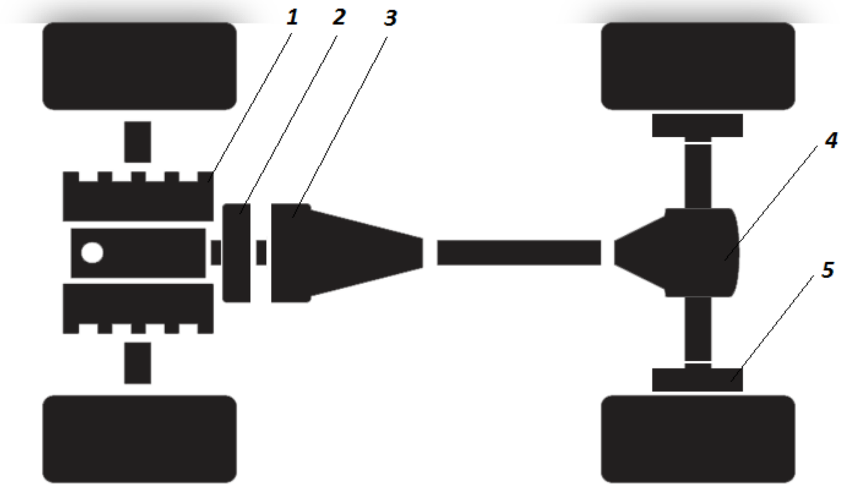

Figure 1. Kinematic scheme of the rear-wheel drive 4 - wheel vehicle: 1 – internal combustion engine; 2 – clutch; 3 – change speed gearbox; 4 – final drive; 5 – final reduction gear.

The determination of the moment of inertia to the axis of rotation of the crankshaft is made under the assumption that the internal combustion engine through the kinematic transmission must report the mechanical energy necessary to drive the four wheels. Each of the wheels has a weight m_{wheels} and radius r_{wheels}, taking into account the deformation of the tire . On each wheel in the first approximation, accounts for the $1/4$ weight of the vehicle with the driver $m_{vehicle}$ and weight of cargo m_{cargo}. In the case of a particular vehicle and its cargo, the ratio may be different.

According to [11], the moment of inertia of the system of rotating loads is the moment of inertia of the system consisting only of elements rotating at an angular velocity of the internal combustion engine shaft ω_{engine}, but having a reserve of kinetic energy equal to the reserve of kinetic energy of the real system.

From the condition of invariance of the kinetic energy, it follows that for a system consisting of four wheels connected through the gears of the internal combustion engine and rotating at an angular speed ω_{wheels}, having, taking into account the rolling friction on the surface and sliding friction in the supports, the total moment of inertia $J_{\Sigma_{wheels}}$, the speed of the vehicle $V_{\mathrm{vehicle}} = \omega_{wheels} r_{wheels}$, in disregard of the action of air resistance, we obtain

$$J\ (\omega)\frac{\omega_{\text{engine}}^2}{2}=J_{\text{engine}}(\omega)\frac{\omega_{\text{engine}}^2}{2}+J_{transmis.}(\omega)\frac{\omega_{\text{engine}}^2}{2}+J_{\Sigma_{wheels}}(\omega)\frac{\omega_{wheels}^2}{2}+ \\ +(\text{m}_{\text{vehicle}}+m_{cargo})\frac{V_{vehicle}^2}{2}, \tag{1}$$

where $J\ (\omega)$ - the moment of inertia transmission units, brought to the axis of rotation of the crankshaft, taking into account mechanical losses; $J_{\text{engine}}(\omega)$ - the moment of inertia of the moving reciprocating and rotating parts of the engine, brought to the axis of rotation of the crankshaft, taking into account mechanical losses; $J_{\text{transmis.}}(\omega)$ - the moment of inertia of the transmission units, brought to the axis of rotation of the crankshaft, taking into account mechanical losses; $J_{\Sigma_{wheels}}(\omega)$ - the moment of inertia of the wheels of the vehicle, brought to the axis of rotation of the crankshaft of the engine, taking into account mechanical losses.

From here the required reduced moment of inertia of the system is

$$J(\omega)=J_{\text{engine}}(\omega)+J_{transmis.}(\omega)+J_{\Sigma_{wheels}}(\omega)\left(\frac{\omega_{wheels}}{\omega_{\text{engine}}}\right)^2+\frac{(m_{\text{vehicle}}+m_{cargo})V_{\text{vehicle}}^2}{\omega_{\text{engine}}^2}. \tag{2}$$

The gear ratio between the internal combustion engine and the drive wheel is equal to the product ratio change speed gearbox k_{CSG} ratio final drive k_{FD} and, in the case of an final reduction gear, ratio final reduction gear k_{FRG}.

Then (2) can be represented as

$$J\ (\omega)=J_{engine}(\omega)+J_{transmis.}(\omega)+J_{\Sigma_{wheels}}(\omega)\left(\frac{1}{k_{CSG}k_{FD}k_{FRG}}\right)^2+\frac{(\text{m}_{\text{vehicle}}+m_{cargo})V_{\text{vehicle}}^2}{\omega_{\text{engine}}^2}= \\ =J_{engine}(\omega)+J_{transmis.}(\omega)+\left(J_{\Sigma_{wheels}}(\omega)+(m_{\text{vehicle}}+m_{cargo})r_{wheels}^2\right)\cdot\left(\frac{1}{k_{CSG}k_{FD}k_{FRG}}\right)^2. \tag{3}$$

The dependence of the internal combustion engine torque on the rotational speed of the engine crankshaft, expressed through the angular acceleration of the crankshaft and the moment of inertia of the loads rotating and translational motion, is given

$$M(\omega)=J\ (\omega)\varepsilon(\omega)=\left[J_{engine}(\omega)+J_{transmis.}(\omega)+\left(\frac{J_{\Sigma_{wheels}}+(\text{m}_{\text{vehicle}}+m_{cargo})r_{wheels}^2}{k_{CSG}^2k_{FD}^2k_{FRG}^2}\right)\right]\varepsilon(\omega). \tag{4}$$

Calculated torque equation during acceleration of a vehicle with a driver when the internal combustion engine is operating according to the external characteristic (when the accelerator is pressed all the way)

$$M(\omega)=\left[J_{engine}(\omega)+J_{transmis.}(\omega)+\left(\frac{J_{\Sigma_{wheels}}+m_{\text{vehicle}}r^2_{wheels}}{k^2_{CSG}k^2_{FD}k^2_{FRG}}\right)\right]\varepsilon_1(\omega), \tag{5}$$

where $\varepsilon_1(\omega)$ - angular acceleration of the crankshaft of the internal combustion engine during acceleration of the vehicle only by the driver.

Calculated torque equation during acceleration of a vehicle with a driver and a cargo when the internal combustion engine is operating according to the external characteristic (when the accelerator is pressed all the way)

$$M(\omega)=\left[J_{engine}(\omega)+J_{transmis.}(\omega)+\left(\frac{J_{\Sigma_{wheels}}+(m_{\text{vehicle}}+m_{\text{cargo}})r^2_{wheels}}{k^2_{CSG}k^2_{FD}k^2_{FRG}}\right)\right]\varepsilon_2(\omega), \tag{6}$$

where $\varepsilon_2(\omega)$ - angular acceleration of the crankshaft of the engine during acceleration of a vehicle with a driver and weight m_{cargo} .

Equating (5) and (6), we determine the amount

$$J_{engine}(\omega)+J_{transmis.}(\omega)=\frac{m_{\text{cargo}}r^2_{wheels}}{k^2_{CSG}k^2_{FG}k^2_{FRG}}\cdot\frac{\varepsilon_2(\omega)}{\varepsilon_1(\omega)-\varepsilon_2(\omega)}-\frac{J_{\Sigma_\kappa}+m_{vechicle}r^2_{wheels}}{k^2_{CSG}k^2_{FG}k^2_{FRG}}. \tag{7}$$

Substituting (7) into (5) or (6)

$$\mathrm{M}(\omega)=\frac{m_{\text{cargo}}r^2_{wheels}}{k^2_{CSG}k^2_{FG}k^2_{FRG}}\cdot\frac{\varepsilon_1(\omega)\varepsilon_2(\omega)}{\varepsilon_1(\omega)-\varepsilon_2(\omega)}. \tag{8}$$

From (8) determine the traction force developed by the driving wheels:

$$\mathrm{F}(\omega)=\frac{M(\omega)}{r_{vechicle}}=\frac{M_{\text{cargo}}r_{vechicle}}{k^2_{CSG}k^2_{FG}k^2_{FRG}}\cdot\frac{\varepsilon_1(\omega)\varepsilon_2(\omega)}{\varepsilon_1(\omega)-\varepsilon_2(\omega)}. \tag{9}$$

Similarly, the dynamic method of controlling the traction force drive wheels, based on the equality of the traction forces on the drive wheels during acceleration without weight and with weight, can also be solved by analyzing the dynamics of the linear motion of the wheeled vehicle. The scheme of the method implementation is shown in figure 2.

At the initial stage, the vehicle 1, the change in the position of the engine controls of which is carried out according to a certain law, accelerates in a specific gear in a horizontal section without slipping, while the total weight of the vehicle is $m_{vehicle}$, and the acceleration of the vehicle is *a*.

Then on the vehicle 1 is placed a cargo of unknown weight 2 and a vehicle 1, the change in the position of the engine controls of the engine which is carried out according to the same specific law, accelerates at a particular gear in a loaded state in a horizontal section without slipping when the weight of the cargo m_{cargo}, and the acceleration of the car with the cargo is a_1.

The traction force developed by the driving wheels of the vehicle will be denoted by F.

Assuming in the first approximation that the frictional force of the wheels on the roadway during the first and second acceleration of the vehicle remains unchanged, we write down the projections on the Ox axis acting on the vehicle during the first and second acceleration:

$$F = m_{vehicle} a, \tag{10}$$

$$F = \left(m_{vehicle} + m_{cargo}\right) a_1. \tag{11}$$

Since the first and second acceleration of the vehicle was carried out according to the same definite law of changing the position of the engine controls on one particular gear in a horizontal stretch without slipping, then, respectively, traction force drive wheels developed every time the same.

Figure 2. Scheme for the implementation of the dynamic method for controlling of weight of cargo transported by vehicle: 1 – vehicle; 2 – cargo.

Equating (10) to (11) we determine the unknown mass $m_{vechicle}$:

$$m_{vechicle} = \frac{F}{a}, \tag{12}$$

$$m_{vechicle} = \frac{F}{a_1} - m_{cargo}. \tag{13}$$

Equating (12) to (13) we determine the unknown traction force developed by the driving wheels of the wheeled vehicle F:

$$F = m_{cargo} \frac{a \cdot a_1}{a - a_1}. \tag{14}$$

An application for the invention of the Russian Federation [9] has been filed for the considered dynamic method for controlling the traction force of the drive wheels of a motor vehicle.

3. Conclusion

Control traction force of the drive wheels of the vehicle is possible when using the drive motor as the diagnostic indicator of the angular acceleration of the shaft when the vehicle is moving with or without a cargo.

Journal of Physics: Conference Series **1399** (2019) 055053 doi:10.1088/1742-6596/1399/5/055053

Control of the drive wheels of the vehicle is possible when using as a diagnostic indicator of linear acceleration of the vehicle when driving with a cargo of known mass and without it.

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