Dynamic method of control of weight of cargo transported by vehicle

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Abstract. The existing methods of control of the total weight of vehicle are based mainly on strain measurement methods and are implemented in the form of mobile or stationary weighing platforms. The main drawback of such systems is the need to enter the vehicle on the weighing platform. In the context of a developed road network, this approach does not allow to track the weight of all vehicles moving on roads, and thus creates conditions for incomplete identification of vehicle, the total weight of which exceeds the permitted for a particular road.

Existing strain gauge weighing systems, placed directly on the vehicle, tend to deteriorate the accuracy of weighing for a certain time of operation. The several works performed in the 90s of the twentieth century describes the method of dynamic control of the weight of vehicle and train. However, the method does not contain a clear theoretical scientific justification and is based on experimental data, so this method has not been further developed. The article presents a theoretical justification of the dynamic method of controlling the weight of vehicle, implemented on the basis of the analysis of the dynamics of angular accelerations of the motor of a vehicle during acceleration on identical surfaces in the empty and loaded state.

1. Introduction

The existing methods of control of the total weight of vehicle are based mainly on strain measurement methods and are implemented in the form of mobile or stationary weighing platforms, [1-6]. The main drawback of such systems is the need to enter the vehicle on the weighing platform. In the context of a developed road network, this approach does not allow to track the weight of all vehicles moving on roads, and thus creates conditions for incomplete identification of vehicle, the total weight of which exceeds the permitted for a particular road.

Existing strain gauge weighing systems, placed directly on the vehicle, tend to worsen the accuracy of weighing for a certain time of operation [7].

In scientific works [8-10] the method of dynamic control of the weight of vehicle and train is described. However, the method does not contain a clear theoretical scientific justification and is based on experimental data.
Thus, to date, to maintain the road network in the normative state, there is no possibility to control the weight of all vehicles moving on the roads and the actual task is to develop methods for controlling the weight of vehicle directly during their movement on the roads.

2. Development of a dynamic method of control of weight of cargo transported by vehicle

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

![Kinematic scheme of the rear-wheel drive 4-wheel vehicle](image)

**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 \(m_{\text{wheels}}\) and radius \(r_{\text{wheels}}\), 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 \(m_{\text{vehicle}}\) and weight of cargo \(m_{\text{cargo}}\). 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 \(\omega_{\text{engine}}\), 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 \(\omega_{\text{wheels}}\), having, taking into account the rolling friction on the surface and sliding friction in the supports, the total moment of inertia \(J_{\Sigma_{\text{transm.}}}\), the speed of the vehicle \(V_{\text{vehicle}} = \omega_{\text{wheels}}r_{\text{wheels}}\), in disregard of the action of air resistance, we obtain

\[
J(\omega)\frac{\omega^2_{\text{engine}}}{2} = J_{\text{engine}}(\omega)\frac{\omega^2_{\text{engine}}}{2} + J_{\text{transm.}}(\omega)\frac{\omega^2_{\text{engine}}}{2} + J_{\Sigma_{\text{transm.}}}(\omega)\frac{\omega^2_{\text{wheels}}}{2} +
\]

\[+(m_{\text{vehicle}}+m_{\text{cargo}})\frac{V^2_{\text{vehicle}}}{2},\]

(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; \(J_{\text{transm.}}(\omega)\) - the moment of inertia of the transmission units, brought
to the axis of rotation of the crankshaft, taking into account mechanical losses; \( J_{\text{wheels}}(\omega) \) - 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

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

(2)

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

Then (2) can be represented as

\[
J(\omega) = \left( J_{\text{engine}}(\omega) + J_{\text{transmis.}}(\omega) + J_{\Sigma_{\text{wheels}}}(\omega) \right) \left( \frac{1}{k_{\text{CSG}} k_{\text{FD}} k_{\text{FRG}}} \right)^2 + \frac{(m_{\text{vehicle}} + m_{\text{cargo}}) V_{\text{vehicle}}^2}{\omega_{\text{engine}}^2} = 
\]

\[
= \left( J_{\text{engine}}(\omega) + J_{\text{transmis.}}(\omega) + J_{\Sigma_{\text{wheels}}}(\omega) \right) \left( \frac{1}{k_{\text{CSG}} k_{\text{FD}} k_{\text{FRG}}} \right)^2 \left( m_{\text{vehicle}} + m_{\text{cargo}} \right) V_{\text{vehicle}}^2.
\]

(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

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

(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)

\[
M(\omega) = \left[ J_{\text{engine}}(\omega) + J_{\text{transmis.}}(\omega) + \left( J_{\Sigma_{\text{wheels}}} \right) \left( \frac{r_{\text{wheels}}^2}{k_{\text{CSG}} k_{\text{FD}} k_{\text{FRG}}} \right) \right] \varepsilon_1(\omega),
\]

(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)

\[
M(\omega) = \left[ J_{\text{engine}}(\omega) + J_{\text{transmis.}}(\omega) + \left( J_{\Sigma_{\text{wheels}}} \right) \left( \frac{r_{\text{wheels}}^2}{k_{\text{CSG}} k_{\text{FD}} k_{\text{FRG}}} \right) \right] \varepsilon_2(\omega),
\]

(6)
where $\varepsilon_2(\omega)$ - angular acceleration of the crankshaft of the engine during acceleration of a vehicle with a driver and weight $m_{argo}$.

Equating (5) and (6), we define $m_{argo}$:

$$m_{argo} = \frac{k_{CSG}^2 k_{FD} k_{FRG}^2}{r_{vehicle}^2} \left( J_{engine}(\omega) + J_{transmis}(\omega) + \left( \frac{J_{vehicle} + m_{vehicle} k_{vehicle}^2}{k_{CSG}^2 k_{FD}^2 k_{FRG}^2} \right) \right) \frac{\varepsilon_1(\omega) - \varepsilon_2(\omega)}{\varepsilon_1(\omega)}.$$  \hspace{1cm} (7)

The presented dynamic method for controlling the weight of cargo transported by vehicle is protected by a patent for the invention of the Russian Federation [12].

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

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

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 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:

$$F = m_{vehicle}a,$$  \hspace{1cm} (8)

$$F = (m_{vehicle} + m_{cargo})a_1.$$  \hspace{1cm} (9)
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 drive wheels developed every time the same.

Equating (8) to (9) we determine the unknown mass \( m_{\text{cargo}} \):

\[
m_{\text{cargo}} = M \left( \frac{a - a_1}{a_1} \right).
\]  

(10)

An application for a patent has been filed for a of a dynamic method of control of weight of cargo transported by vehicle [13].

3. Conclusion

Thus, knowing the weight of an empty vehicle and measuring its linear acceleration values during acceleration using the same law of power supply on identical sections of the track in an empty and laden condition, you can determine the weight of the cargo transported by this vehicle.

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