EVALUATION OF THE FUEL EFFICIENCY A WHEELED TRACTOR EQUIPPED WITH AN ECVT TRANSMISSION DURING THE TRANSPORTATION OF PARTIALLY FILLED TANKS

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Abstract - Transport operations are one of the key components of the operational load of wheeled tractors in the agro-industrial complex. In addition to traction work during soil cultivation, tractors are widely used for inter-farm transportation of agricultural products, fertilizers, seeds, materials, and auxiliary equipment. The share of transport operations in the annual operating time of tractors can account for a significant portion of total activity. With increasing production intensity and stricter environmental requirements, the implementation of modern solutions, in particular eCVT transmissions, becomes especially relevant, as they improve energy efficiency and reduce negative environmental impact. The purpose of this article is to investigate changes in instantaneous fuel consumption of a wheeled tractor equipped with an eCVT transmission during the transportation of a partially filled towed tank by means of simulating standardized driving cycles. The research objective is achieved through the use of a mathematical model of a tractor-trailer unit operating under transport conditions. The practical significance of the study lies in determining the critical filling level of the towed tank at which the maximum increase in instantaneous fuel consumption is observed, caused by oscillatory effects arising during the operation of a container with a free liquid surface. The scientific novelty of the work consists in establishing, for the first time, the relationships between the fuel efficiency of a wheeled tractor equipped with an eCVT transmission and the transportation of a partially filled tank, which makes it possible to reduce the relative increase in instantaneous fuel consumption at critical filling levels when simulating energy-intensive driving cycles.

Keywords: Wheeled tractor, Engine, eCVT transmission, Transport operation, Tank, Free liquid surface, Fuel efficiency, Driving cycles, Modeling.

1. Introduction

The transport component is one of the key elements in the structure of operational loads of wheeled tractors in the agro-industrial complex. Alongside traction operations during soil cultivation, tractors are widely involved in intra-farm and inter-farm transport and logistics tasks. These include the transportation of grain crops, root crops, and green biomass, the delivery of fertilizers, seeds, spare parts, and tools, as well as the execution of logistical functions during the harvesting period. According to field studies, the share of transport modes in the annual operating time of tractors can range from 10 to 60%, especially for low-power and general-purpose machines operating under conditions of fragmented land use.

Transport activities encompass the movement of cargo, materials, and auxiliary equipment, the supply of resources to agricultural implements, the transportation of harvested products to storage or processing facilities, as well as technological transport operations within tractor-trailer combinations. In addition, tractors perform auxiliary logistical tasks, including the delivery of fuel and technical fluids and the recovery of disabled machinery. With increasing production intensity and greater spatial dispersion of farmland, the role of transport operations in the overall load profile of tractors continues to grow.

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The tightening of environmental requirements and the need to reduce pollutant emissions stimulate the adoption of modern technical solutions. A promising direction of development is the application of eCVT transmissions, which provide improved energy efficiency in transport modes and a reduction in negative environmental impact.

2. State of Transmission Design Research

Efforts to improve energy conservation, reduce emissions, and enhance the energy efficiency and environmental performance of agricultural machinery have become one of the key directions in the development of the modern agro-industrial complex, while the existing limitations and shortcomings of new technologies in this field constitute significant challenges that require comprehensive scientific and practical solutions [2, 3]. In order to meet contemporary requirements, manufacturers of agricultural machinery are increasingly following the example of the automotive industry by moving toward electrification [4].

Although the configurations of electrified power units become significantly more complex and require the integration of advanced electronic control systems, electrification provides an important advantage – the ability to decouple external loads from the engine. This decoupling allows the engine to operate more efficiently, reduces fuel consumption, and improves the overall environmental performance of the equipment [5, 6]. These benefits have driven a growing interest in electrified and hybrid powertrain solutions for agricultural vehicles, particularly in the context of sustainable development and stringent emission regulations.

Purely electric tractors require high-capacity batteries to ensure prolonged operation and high performance. However, current technological limitations in the production of battery systems create substantial difficulties for the practical deployment of high-power electric machines [7, 8]. Specifically, the limited energy capacity, significant weight, and volume of batteries adversely affect operational duration, vehicle mobility, and overall operating efficiency, complicating their use in intensive agricultural operations where tractors must work continuously under heavy loads for extended periods. These constraints highlight the challenges of relying solely on battery electric propulsion in demanding field conditions, particularly when long operational autonomy and rapid refueling are crucial.

In this context, hybrid tractors present a promising solution, as they combine the advantages of an internal combustion engine with those of an electric motor, helping to offset the limitations of battery systems and providing stable performance across traction and transport operating modes [9]. Hybrid powertrains can utilize the internal combustion engine to generate electrical energy for the electric motor or directly supply mechanical power to the drive wheels, depending on the selected architecture. Such flexibility enables better fuel economy, reduced emissions, and improved adaptability to varying field conditions, making hybrid configurations especially attractive for high-power agricultural applications.

Contemporary scientific and applied research identifies three principal types of high-power hybrid tractors: series hybrid, parallel hybrid, and combined (series – parallel) hybrid systems. Each of these architectures possesses specific design features, power transmission principles, and operational characteristics that determine their suitability for different agronomic tasks [9 – 11]. Series hybrid systems, for instance, decouple the mechanical connection between the engine and the drive wheels, using the engine primarily as a generator. Parallel hybrids, in contrast, allow both the engine and electric motor to deliver power directly to the drivetrain. Combined hybrid systems integrate both approaches to optimize performance and efficiency across a wider range of operating conditions.

Most scientific works, including studies [12, 13], focus on the behavior of hybrid transmissions during heavy traction work, which is unsurprising, as these operations are among the most energy-intensive in agricultural practice. Heavy traction tasks such as plowing, tilling, and pulling large implements require substantial continuous power and place considerable demands on both the engine and hybrid system components. However, research on transport operations – particularly the transportation of solid or liquid loads – also demands attention, since these tasks require not only adequate traction performance but also specific speed characteristics to ensure timely and efficient delivery of materials. Balancing energy use, system responsiveness, and overall efficiency during transport operations presents a distinct set of challenges compared to purely traction-focused work.

The investigation of hybrid tractor behavior is becoming increasingly relevant in modern agricultural engineering research [14], especially because the simulation of their motion using drive cycles enables the evaluation of powertrain efficiency and the influence of design decisions on operational performance parameters [15].

Such simulation-based studies facilitate the analysis of vehicle performance without the significant time and cost associated with extensive field testing.

They also allow researchers to explore a wide range of operating scenarios [16], engine load profiles, and control strategies in a controlled virtual environment.

Over the last several decades, the automotive industry has pursued similar objectives through numerous experimental campaigns conducted in real-world conditions, aimed at developing standardized drive cycles. These drive cycles are used to assess the effects of design changes on the performance of individual components, subsystems, or whole vehicles, as well as to determine exhaust emissions. Standardized drive cycles are typically developed for road vehicles and are represented as time histories of vehicle speed and road profile. They are intended to reflect typical operating conditions, focusing on real engine operating points, speed ranges, and acceleration patterns during limited periods of time.

It is essential that such cycles are applied without time scaling to avoid introducing any artificial or unrealistic alterations in the modeling and evaluation process. Considering this, it is advisable to utilize several standardized drive cycles to comprehensively verify the behavior of the power units of a tractor equipped with an electronic continuously variable transmission (eCVT). The use of multiple cycles ensures a robust assessment of system performance across different working regimes, including urban, rural, and mixed operational profiles.

3. Materials and Methods

3.1. Modeling of a Wheeled Tractor with a Towed Tank Trailer Motion

The development of a mathematical model of the motion of a machine-tractor unit (Fig. 1) performing transportation operations involving the carriage of liquid cargo requires the researcher to take into account the operating characteristics of key system components, such as the internal combustion engine, the transmission, the wheeled propulsion system, as well as the effects of mass redistribution of the liquid within the towed tank. Such a problem formulation necessitates establishing interrelations between the individual components of the system.

The description of the motion process of the machine-tractor unit should be based on the consideration of both linear and nonlinear functions. This approach makes it possible to investigate real operating conditions of the system in which inertial and force elements interact. Since the objective of the dissertation research is to identify the patterns governing changes in techno-economic performance indicators during transportation operations, it is appropriate to employ equations of a quasi-dynamic equilibrium state of the machine-tractor unit components in the model, accounting for both internal and external disturbances.

At the same time, modeling the motion of a machine-tractor unit during transportation of liquid cargo is complicated by a number of factors:

- the system dynamics are significantly affected by the movement of the liquid inside the tank, which results in variable inertial loads and oscillations of the aggregate's center of mass;
- the elastic-damping properties of the hitching device between the tractor and the trailer cause phase shifts between the tractive force in the coupling and the vehicle speed;
- the nonlinear characteristics of the tires and the wheel–road interaction lead to variable rolling resistance forces, especially when traveling over uneven surfaces or during acceleration and deceleration;
- variations in the cargo mass due to partial filling or liquid motion during travel alter the inertial parameters of the system and its dynamic stability.

During the investigation of variations in the operational performance of an internal combustion engine, scientific practice widely employs a simplified approach in which the internal working processes of converting the chemical energy of fuel into mechanical work are not considered. Within this approach, when describing the mechanical operation of the engine, it is also assumed that the harmonic components of the torque are neglected.

Under real operating conditions, the torque at the engine crankshaft is a non-constant quantity that varies within each working cycle due to the alternation of strokes, fluctuations in cylinder pressure, as well as the action of inertial forces of the valve train components. These oscillations are periodic in nature and can be represented as the sum of an average (steady) torque and harmonic (pulsating) components.

The adopted assumptions make it possible to apply the equation of dynamic equilibrium of crankshaft motion, which has been well validated [17] and is widely used in scientific studies devoted to the modeling of internal combustion engine operation:

$$J_e \cdot \frac{d\omega_1}{dt} = M_e - M_D, \tag{1}$$

where J_e is the moment of inertia of the masses referred to the crankshaft; ω_1 is the angular velocity of the crankshaft; M_e is the actual (effective) torque at the crankshaft; M_D is the resisting torque accounting for the load from the transmission, auxiliary units, and friction losses.

Within the framework of the present study, the operation of individual transmission components is described on the basis of a mathematical model representing the equations of a quasi-dynamic equilibrium state of the machine–tractor unit subsystems.

In this context, a simplified representation of the transmission is considered appropriate, as it enables efficient modeling of its behavior without excessive complexity. In particular, the transmission is modeled as a torsional vibration scheme with two driven axles, which provides the possibility of automatically determining the interaction parameters between the wheeled propulsion system and the soil. Such a simplification also makes it possible to account for the distribution of power flows in the drives of the front and rear axles of the tractor, which is essential for an accurate assessment of the energy performance of the unit.

Classically, the solution of such problems involves referring all parameters either to the internal combustion engine or directly to the wheeled drive. In the framework of the present study, the parameters were referred to the internal combustion engine, which allows for unifying the calculation scheme and ensures a correct comparison of transmission characteristics under different operating conditions. It should be noted that the choice of the transmission is justified in studies [9, 18].

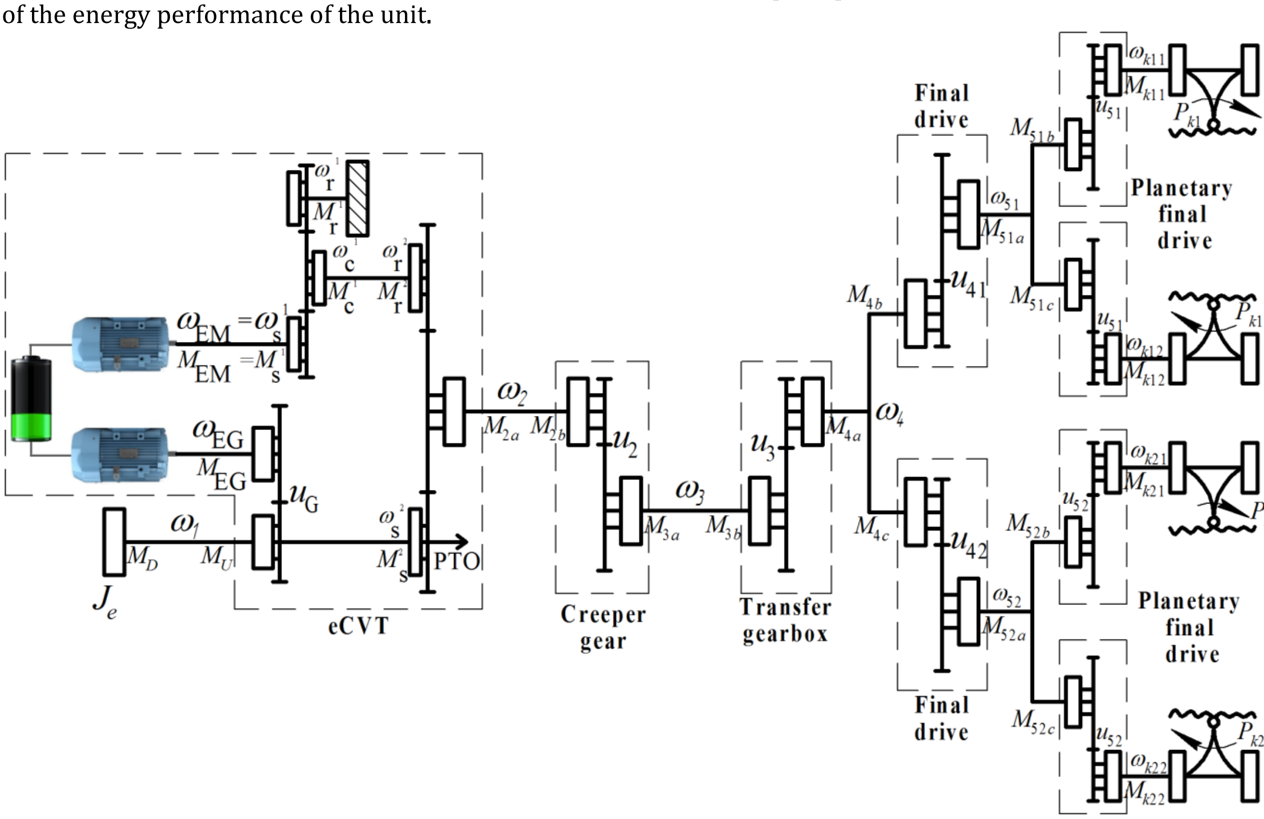

Figure 1: Kinematic scheme of an eCVT transmission a wheeled tractor

When performing a mathematical description of the motion components of a tractor-trailer unit, it should be noted that the model must be based on the motion described in absolute coordinates of three-dimensional space and account for the interaction of the main inertial, mass, and force parameters of the system under study. At the same time, spatial vibrations of the cabin, operator seat, and other local oscillatory effects have an insignificant impact on the assessment of technical and economic indicators and are therefore deliberately neglected. This approach allows for a significant simplification of the mathematical apparatus of the model, reduces the number of input parameters and computational effort, while maintaining sufficient accuracy for analyzing changes in the technical and economic performance of the tractor-trailer unit-specifically, engine power efficiency, specific fuel consumption, and unit productivity.

This enables the research to focus on the main factors determining the energy efficiency of the tractor-machine complex.

Initially, the relationship between the transmission output elements and the wheels is presented. For this purpose, the following equilibrium equations are used:

$$\begin{cases} J_{k1} \cdot d\omega_{k11}/dt = M_{k11} - 0,5P_{k1} \cdot r_{k1}; \\ J_{k1} \cdot d\omega_{k12}/dt = M_{k12} - 0,5P_{k1} \cdot r_{k1}; \\ J_{k2} \cdot d\omega_{k21}/dt = M_{k21} - 0,5P_{k2} \cdot r_{k2}; \\ J_{k2} \cdot d\omega_{k22}/dt = M_{k22} - 0,5P_{k2} \cdot r_{k2}, \end{cases}, \tag{2}$$

where J_{k1}, J_{k2} are the moments of inertia of the front and rear wheels; P_{k1}, P_{k2} are the tangential traction forces on the front and rear wheels of the tractor; r_{k1}, r_{k2} are the dynamic radii of the front and rear wheels.

Fig. 2 shows a schematic of the tractor-trailer unit with a towed tank. Several assumptions were introduced in forming the motion scheme of the tractor-trailer unit:

- Since the motion of a wheeled tractor with a towed tank is being studied, the component P_{krz} can be neglected.

This assumption is further supported by the fact that only straight-line motion will be simulated, without considering transient processes (acceleration and braking). Accordingly, vertical vibrations of the liquid are also neglected;

- When analyzing the angular motion of the tractor, it is assumed that it is influenced solely by the horizontal component of the towed tank's movement;
- In studying wheel-surface interaction, the dynamic radius r_k is assumed equal to the static radius r_c;
- The clearance in the drawbar coupling is not considered.

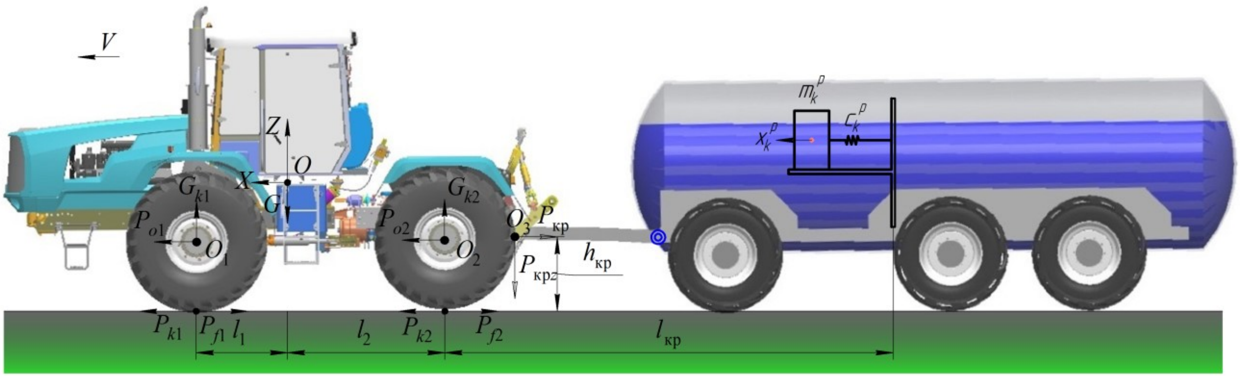

Figure 2: Calculation scheme of a wheeled tractor with a trailer tank

Determining the change in the tractor's energy performance is achieved by calculating a system of differential equations:

$$\begin{cases} m \cdot \ddot{X} = 2\left(P_{o1} + P_{o2}\right) - P_{\kappa p}; \\ m \cdot \left(\ddot{Z} + g\right) - 2\left(G_{k1} + G_{k2}\right) = 0; \\ 0.5 J_o \cdot \ddot{\varphi}_o + G_{k1}\left(X_{o1} - X\right) + G_{k2}\left(X_{o2} - X\right) = \\ = 0.5 P_{\kappa p} \cdot h_{\kappa p} - P_{o1}\left(Z_{o1} - Z\right) + P_{o2}\left(Z_{o2} - Z\right), \end{cases} \tag{3}$$

where X, Z, φ_o are generalized absolute coordinates of tractor movement; P_{o1}, P_{o2} are forces that push the frame of a wheeled tractor forward; G_{k1}, G_{k2} are loads on the wheels of the front and rear axles of a wheeled tractor acting in a vertical plane; $P_{\kappa p}$, $P_{\kappa pz}$ are horizontal and vertical components of the traction force on the hook of a wheeled tractor; x_{o1}, x_{o2}, z_{o1}, z_{o2} are horizontal and vertical coordinates of the points of the wheeled tractor frame in the absolute coordinate system.

The radial load acting on the driving wheels of the tractor is determined based on the following expressions:

$$G_{k1} = G\frac{l_2}{2L} - \left(P_{k1} - G \cdot f \cdot \frac{l_2}{L}\right) \cdot \frac{h_{\text{kp}}}{2L}; \tag{4}$$

$$G_{k2} = G\frac{l_1}{2L} - \left(P_{k2} - G \cdot f \cdot \frac{l_1}{L}\right) \cdot \frac{h_{\text{kp}}}{2L}, \tag{5}$$

where L is the wheelbase of the tractor; l_1 and l_2 are the distances from the center of gravity to the front and rear axle lines; f is the rolling resistance coefficient of the wheels; h_{kp} is the height of the point of application of the traction force on the hitch above the supporting surface.

The tangential traction force P_k is defined as follows:

$$P_{k1} = \frac{M_e \cdot u_{tr} \cdot \eta_{tr}}{r_c} \cdot \frac{2G_{k1}}{G}; \tag{6}$$

$$P_{k2} = \frac{M_e \cdot u_{tr} \cdot \eta_{tr}}{r_c} \cdot \frac{2G_{k2}}{m \cdot g}, \tag{7}$$

where u_{tr} is the overall transmission ratio of the tractor, η_{tr} is the overall efficiency of the transmission.

The next step is to determine the force that propels the chassis of the wheeled tractor. This force is related to the traction force P_k, calculated based on the wheel-surface interaction, and the rolling resistance forces P_f acting on each wheel:

$$P_{o1} = P_{ks1} - P_{f1}; \tag{8}$$

$$P_{o2} = P_{ks2} - P_{f2}. \tag{9}$$

It is commonly accepted [19, 20] to evaluate the longitudinal adhesion capability of a wheel using the following formula:

$$P_{ks} = G_k \cdot \varphi_x = G_k \cdot f\left(S_x\right). \tag{10}$$

where φ_x is the longitudinal adhesion coefficient; S_x is the relative longitudinal slip of the wheel.

The only remaining undefined parameter is the hitch traction force $P_{кр}$. Its magnitude depends on the disturbing forces acting in the horizontal plane and requires a separate mathematical description.

3.2. Determination of the Components Fluid Motion in a Tractor Tank with a Free Surface

On the side of the running gear, the tank shell is subjected to sources of low-frequency disturbances in the form of force and torque impulses. Such influences can be random, for example when overcoming road irregularities, or regular, such as during the operation of a vacuum pump, leading to the generation of decaying free or non-decaying forced liquid oscillations. The primary source of free low-frequency oscillations, as well as the medium for converting forced oscillations, is the free surface of the liquid. Kinematic disturbance waves propagate from the tank's side walls, causing oscillatory movements of the liquid surface. This is accompanied by changes in the normal pressure in the phase opposite to the vertical displacements of the surface, caused by gravitational forces and surface tension, resulting in free oscillations. As the liquid depth increases, the amplitude of free oscillations decreases according to an exponential law. Viscous friction of the liquid causes damping of oscillations and limits the amplitude of forced oscillations in resonance zones.

It is assumed that all types of linear liquid oscillations can be described within a single approach based on the determination of natural frequencies and the shapes of partial oscillations. The motion of the rigid shell includes three translational and three rotational degrees of freedom. Shell deformations and elastic connections are neglected, as their characteristic frequencies significantly exceed 100 Hz. During vertical oscillations, liquid sloshing does not occur, and the frequency is determined by its compressibility and is much higher, so such oscillations are not considered. The remaining oscillation modes are grouped according to the type of motion: oscillations in the zOy plane, oscillations in the xOy plane, and rotation about the Oy axis. For the first two groups, the velocity vector field is planar, whereas for rotational oscillations it is three-dimensional.

The formation of the physical model of liquid motion in a closed container (Fig. 3) is based on the well-known Navier–Stokes equation, which accounts for internal friction proportional to viscosity, supplemented by the consideration of a conditional force associated with the external friction of the liquid against the tank walls.

Within the interior of the container, this conditional force is transmitted through partial layers of liquid moving together with the walls [21].

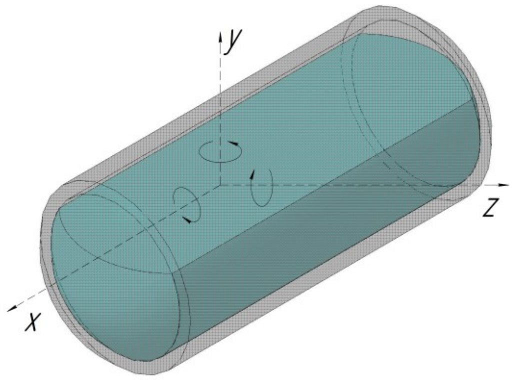

Figure 3: Equivalent scheme of a tank for transporting liquids

In [21], scientific principles are presented in which the cylindrical shape of the container is proposed to be equivalently replaced with a shape according to the following conditions: for H ≤ R, a rectangular shape is used (Fig. 4a), and for H > R, a trapezoidal shape is applied (Fig. 4b).

By determining the relative mass δM [21], it becomes possible to calculate the oscillatory motion in the form of a conventional equation for a single-mass oscillating system. The elastic–mass characteristics of the partial oscillators in the *xOy* plane are defined by the following equations:

$$m_k^p \ddot{x}_k + f_k^p \cdot \dot{x}_k + c_k^p \cdot x_k = m_k^p \cdot g, \quad k = 1,2,3,\dots; \quad (11)$$

$$m_k^p = \delta M_k \cdot m_p; \quad (12)$$

$$m_p = 2\rho \cdot S \cdot L; \quad (13)$$

$$f_k^p = 2d_p \cdot \nu_k \cdot m_k^p; \quad (14)$$

$$c_k^p = m_k^p \cdot (2 \cdot \pi \cdot \nu_k)^2, \quad (15)$$

where m_k^p, m_p are the partial mass and total mass of the liquid; $\ddot{x}_k$, $\dot{x}_k$, x_k are acceleration, velocity, and displacement of the liquid layer; f_k^p is the damping coefficient of the *k*-th liquid layer; d_p is the logarithmic decrement of damping of these oscillations [21]; c_k^p is the stiffness (or elasticity) coefficient of the *k*-th liquid layer oscillations.

Considering the discrete model (Equation (11)) of liquid motion in a closed container, it is possible to compute the magnitude of the hitch traction force $P_{кр}$

$$P_{кр} = \sum_{k=1}^{3} \left[f_k^p \cdot \left(\dot{X} - \dot{x}_k \right) + c_k^p \cdot \left(X - x_k \right) + m_k^p \cdot g \cdot f \right]. \quad (16)$$

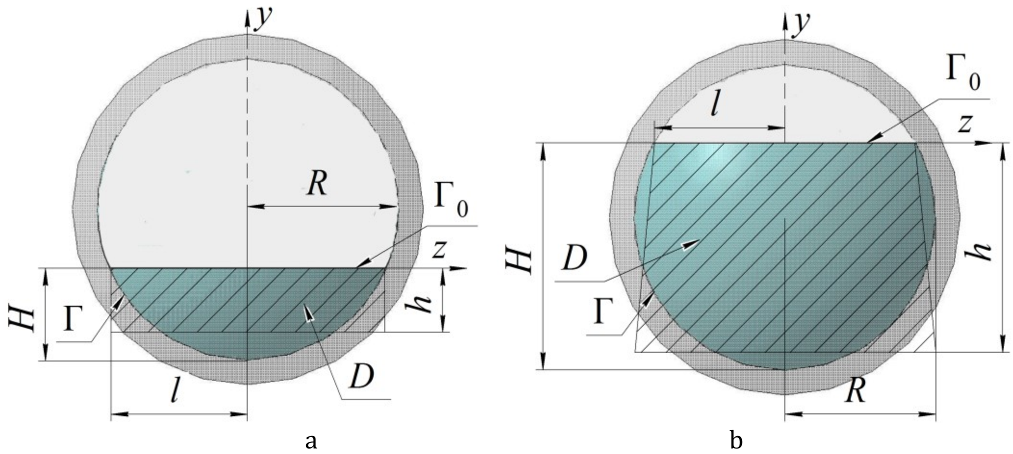

Figure 4: Simplification of the shape of a cylindrical container into a rectangular one (a) and a trapezoid (b)

3.3. Researched Driving Test Cycles for Wheeled Tractors

One of the key reasons for using standardized driving test cycles is the practical impossibility of conducting a full-factorial experiment that would cover the entire spectrum of possible combinations of crankshaft rotational speeds, loads, temperature conditions, fuel injection parameters, boost modes, and electronic control system algorithms. For a highly complex engine, a full-factorial experimental plan would require thousands or even tens of thousands of test points, making it practically unfeasible due to excessive time, fuel, engine wear, and the stringent requirements for measurement accuracy and speed of the instrumentation. Additionally, laboratory conditions often cannot adequately reproduce the real operating modes of wheeled tractors, construction machinery, or transport vehicles, especially under sudden load changes, complex road conditions, micro-transient processes, and the inertial response of the boost and control systems.

Therefore, instead of full-factorial experiments – which are extremely labor-intensive, time-consuming, and virtually unattainable for engines with wide ranges of load and rotational speeds – it is advisable to use standardized driving test cycles. For simulating the transport operations of wheeled tractors and other self-propelled machinery, the ELR (European Load Response) [22], EPA Nonregulatory Nonroad Duty Cycles [23], and NRTC (Non-Road Transient Cycle) [24] are widely employed. Each of these cycles is designed to replicate specific aspects of real engine operation, including variable load patterns, motion dynamics, transient regimes, and typical field and transport conditions.

4. Results

Before proceeding to the simulation of driving cycles for the wheeled tractor with an eCVT transmission, it should be noted that the tractor is equipped with an IVECO FPT NEF 67 (N67) ENTX20.00 internal combustion engine. For the operation of the eCVT, Bosch SMG230/132 electric machines were selected. Using numerical modeling, spatial graphs of the power units' external speed characteristics were constructed.

Fig. 5 shows the distribution of operating points for the changes in effective power of the internal combustion engine and the electric motor during the simulation of ELR, EPA, and NRTC cycles with the eCVT transmission. Fig. 5 is constructed according to the spatial principle [25, 26] for the purpose of greater clarity of the obtained results

Comparing the results of effective power in Fig. 5 (*a, b, c*), it can be noted that with the use of the proposed electromechanical transmission, the operating points shift toward lower angular speeds and reduced torque. This condition is beneficial in terms of fuel efficiency; however, under these conditions, the internal combustion engine does not operate at its optimal load, meaning it is underloaded. This, in turn, provides grounds for selecting a different internal combustion engine.

From Fig. 5 (*d, e, f*), it is observed that the electric motor's power does not reach peak levels, allowing it to operate within the constant torque region.

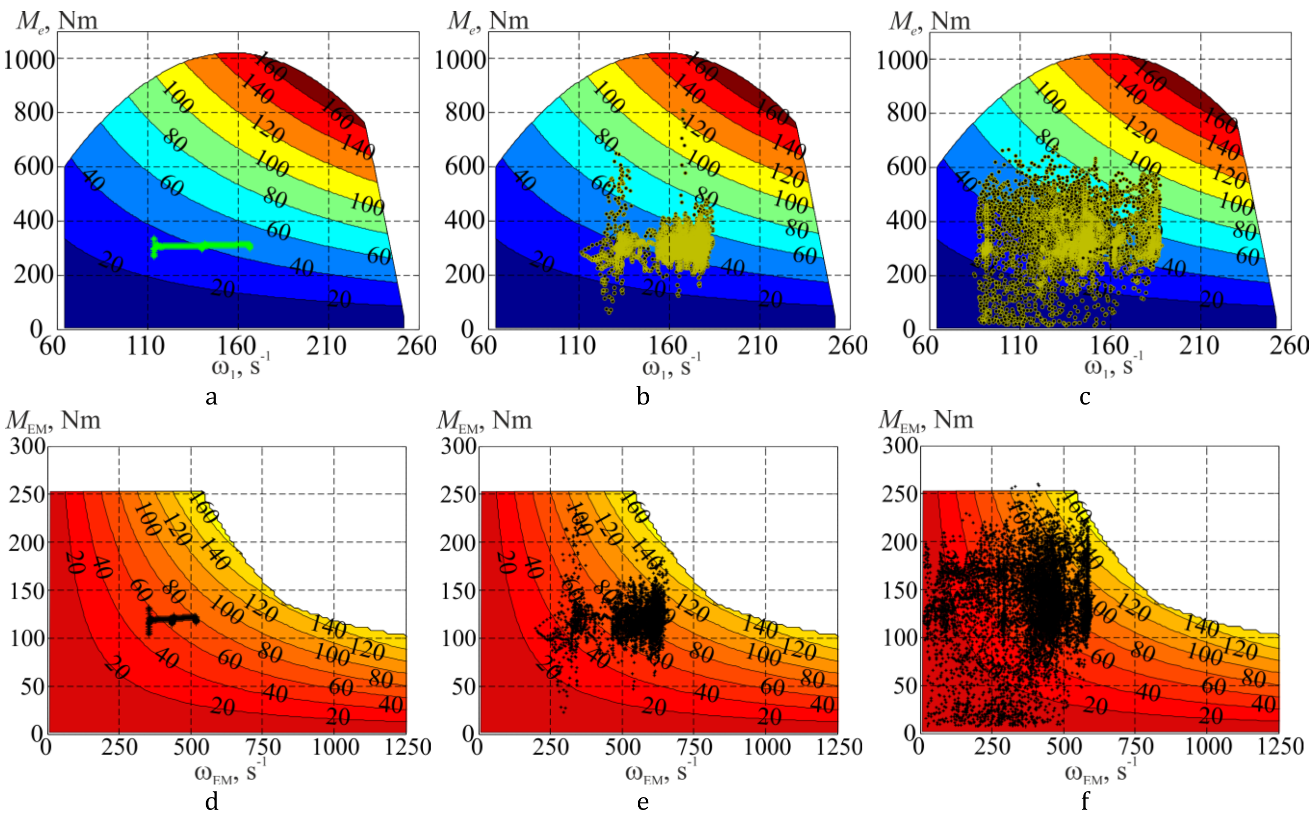

Figure 5: Change in the distribution of operating points of the effective power of the internal combustion engine (a, b, c) and the electric motor (d, e, f) during the simulation of the ELR (a, d), EPA (b, e), and NRTC (c, f) driving cycles

The instantaneous fuel consumption of the internal combustion engine is determined during the simulation of the ELR, EPA, and NRTC cycles with the eCVT transmission (Fig. 6).

5. Discussion

Analyzing the results in Fig. 6*a*, it is noted that during the simulation of the ELR cycle with a wheeled tractor equipped with an eCVT and a fully filled tank (*H* = 1.6 m), the instantaneous fuel consumption is G_M = 805 g. With an empty tank (*H* = 0 m), G_M = 280.9 g. This outcome is due to the eCVT's ability to implement variable transmission ratios and partially split the power flow.

For a tractor with a mechanical transmission, the maximum impact of fluid sloshing occurs at *H* = 1.3 m, with G_M = 1567 g, which is 4.72% higher than without considering fluid sloshing. For the eCVT tractor at *H* = 1.3 m, G_M = 770.1 g (Fig. 6*a*), 4.35% higher than the non-sloshing case (G_M = 736.6 g). The smaller relative increase with eCVT is due to torque redistribution between the engine and electric motor and smoother traction force generation.

From Fig. 6*b*, during the EPA cycle, the eCVT tractor with a full tank (*H* = 1.6 m) shows G_M = 1494 g, and with an empty tank (*H* = 0 m), G_M = 549 g. For the mechanical tractor at *H* = 1.3 m, G_M = 3068 g, 6.76% higher than without sloshing; for eCVT at *H* = 1.3 m, G_M = 1447 g (5.57% higher than G_M = 1370 g). The trend mirrors that of the ELR cycle.

For the NRTC cycle (Fig. 6*c*), the eCVT tractor with H = 1.6 m has GM = 2955 g, and with *H* = 0 m, G_M = 1214 g. For the mechanical tractor at *H* = 1.3 m, G_M = 6557 g, 9.22% higher than without sloshing; for eCVT at *H* = 1.3 m, G_M = 2962 g, 8.07% higher than G_M = 2723 g, consistent with ELR and EPA trends.

The significant reduction in fuel consumption with eCVT compared to mechanical transmission (approx. 50.6% for ELR, 52% for EPA, 54% for NRTC, both empty and full tanks) demonstrates more efficient matching of tractor traction characteristics. Relative differences between fill levels persist for both transmission types, confirming the dominant effect of transported mass on energy consumption, while eCVT significantly reduces the powertrain's sensitivity to load changes.

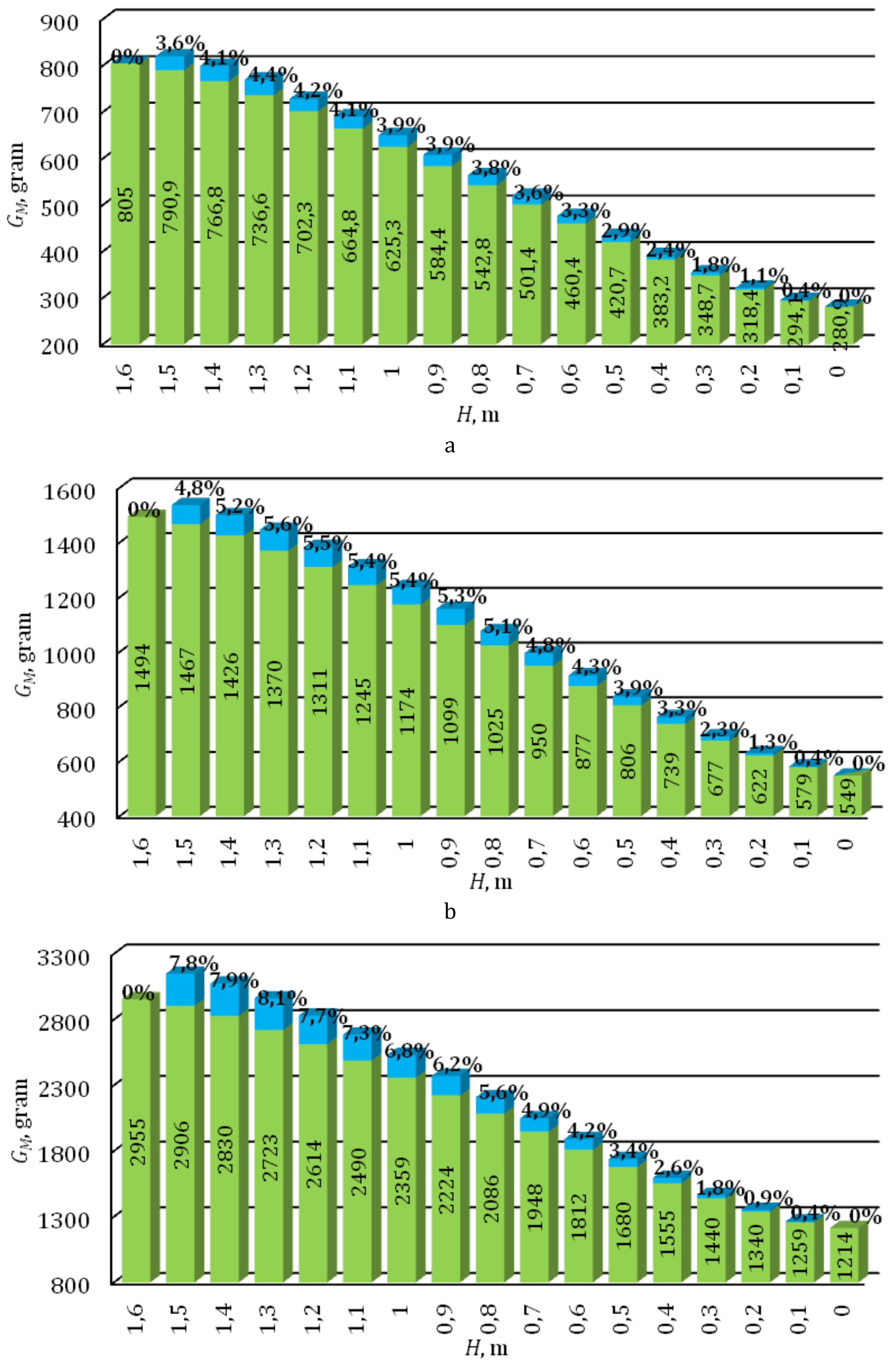

Figure 6: Change in instantaneous fuel consumption G_M during the simulation of the ELR (a), EPA (b), and NRTC (c) cycles with an eCVT, depending on the tank filling level HHH at a mass of 16 t

6. Conclusions

To evaluate the fuel efficiency of the machine-tractor unit, a validated mathematical model of the internal combustion engine crankshaft motion was used, based on the classical engine equilibrium equation. A simplified structural diagram of the eCVT transmission for the wheeled tractor is presented. A mathematical model of the wheeled tractor with a trailer tank during transport operations was proposed, taking into account traction forces and resistance forces. It was determined that, regardless of transmission type, the maximum increase in instantaneous fuel consumption during transport of a 16 t tank occurs at H = 1.3 m. For eCVT: ELR cycle GM = 770.1 g (relative increase 4.35%, absolute 33.5 g); EPA cycle GM = 1447 g (5.57%, 77 g); NRTC cycle GM = 2962 g (8.07%, 239 g). The study shows that instantaneous fuel consumption nearly halves when using eCVT compared to a standard mechanical transmission.

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