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EXPERIMENTAL EVALUATION OF METHODS FOR CALCULATING VEHICLE ACCELERATION TIME

The article demonstrates that the advancement of automobiles inevitably leads to a dramatic increase in the complexity of the mathematical description of their motion. Modern vehicles are complex technical systems comprising numerous interconnected components and subsystems - electronic, mechanical, hydraulic, and pneumatic - each of which influences the overall dynamic properties of the vehicle. Consequently, the use of simple analytical relationships becomes ineffective: they do not fully account for the real physical processes occurring during acceleration, braking, gear shifting, or changing road conditions. To calculate acceleration time standards during vehicle diagnostics, it is advisable to use numerical methods that allow for the consideration of a large number of parameters, including engine power and torque, transmission characteristics, air and rolling resistance, mass distribution, and road surface conditions. Such methods ensure the required accuracy regardless of the complexity of the mathematical model and allow for the description of nonlinear processes, transient conditions, and random parameter deviations.

The use of numerical models also opens up extensive opportunities for conducting virtual tests and preliminary predictions of vehicle performance without the need for costly in-kind experiments. This is especially important when developing automatic diagnostic systems that can analyze unit operating parameters in real time and identify deviations from the norm. Furthermore, numerical methods make it possible to optimize vehicle design at the design stage, select optimal engine and transmission operating modes, reduce fuel consumption, and improve driving safety. The ideas presented in the article are supported by calculations of specific examples and a comparison of the obtained results with experimental data, demonstrating the validity and practical applicability of the proposed approach. The results of this work can be used in the development of intelligent technical control systems, modeling vehicle dynamics, and improving methods for assessing their performance.

Key words: car, wheel, acceleration, calculation, numerical methods, discretization, step, Hyundai Accent, Honda Accord.

INTRODUCTION

The modern automotive and transportation industries are increasingly focused on reducing their negative impact on the environment, particularly thermal and greenhouse emissions. Addressing this challenge is directly linked to the need to reduce fuel consumption, especially hydrocarbon fuel. An important area of development in this area is improving the theoretical foundations of vehicle and engine design, which allows for the creation of more fuel - efficient and environmentally friendly vehicles.

Maintaining the technical condition of rolling stock is a key task for transportation workers. This requires promptly identifying and eliminating faults, which is achieved through regular diagnostics and forecasting of vehicle technical conditions. Modern vehicles are typically equipped with built-in self-diagnostic systems capable of monitoring the condition of key components and assemblies. However, in Ukraine, the share of such vehicles remains relatively small. According to [1], the average age of the passenger car fleet is approximately 15.5 years, indicating a significant degree of wear and tear and obsolescence.

Furthermore, the number of specialized traction testers required for accurate diagnostics remains limited. In this regard, simplified methods for checking the technical condition of a vehicle, which can be performed directly by the user without the use of complex equipment or the need for specialized knowledge, are particularly important. Such methods include, for example, determining the vehicle's acceleration or coastdown time and subsequently comparing the obtained values with established standards [2]. These approaches allow for a quick and accurate assessment of the engine, transmission, and chassis, making them particularly relevant in conditions where diagnostic resources are limited.

AIM AND TASKS OF THE RESEARCH

Traditionally, the unsteady motion of a car during acceleration or coasting is described using a differential equation:

$$\frac{dv}{dt} = j_a = \frac{P_p}{m_a \cdot \delta} = \frac{P_к - \Sigma P_c}{m_a \cdot \delta} \tag{1}$$

where P_p – is the force accelerating the vehicle;

$P_к$ – is the circumferential force on the drive wheels;

ΣP_c – is the total resistance force to movement;

m_a – is the mass of the vehicle with the load;

δ – is the coefficient for taking into account rotating masses (the product $m_a * \delta$ is the reduced mass of the vehicle).

$$P_к = \frac{M_e \eta_{mp} i_к i_0}{r_к}; \quad \Sigma P_c = \pm P_i + P_f + P_w; \quad P_i = m_a g \cdot i;, \qquad P_f = m_a g \cdot f;$$

$$P_w = 0{,}5 \rho C_x F \cdot v^2 \tag{2}$$

where M_e – is the effective engine torque, Nm;

η_{tr} – is the transmission efficiency;

$i_к$, i_0 – are the gear ratios of the transmission and final drive;

$r_к$ – is the dynamic radius of the wheel, m;

P_i, P_f, P_w – are the downward rolling resistance, rolling resistance, and air resistance, respectively, in N;

g – is the acceleration due to gravity, 9.81 m/s²;

i – is the slope;

f – is the rolling resistance coefficient;

ρ – is the air density, kg/m³;

C_x – is the aerodynamic drag coefficient;

F – is the frontal area of the vehicle, m²;

v is the speed, m/s.

RESEARCH RESULTS

The process of simulating the unsteady motion of a vehicle on a roller rig is described in a virtually identical manner. The fundamental difference is that in this case, it's not the vehicle itself that accelerates, but the rig's rollers, which are connected to the flywheels and other rotating masses. The system is subject to a different type of resistance than when the vehicle is moving on the road.

Differential equation (1), which describes the dynamics of such a process, is solved relatively simply in cases where all its parameters and functions depend on velocity to a power of no greater than two. After combining the coefficients for equal powers of velocity and dividing the resulting expression by the total reduced mass of the system, the equation can be represented in the following form:

$$\frac{dv}{dt} + Av^2 + Bv + C = 0 \quad \text{or} \quad \frac{dv}{Av^2 + Bv + C} = -dt.$$

The integral of the right side is –t. The integral of the left side:

$$\int \frac{dv}{Av^2 + Bv + C} = \frac{1}{\sqrt{-\Delta}} \ln\left|\frac{2Av + B - \sqrt{-\Delta}}{2Av + B + \sqrt{-\Delta}}\right|,$$

Where $\Delta = 4AC - B^2$, and $\Delta < 0$.

$$t = \frac{1}{\sqrt{-\Delta}} \left(\ln\left|\frac{2Av_2 + B - \sqrt{-\Delta}}{2Av_2 + B + \sqrt{-\Delta}}\right| - \ln\left|\frac{2Av_1 + B - \sqrt{-\Delta}}{2Av_1 + B + \sqrt{-\Delta}}\right| \right) \tag{3}$$

If $\Delta > 0$, then

$$t == \frac{1}{\sqrt{\Delta}} \left(arctg \frac{2Av_2 + B}{\sqrt{\Delta}} - arctg \frac{2Av_1 + B}{\sqrt{\Delta}} \right) \tag{4}$$

DISCUSSION OF THE RESEARCH RESULTS

This method ensures satisfactory agreement between calculated data and experimental results in cases where the dependence of engine torque on engine speed can be accurately approximated by a square parabola. However, even for carbureted engines, such a simplified description of the torque characteristic is only a rough approximation and does not always yield reliable results.

In situations where the differential equation of dynamics includes third-degree terms, E. A. Belogurov [2] proposed the following general solution:

$$t = \left(\frac{\ln(v_1 - y_1)}{\ln(v_2 - y_1)}\right) + \frac{B_1}{2}\left(\frac{\ln\left|v_1^2 + pv_1 + q\right|}{\ln\left|v_2^2 + pv_2 + q\right|}\right) + \left(C_1 - \frac{B_1 p}{2}\right)\frac{2}{\sqrt{4q - p^2}}\left(arctg\frac{2v_1 + p}{\sqrt{4q - p^2}} - arctg\frac{2v_2 + p}{\sqrt{4q - p^2}}\right) \quad (5)$$

The resulting formula is already significantly cumbersome and inconvenient for practical application. Meanwhile, for modern injection engines with electronic control systems, the dependence of torque on crankshaft speed is becoming increasingly complex. In such cases, the torque curve is described not by simple second- or third-degree polynomials, but by expressions of the fourth to sixth degree, or even by several functions with a piecewise approximation (Fig. 1). A similar situation is observed for driving resistances, which also have a complex nonlinear dependence on vehicle speed.

With such a complex nature of parameter changes, the use of labor-intensive analytical approximations and subsequent integration of cumbersome expressions becomes pointless. This approach requires significant computational effort and does not guarantee high accuracy, since the original dependences are still only an approximate description of real processes. The result is only a rough estimate of the desired quantities, not their reliable values, which reduces the practical value of the obtained results.

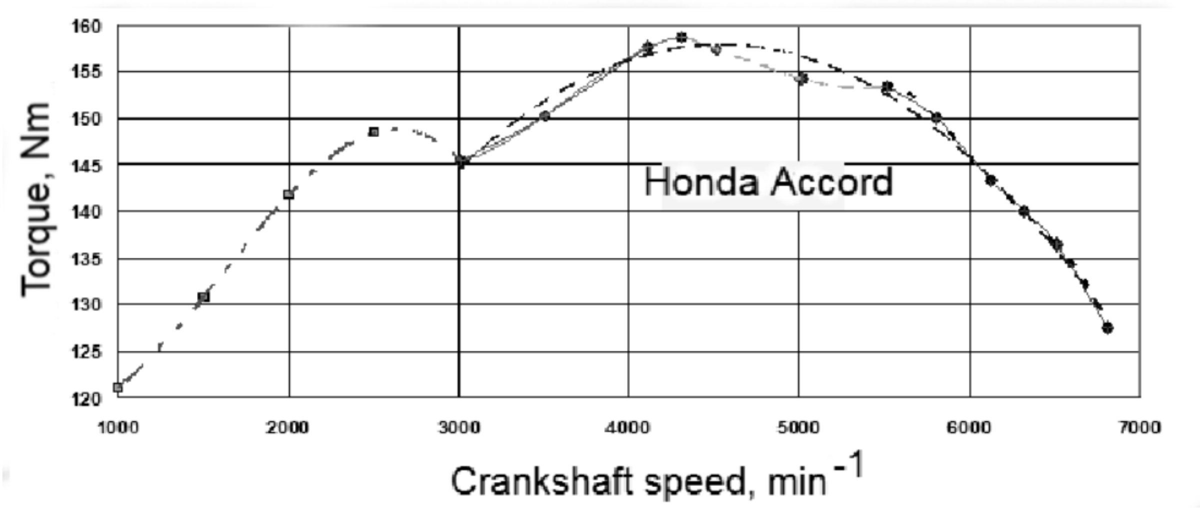

Fig. 1. Torque on the drive wheels of the Honda Accord [3].

Table 1

Piecewise approximation of characteristic sections of the torque curve at the drive wheels of a Honda Accord

Speed range	Approximation formula
From 1000 to 3009	$y = -7{,}1602\cdot10^{-9}x^3 + 3{,}4648\cdot10^{-5}x^2 - 3{,}3273\cdot10^{-2}x + 126{,}88$
From 3009 to 6806	$y = -5{,}7073\cdot10^{-6}x^2 + 5{,}1800\cdot10^{-2}x + 40{,}369$
From 3009 to 4311	$y = -8{,}0338\cdot10^{-9}x^3 + 8{,}7483\cdot10^{-5}x^2 - 3{,}0390\cdot10^{-1}x + 486{,}63$
From 4311 to 5518	$y = 4{,}0051\cdot10^{-9}x^3 - 5{,}5474\cdot10^{-5}x^2 + 2{,}4920\cdot10^{-1}x - 205{,}54$
From 5518 to 6806	$y = -3{,}1948\cdot10^{-11}x^4 + 7{,}8463\cdot10^{-7}x^3 - 7{,}2183\cdot10^{-3}x^2 + 29{,}461x - 44839$

Validation of Numerical Methods. Numerical methods for solving vehicle dynamics problems are well known and widely used in practice. The examples below examine real-world cases of vehicle diagnostics, particularly when testing their traction characteristics. The main goal of such diagnostic procedures is to determine the actual engine torque as a fraction of its nominal value. This fraction is determined by comparing the calculated vehicle acceleration time with the experimentally measured one.

The purpose of this analysis is to evaluate the accuracy of the numerical method—specifically, to identify and quantify two error components: the error associated with the numerical solution of the differential equation and the error caused by the approximation of the initial relationships.

Example 1. It is necessary to evaluate methods for calculating the acceleration time of a Honda Accord on a road section from 60 to 100 km/h (the so-called elasticity test). During field tests conducted by Autoreview magazine [4], the car demonstrated an acceleration time of 6.7 seconds in Drive mode.

The torque at the vehicle's drive wheels, i.e., the torque measured by the engine's external speed characteristic (ESC) minus transmission losses, is shown in Figure 1. Figure 2 shows the automatic transmission (AT) gear shift sequence during acceleration. The graph shows that acceleration from 60 to 100 km/h is accomplished in second gear, with the crankshaft speed range being from 3179 to 5298 min^{-1}.

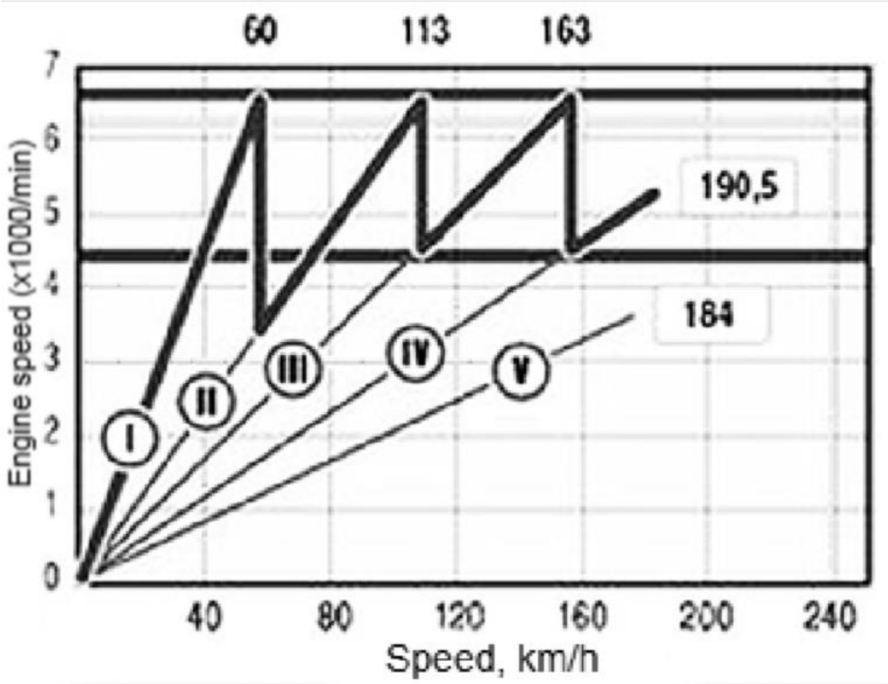

Fig. 2. Acceleration graph of the Honda Accord [4]

The vehicle's final drive ratio is i_0 = 4.437. Goodyear UG 500 205/55 R16 tires have a dynamic radius r□ = 0.307 m. These tires are classified as ECO in terms of rolling resistance [5].

The vehicle's unladen weight, determined based on weighing results at the test site, is 1,280 kg. Since information on the actual load is unavailable, the minimum standard load for a passenger car of 180 kg, is assumed in the calculations. Therefore, the total vehicle weight, including the driver, passengers, and measuring equipment, is m_a = 1,460 kg.

The reduced mass of the rotating wheels is 43 kg, and that of the driven transmission elements is 6.4 kg (determined experimentally using the bifilar suspension and coasting methods). The calculated effective mass of the engine and transmission in second gear is 149 kg. Consequently, the total effective mass of the vehicle during acceleration is 1,658 kg.

The manufacturer's stated aerodynamic drag coefficient is C_x = 0.31 [6]. However, according to the results of processing the experimental coastdown data, performed using the methodology [5] based on tests conducted by Autoreview magazine, the actual value of the coefficient was C_x = 0.381.

Vehicle frontal area (calculated):

$$F = 0{,}84 \cdot B \cdot H$$

where B, H are the width and height of the vehicle, respectively, in m.

$$F = 0{,}84 \cdot 1{,}715 \cdot 1{,}44 = 2{,}074 \text{ м}^2$$

The air density on the day of the Autoreview experiment is unknown. Therefore, the standard average value commonly used in similar problems, ρ = 1.2 kg/m³, was adopted in the calculations.

The dependence of the torque on the drive wheels on the crankshaft speed in the range of 3000–6800 min^{-1} is shown in Figure 1 (dashed line, second line of the caption). For this frequency range, this dependence is approximated by the following equation:

$$M_e = -5{,}71 \cdot 10^{-6} \cdot n^2 + 5{,}18 \cdot 10^{-2} n + 40{,}4. \quad (6)$$

Substituting (5) into (2), we obtain the dependence of the traction force on the torque; in numbers for the 2nd gear:

$$P_к = \frac{M_e \cdot 1{,}534 \cdot 3{,}945}{0{,}303} = M_e \cdot 17{,}8 = -1{,}140 \cdot 10^{-4} \cdot n^2 + 1{,}035 \cdot n + 806. \quad (7)$$

Further calculations will become more convenient if we switch from crankshaft revolutions to vehicle speed in m/s:

$$n = 9{,}55 \frac{v \cdot i_k \cdot i_0}{r_k} = 9{,}55 \frac{v \cdot 1{,}534 \cdot 3{,}945}{0{,}303} \approx 191 v. \quad (8)$$

Then the dependence of the traction force on the speed will take the form

$$P_к = -4{,}15 \cdot v^2 + 197 \cdot v + 806. \quad (9)$$

The equation for the dependence of air resistance on speed:

$$P_w = 0{,}5 \rho C_x F \cdot v^2 = 0{,}5 \cdot 1{,}2 \cdot 0{,}38 \cdot 2{,}074 \cdot v^2 = 0{,}473 v^2 \quad (10)$$

Based on the vehicle coasting [4], the rolling resistance coefficient f = 0.0122 was calculated for a speed of 6.6 m/s (23.75 km/h), therefore the f(v) dependence was adopted according to the upper curve for ECO tires [7], which gives a close value of 0.01238 for this speed:

$$f(v) = 1{,}36 \cdot 10^{-7} \cdot v^2 - 7{,}54 \cdot 10^{-6} \cdot v + 0{,}01248 \quad (11)$$

Coefficient that takes into account the reduction in rolling resistance when tires are not fully loaded (at G_{max} = 615 kg for tires with index 91):

$$K_G = 1{,}3 - 0{,}3 \cdot 4 \cdot G_{max} / m_a = 1{,}3 - 0{,}3 \cdot 4 \cdot 615 / 1460 \approx 0{,}80 \quad (12)$$

Then the dependence of the rolling resistance force on speed:

$$P_f = m_a g \cdot f(v) \cdot K_G = 2{,}73 \cdot 10^{-2} \cdot v^2 - 0{,}471 \cdot v + 152. \quad (13)$$

The force accelerating the car:

$$P_p = P_к - \Sigma P_c = -4{,}11 \cdot v^2 + 176 v + 565. \quad (14)$$

The equation of motion of a car during acceleration:

$$\frac{dv}{dt} = j_a = -2{,}19 \cdot 10^{-3} v^2 + 9{,}39 \cdot 10^{-2} v + 3{,}22 \cdot 10^{-1} \ . \qquad (15)$$

The solution using formula (3) yielded a time value of 6.6996321 s, which is closest to the 6.7 s measured in the test, with the engine torque reduced to 89.4%. The results of the numerical calculations are shown in Table 2.

Table 2

Results of calculating acceleration times from 60 to 100 km/h using a numerical method with different speed increments

Step, km/h	Time from 60 to 100 km/h	Absolute calculation error Δt, s	Relative error of calculation ε, %
By solving the differential equation	**6,6996321**	0	0
0,5	6,699673	$4{,}072 \cdot 10^{-5}$	0,000608
1	6,699795	0,000163	0,002432
2	6,700284	0,000652	0,009733
5	6,703712668	0,004081	0,060907
10	6,716027314	0,016395214	0,244718124
20	6,766356427	0,066724327	0,995940172

As the table shows, even at a 10 km/h increment, the numerical calculation result deviates from the differential equation solution by less than 0.02 s. This is a relatively small error, as acceleration time is typically measured to the first decimal place. Therefore, using a numerical calculation instead of a general solution to the differential equation is acceptable.

In the example shown, the diagnosis was: torque is reduced to 0.894 of the nominal value. This value is excessively low for a practically new car, too close to the maximum of 0.85. It is possible that the diagnosis is incorrect, caused by an excessively rough approximation of the section of the torque curve from 3000 to 6800 min–1. To estimate the error caused by the approximation, the torque characteristic was represented as a piecewise smooth curve, broken down into characteristic sections, and approximated by polynomials of degrees 3–4 (Fig. 1). The acceleration calculation was performed in 5 km/h increments – as shown in Table 1, this yields an error of less than 0.1%. The calculated acceleration time at a torque of 0.894 was 7.04 s (versus 6.7 s for the general approximation of the 3000…6800 min–1 section). A discrepancy of 0.34 s is 80 times greater than the error of the numerical method of 0.00408 s.

The diagnosis made using the piecewise approximation is 93.3% of the nominal value. Since the car is new, this is a more probable estimate.

A more refined solution in general would require solving three differential equations for three sections, where the dependencies are described by polynomials of degrees 3 and 4, and the integrals are not reducible to tabular ones. Under these conditions, the numerical method should be considered more rational.

Example 2. Evaluate methods for calculating the acceleration time in fourth gear (iк = 1.031) for the drive wheels of a Hyundai Accent vehicle using the PDS KHADII test rig (certified by the Institute of Metrology, certificate No. 100-2151/2006). In our experiment, the average acceleration time from 50 to 70 km/h was 1.60 s with a load (additional resistance) of Pн = 600 N created by the rig's hydraulic brake.

The engine's maximum torque curve is shown in Fig. 3. The actual torque as a percentage of the nominal torque is Km = 0.9835 (our measurement on the rig in steady-state mode). The final drive ratio is i0 = 4.412. We will adopt the efficiency of the mechanical transmission as ηтр = 0.92. The dynamic radius of Fulda Montero 2 195/65 R15 91T tires on a working roller with a diameter of 0.24 m is rd = 0.295 m. The load on the drive wheels, taking into account the operator's weight, is 800 kg. The reduced mass of the rig is 200 kg (included in the design, verified using the drop weight method). The reduced mass of the two drive wheels and the driven part of the transmission is 27.5 kg (our measurements). The calculated reduced mass of the engine with the driving part of the transmission in 4th gear is 61.8 kg. Total reduced mass of the system tpr.s = 289.3 kg.

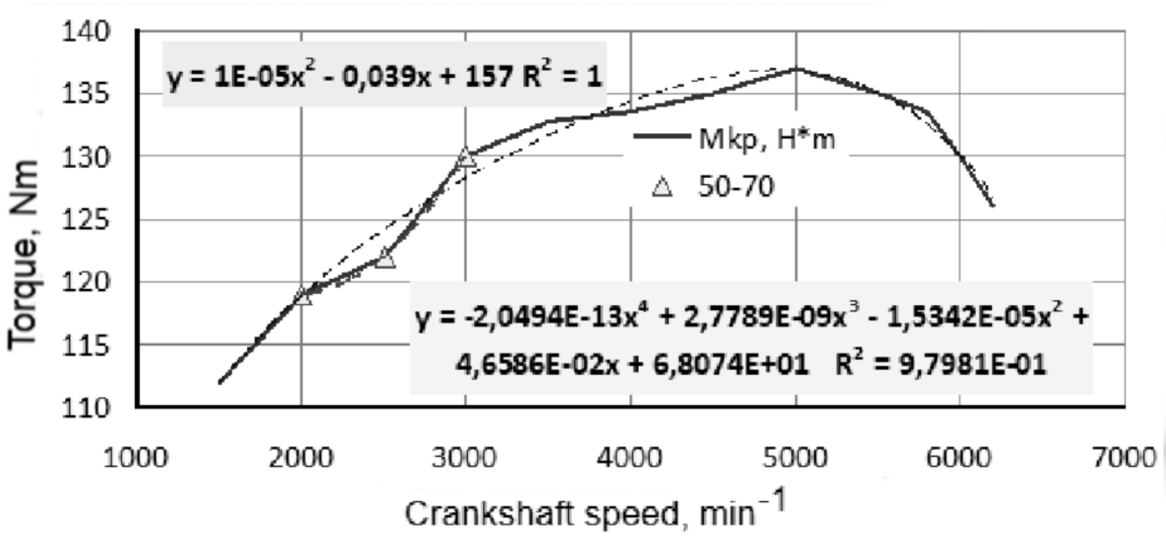

Fig. 3. External speed characteristics of the Hyundai Accent 1.4 petrol engine [8]

The generally accepted range for testing traction properties from 50 to 70 km/h corresponds to speeds from 2045 to 2863 min–1. This section is marked in Fig. 3 with triangular markers and is approximated by a second-degree polynomial.

$$M_e = 1 \cdot 10^{-5} n^2 - 3{,}9 \cdot 10^{-2} n + 157 \tag{16}$$

Traction force on this section in 4th gear:

$$P_к = \frac{M_e \cdot i_к \cdot i_0 \cdot \eta_{mp}}{r_d} \cdot K_m = \frac{M_e \cdot 1{,}031 \cdot 4{,}412 \cdot 0{,}92}{0{,}295} \cdot 0{,}9835 = \\ = M_e \cdot 13{,}95 = 1{,}372 \cdot 10^{-4} \cdot n^2 - 0{,}535 n + 2154. \tag{17}$$

Relationship between the crankshaft rotation speed and the linear speed of the wheel contact with the rollers in m/s:

$$n_{eng} = 9{,}55 \frac{v \cdot i_k \cdot i_0}{r_d} = 9{,}55 \frac{v \cdot 1{,}031 \cdot 4{,}412}{0{,}295} \approx 147 v. \tag{18}$$

Traction force versus speed:

$$P_к = 2{,}975 \cdot v^2 - 78{,}8 \cdot v + 2154. \tag{19}$$

The dependence of the total resistance force to rotation of the wheels and rollers of the stand on the speed (our measurement):

$$P_\Sigma = 0{,}460 \cdot v^2 - 8{,}98 \cdot v + 404. \tag{20}$$

Accelerating force:

$$P_р = P_к - P_\Sigma - P_н = 2{,}515 \cdot v^2 - 69{,}8 \cdot v + 1150. \tag{21}$$

The acceleration equation for a system with reduced mass tpr.s = 289.3 kg:

$$\frac{dv}{dt} = j_a = 8{,}693 \cdot 10^{-3} \cdot v^2 - 2 \cdot 0{,}1206 \cdot v + 5{,}06 . \qquad (22)$$

Here Δ>0, and the solution using formula (4) yielded an acceleration time from 50 to 70 km/h of 1.6009506 s. The results of the numerical calculation are shown in Table 3.

Table 3 Results of calculating acceleration time from 50 to 70 km/h using a numerical method with different speed increments

Step, km/h	Time from 50 to 70 km/h	Absolute calculation error Δt, s	Relative error of calculation ε, %
By solving the differential equation	**1,600951**	0	0
1	1,600897	–0,00005	–0,0034
2	1,600734	–0,00022	–0,0135
5	1,599599	–0,00135	–0,0844
10	1,595568	**–0,00538**	–0,3362

An attempt to apply a general approximation of the torque curve to this example using a fourth-degree polynomial (Fig. 3) yielded a clearly erroneous result of 2.2 s.

In both examples, the acceleration was described by a second-degree polynomial. If we analyze more complex cases, using fourth- to sixth-degree polynomials, the general solution will be cumbersome and difficult to use, and the numerical method will be the only viable solution.

CONCLUSIONS

The numerical method can yield negligible errors even with a fairly coarse partitioning of the range into intervals.

The error caused by a generalized approximation can be excessively large, so it is better to use a piecewise approximation.

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Ю.В. Зибцев, П.А. Ворошилов, І.М. Шевердін. **Експериментальна оцінка методів розрахунку часу розгону транспортного засобу**

У статті показано, що розвиток автомобілів неминуче призводить до різкого зростання складності математичного опису їхнього руху. Сучасні транспортні засоби – це складні технічні системи, що складаються з численних взаємопов'язаних компонентів та підсистем – електронних, механічних, гідравлічних та пневматичних – кожна з яких впливає на загальні динамічні властивості транспортного засобу. Отже, використання простих аналітичних співвідношень стає неефективним:

вони не повністю враховують реальні фізичні процеси, що відбуваються під час розгону, гальмування, перемикання передач або зміни дорожніх умов. Для розрахунку норм часу розгону під час діагностики транспортного засобу доцільно використовувати числові методи, що дозволяють враховувати велику кількість параметрів, включаючи потужність і крутний момент двигуна, характеристики трансмісії, опір повітря та коченню, розподіл маси та стан дорожнього покриття. Такі методи забезпечують необхідну точність незалежно від складності математичної моделі та дозволяють описувати нелінійні процеси, перехідні режими та випадкові відхилення параметрів.

Використання числових моделей також відкриває широкі можливості для проведення віртуальних випробувань та попереднього прогнозування характеристик транспортних засобів без необхідності проведення дорогих натуральних експериментів. Це особливо важливо при розробці автоматичних діагностичних систем, які можуть аналізувати робочі параметри агрегатів у режимі реального часу та виявляти відхилення від норми. Крім того, числові методи дозволяють оптимізувати конструкцію транспортного засобу на етапі проектування, вибирати оптимальні режими роботи двигуна та трансмісії, зменшувати витрату палива та підвищувати безпеку руху. Ідеї, представлені в статті, підтверджуються розрахунками на конкретних прикладах та порівнянням отриманих результатів з експериментальними даними, що демонструє обґрунтованість та практичну застосовність запропонованого підходу. Результати цієї роботи можуть бути використані при розробці інтелектуальних систем технічного керування, моделюванні динаміки транспортних засобів та вдосконаленні методів оцінки їхньої продуктивності.

Ключові слова: автомобіль, колесо, прискорення, розрахунок, числові методи, дискретизація, крок, Hyundai Accent, Honda Accord.

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