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METHOD OF SIMULATION OF INTERNAL TECHNICAL CAPABILITIES OF VEHICLES

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Abstract: *The operational characteristics of a car on a highway depend on its traction and dynamic capabilities (TDC) and design and technical parameters, in other words, on their implementation. The tortuodynamic capabilities of a car depend primarily on the external or model (frequent) characteristics of its engine. The article presents the method of simulation of internal technical capabilities of vehicles. On the basis of experimental tests and analysis, an external or model characteristic of the engine is created. Based on this, the interpolation coefficient of equations is determined, corresponding to the number of revolutions of the crankshaft. In this case, the number of revolutions is determined. For the type of vehicle moving along the route, the correction factor, the control factor is the number of gears for transmission, the number of gears of the main transmitter, the transmission efficiency, the dynamic coefficient and the vehicle wheel rolling coefficient values are determined.*

Key words: *engine characteristic, internal technical capabilities, operational characteristics, simulation method, transmission efficiency.*

1 Introduction

Growing concerns about environmental issues such as global warming and pollutant emissions, as well as projected depletion of oil reserves, have made energy efficiency and the reduction of pollutant emissions a major selling point for cars. As a result, the design of a vehicle's powertrain becomes more complex, as it must achieve additional goals without sacrificing other characteristics such as power, torque, or driving pleasure. In this context, due to the exponential growth of computing power, the ground transportation industry has recognized that fast, efficient and cost-effective development of engines and vehicles requires the use of digital simulation at every stage of the design process [1].

The operational characteristics of a car on a highway depend on its traction and dynamic capabilities (TDC) and design and technical parameters, in other words, on their implementation.

The tortuodynamic capabilities of a car depend primarily on the external or model (frequent) characteristics of its engine. This characteristic is represented by the torque (Me) of the engine crankshaft. The torque of the engine crankshaft (Me) is formed depending on the degree of opening of the throttle valve of the carburetor at different values of the number of revolutions n of the shaft, i.e. $M\text{e}(\text{п}) = f(\Delta Ty)$, where ΔTy – is a percentage of the degree of throttle opening in y - gear.

For example, the crankshaft of the engine 3 with different n_1, n_2 and n_3 and the number of revolutions $(n_1 < n_2 < n_3)$ of its torque $M\text{e}(\text{п}_1)$, $M\text{e}(\text{п}_2)$ and $M\text{e}(n_3)$ and the change in values depending on the throttle opening degree ΔTy is shown in Fig. 1.

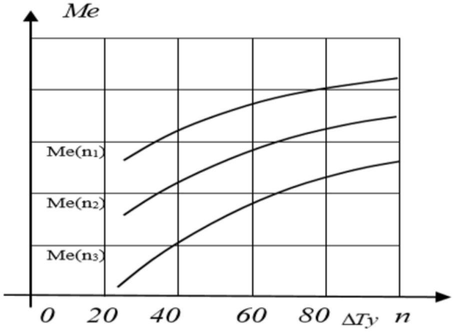

Fig. 1. $\text{Me}(\text{п}) = \text{f}(\Delta\text{Ty})$ relationship graphs $(\text{n}_1 < \text{n}_2 < \text{n}_3)$

Thus, the magnitude of the engine torque *Me* changes depending on the number of shaft revolutions *n*, but this change occurs depending on the degree of opening of the throttle valve ΔTy (Figure 2) [2].

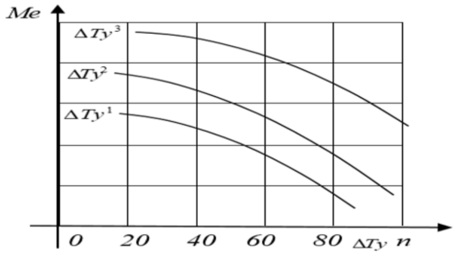

Fig. 2. Me $= \mathrm{f(n)}$ relationship graphs ($\Delta Ty^1 < \Delta Ty^2 < \Delta Ty^3$)

Shadrin and Ivanov [3] consider the main trends in the development of modern automotive industry. The possibilities of using automotive networks in relation to the exchange of data about vehicles in solving research problems, conducting road tests, developing a control system for autonomous vehicles, as well as in ITS applications (intelligent transport systems) are considered.

2 Literature review

The traffic safety rules introduced by the Civil Code are constantly introducing more and more stringent requirements for vehicles. These requirements relate to various aspects of the technical condition of vehicles, both determining traffic safety and affecting the impact of the vehicle on the environment. The law requires regular diagnostics of the technical condition of vehicles in operation. Diagnostic tests, whether in the form of on-road or factory tests, can identify signs of malfunction in the test vehicle without always clearly identifying the cause and location of the damage. The aim of Tucki et al. [4] - development of a simulation station for dynamic testing of vehicles weighing up to 3.5 tons using the simulation programs OpenModelica and SciLab. Achieved imitation of the stand to test the dynamics of vehicles in the form of a dyno.

In the works of Dobrzyński et al. [5] an analysis of the results of the implementation of driving parameters recorded in real operation conditions, mapped at a dynamometer station, is presented. For the analysis, a position was used that allows determining the properties of the vehicle, as well as modeling the user's behavior in the process of overcoming a given speed profile. The principle of their creation required the preparation of equipment that allows recording physical parameters when a car moves along public roads. Obtaining the value became possible after creating driving conditions using a car equipped for this purpose with a compression ignition engine and a maximum permissible mass of not more than 3.5 tons. At the first stage of the work, the boundary conditions necessary for the simulation were determined. The accepted values made it possible to simulate the design parameters of the vehicle, the drive system and the course of the profile of the passed measuring section. At the next stage, we began to simulate the operation of the engine at a dynamometric station that performs the specified work in transient conditions. The analyzes carried out made it possible to realistically assess the possibility of using the process to create progressive development work.

Millo et al. [1] present two examples of numerical simulation applied to the transmission design; while the former focused on evaluating vehicle performance and engine behavior under transient conditions, the latter instead focused on evaluating the fuel saving potential of various hybrid electric vehicle architectures.

The initial data, which is the dependence of the torque (Me) on the number of revolutions of the crankshaft n, that is, the external or general model characteristic of the engine, is determined empirically - by testing on a special stand. Based on the magnitude of the engine torque, the traction force applied to the wheel of the car is $P_к$.

And this force is formed in accordance with the number of gears (y), the number of revolutions of the crankshaft n and the throttle opening degree ΔTy, i.e.

$$P_к = f(y, n, \Delta Ty) \tag{1}$$

The speed of the car V_a is formed depending on the number of revolutions of the crankshaft in the known y - gear n.

The main reason for modeling the speed of the vehicle depending on the number of revolutions and gears of the engine shaft is that the driver uses the fully open throttle position while driving. From statistical observations in the form of well-known experience - tests, it is known that the degree of opening of the throttle valve with a different number of gears (y) is as follows (Figure 3) [6]:

when y = V gear ΔTy =(50÷75) % ; y = IV gear ΔTy =(75÷85) % ;
y = III gear ΔTy =(85÷100) % ; y = I, II gear ΔTy =(95÷100) %.

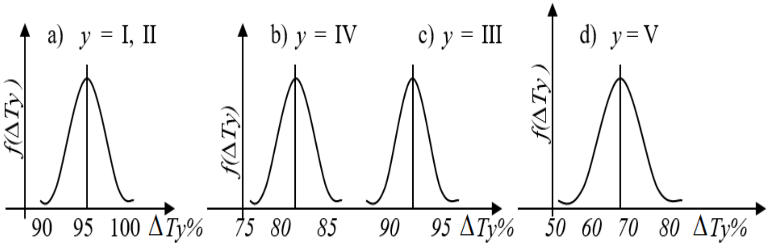

Fig. 3. Distribution of the degree of opening of the throttle valve ΔTy according to the number of different gears (y ={I ÷ V):
a) y ={ I, II }- I and II , b) y = IV, c) y = III and d) y = V by the number of gears.

It is known that the relationship between the real torque (*Me*) of the engine and the number of revolutions *n* can be expressed by the equation of a 3-phase curve of the 2nd order [7] (G.V. Zimelev), i.e.

$$Me = an^2 + bn + c \tag{2}$$

here *a,b,c* – relationship equation coefficients,

n – crankshaft speed.

The study by Ahmed et al. [8] focuses on performance results obtained by comparing two variants of naturally aspirated gasoline engines with a hybrid electric booster gasoline engine using MATLAB Driveline. The comparative study analyzes the output power obtained from the simulation. The new era of hybrid electric technology can deliver comparable performance to naturally aspirated engines is the main focus of research.

3 Methodology

When the car is moving, bumps in the road cause it to vibrate. When this vibration reaches a certain limit, the driver experiences discomfort and fatigue, which affects the reliability and durability of the bearing system. The approach to design and implementation of a virtual vehicle vibration test system is based on virtual reality (VR) and consists of a VR subsystem, a model subsystem, and a virtual instrument subsystem. A system of vibration tests of virtual vehicles based on virtual reality has been created and the results of vibration tests of virtual vehicles have been presented. The data obtained show that the theoretical model of virtual test

(VT) technology refers to the creation of vehicle vibration systems based on VR VT systems. With the system, users can monitor vehicle vibration and its signals in three separate monitoring modes in the time and frequency domains [9].

The position of the connecting line $Me = f(n)$ in the rectangular coordinate system depends on the degree of opening of the throttle valve ΔTy. The position of this line in the rectangular coordinate system is determined by its coefficients a, b and c. Therefore, the values of these coefficients will be different for different values of ΔTy. Thus, equation $Me = f(n)$ is expressed as follows depending on the degree of opening of the throttle valve ΔTy [10, 11]:

$$Me(n) = a(\Delta Ty)n^2 + b(\Delta Ty)n + c(\Delta Ty) \tag{3}$$

The coefficients $a(\Delta Ty)$, $b(\Delta Ty)$ and $c(\Delta Ty)$ of the above equation can be determined based on the Lagrange interpolation formula, i.e.

$$Me(\Delta Ty) = Me_1(\Delta Ty) \cdot \frac{(n-n_2)\cdot(n-n_3)}{(n_1-n_2)\cdot(n_1-n_3)} + Me_2(\Delta Ty) \cdot \frac{(n-n_3)\cdot(n-n_1)}{(n_2-n_3)\cdot(n_1-n_2)} + Me_3(\Delta Ty) \cdot \frac{(n-n_1)\cdot(n-n_2)}{(n_1-n_3)\cdot(n_2-n_3)} \tag{4}$$

Here $Me_1(\Delta Ty), Me_2(\Delta Ty), Me_3(\Delta Ty)$ are the ordinates of 3 points obtained from the external or model characteristics of the engine experimentally or by calculation, depending on $\boldsymbol{\Delta Ty}$, respectively;

The coordinates Me_1, Me_2, Me_3 are calculated with the following n_1, n_2 and n_3 engine crankshaft values:

- $n_1 = n_{max}$ - in the number of revolutions that give maximum torque;
- $n_3 = n_{max}$ - at the speed that gives the engine maximum power;
- $n_2 = \frac{n_1 + n_3}{2} = \frac{n_{\max} Me + n_{\max} Ne}{2}$ - n_2 ва n_3 in the number of revolutions between values n_2 and n_3.

After that, the experimental torque values of the engine Me (ΔTy) are determined in accordance with the corresponding values ΔTy, which determine different degrees of throttle opening, and according to the number of revolutions n_1, n_2, n_3. Based on this, the torques for the number of revolutions n_1, n_2 and n_3 are $Me_1(\Delta Ty), Me_2(\Delta Ty)$ and $Me_3(\Delta Ty)$ coefficients interpolating ΔTy coupling equations, i.e.

a_1, b_1 and c_1 coefficients of equation $Me_1(\Delta Ty) = a_1 + b_1 \Delta Ty + c_1(\Delta Ty)^2$,

a_2, b_2 and c_2 coefficients of equation $Me_2(\Delta Ty) = a_2 + b_2 \Delta Ty + c_2(\Delta Ty)^2$,

a_3, b_3 and c_3 coefficients of equation $Me_3(\Delta Ty) = a_3 + b_3 \Delta Ty + c_3(\Delta Ty)^2$ are determined.

4 Results and discussion

From the expressions above, we can determine the $Me = f(n)$ compound coefficients with a few simple modifications:

$$\left.\begin{aligned}
a(\Delta Ty) &= \frac{Me_1(n_1 - n_3) - Me_2(n_2 - n_3) + Me_3(n_1 - n_2)}{(n_2 - n_3)(n_1 - n_3)(n_1 - n_2)} \\
b(\Delta Ty) &= \frac{n_1^2(Me_1 - Me_3) - n_2^2(Me_1 - Me_3) + n_3^2(Me_1 - Me_2)}{(n_2 - n_3)(n_1 - n_3)(n_1 - n_2)} \\
c(\Delta Ty) &= \frac{Me_1 n_2 n_3(n_2 - n_3) - Me_2 n_1 n_3(n_1 - n_3) - Me_3 n_1 n_2(n_1 - n_2)}{(n_2 - n_3)(n_1 - n_3)(n_1 - n_2)}
\end{aligned}\right\}$$

It is known that $Me(\Delta Ty)$ is a typical engine characteristic obtained at the stand. The characteristic of the engine installed directly in the car is somewhat different from the characteristic obtained during bench tests. Due to this, the torque $M\text{д}(\Delta Ty)$ for the engine running in the car is determined as follows:

$$M\partial(\Delta Ty) = Me(\Delta Ty)K\text{м}$$

where *Км* is the correction factor, *Км=0,88* for two-axle trucks, *Км=0,85* for multi-axle trucks.

$P_\text{к}$ - vehicle traction force:

$$P_к = \frac{Мд(\Delta Ty) \cdot Cy \cdot Cay}{r_g} \cdot \eta = \frac{Me(\Delta Ty) \cdot \text{Км} \cdot \text{Cy} \cdot \text{Cay}}{r_g} \cdot \eta \quad (6)$$

here *Cy* - *y* - number of gears in a gear,

Cay - number of transmissions of the main transmitter,

r_g - dynamic wheel radius,

η - transmission efficiency ($\eta = 0{,}88 \div 0{,}9$) for two-axle vehicles, $\eta = 0{,}84$ for three-axle vehicles.

The dynamic coefficient of a car wheel can be equated to the static wheel coefficient of (r_{CT}) in calculations with moderate accuracy. The value of the statistical coefficient is given in state standards. The dynamic radius of a wheel can also be found from its rolling radius. Wheel radius values are also specified in national standards.

If we substitute the moment $Me(\Delta Ty)$ into formula (3), which converts the traction force of the car into the expression indicated in formula (6), we obtain the following formula:

$$P_к = a(\Delta Ty) \cdot \frac{Км \cdot Cy \cdot Cay}{r_g} n^2 - b(\Delta Ty) \cdot \frac{Км \cdot Cy \cdot Cay}{r_g} n + c(\Delta Ty) \cdot \frac{Км \cdot Cy \cdot Cay}{r_g} \quad (7)$$

If we take into account the presence of the following quantitative relationship between the number of revolutions of the crankshaft of the engine *n* and the speed of the car $\boldsymbol{V}$, i.e. $n = V \cdot Cay \cdot Ca/0{,}377$ then the expression (4.7) after some adjustments takes the following form and changes:

$$P_к = Ay(\Delta Ty)_{г} V^2 + By(\Delta Ty)_{г} V + Cy(\Delta Ty) \qquad (8)$$

where *Ay, By, Cy* are the coefficients representing the equation of the traction force on the driving wheel of the vehicle, which are determined by the following formulas:

$$Ay(\Delta Ty) = a(\Delta Ty) \cdot \text{Км} \cdot \frac{(\text{Ca} \cdot \text{Cay})^3}{0{,}377^2 \cdot r_g \cdot r_F^2}$$

$$By(\Delta Ty) = b(\Delta Ty) \cdot \text{Км} \cdot \frac{(\text{Ca} \cdot \text{Cay})^2}{0{,}377 \quad \cdot r_g \cdot r_F} \tag{9}$$

$$\text{C}y(\Delta Ty) = \text{c}(\Delta Ty) \cdot \text{Км} \cdot \frac{\text{Ca} \cdot \text{Cay}}{r_g}$$

5 Conclusions

Thus, the traction force modeling algorithm $P_к$, which characterizes its technical capabilities in terms of movement, includes the following blocks:

1. Determining the number of gears y used in the current t moment of movement.

- If $y \in \{1{,}2\}$, then $\Delta Ty = \Delta Ty^{max}$ can be accepted.
- If $y>2$, then the exact y values of the number of transmissions are determined ΔTy^{min} and ΔTy^{max} values for ($y \in \{3 \div 5\}$) and a random value ΔTy is modeled, lying in this interval and obeying the normal distribution law.

2. On the basis of experimental tests and analysis, an external or model characteristic of the engine is created. Based on this, the interpolation coefficient of equations $Me_1(\Delta Ty)$, $Me_2(\Delta Ty)$ and $Me_3(\Delta Ty)$ is determined, corresponding to the number of revolutions of the crankshaft n_1, n_2 and n_3 - a_1, b_1 and c_1; a_2, b_2 and c_2; a_3, b_3 and c_3. In this case, the number of revolutions n_1, n_2 and n_3 is determined as follows:

• n_1 revolutions per minute for maximum engine torque up to n_{max},

• n_3 the number of revolutions that provide the engine with maximum power n_{max},

• n_2 is half the sum of n_1 and $n_3$values, ie. it is equal to

$$n_2 = \frac{n_1 + n_3}{2} = \frac{n_{\max} Me + n_{\max} Ne}{2}.$$

3. Based on the found values of the coefficient a_1, b_1 and c_1, Me_1 ordinates are determined, Me_2 ordinates based on a_2, b_2 and $c_2$and Me_3 ordinates based on a_3, b_3 and c_3.

4. The values of the coefficients $a(\Delta Ty)$, $b(\Delta Ty)$ and $c(\Delta Ty)$, interpolating the torque equation of the engine $Me(n)$, are calculated by formulas (1).

5. For the type of vehicle moving along the route, the correction factor *Км*, the control factor $\boldsymbol{Cy}$ is the number of gears for transmission y, the number of gears of the main transmitter $\boldsymbol{Cay}$, the transmission efficiency is η, the dynamic coefficient and the vehicle wheel rolling coefficient r_g, r_F values are determined.

6. In the y-gear of the gearbox, the coefficients of the traction force equation of $Ay(\Delta Ty)$, $By(\Delta Ty)$ and $Cy(\Delta Ty)$ values are calculated by expressions (9).

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