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THEORETICAL STUDY OF THE LONGITUDINAL DYNAMIC STABILITY OF AN AGGREGATE CONSISTING OF A FRUIT-HARVESTING MACHINE AND A TWO-AXLE TRAILER

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Annotatsiya. Maqolada intensiv bog‘lar uchun yaratilayotgan o‘ziyurar meva teruvchi mashina va konteyner tashiydigan ikki o‘qli tirkama platformasidan iborat agregatning bo‘ylama dinamik turg‘unligi nazariy tadqiq etilgan. Erkin jism usuli, Nyutonning ikkinchi qonuni va statik momentlar muvozanati asosida qiyalikda tortish va tormozlanish rejimlari uchun mashina hamda tirkama g‘ildiraklariga ta’sir etuvchi normal reaksiyalar, ilmoq kuchi va ular orqali bo‘ylama turg‘unlik mezonlari aniqlandi: bo‘ylama ag‘darilish va sirpanishning statik chegaraviy burchaklari, oldingi o‘qning yuklanganlik koeffitsienti bo‘yicha boshqariluvchanlik sharti va tormozlanishda ilmoqning siqilmaslik sharti. Tormoz momentining bo‘lakli-chiziqli o‘sishini va ilmoqning elastik-dissipativ xossalarini hisobga oluvchi ikki massali model yordamida o‘tkinchi jarayondagi ilmoq kuchi baholandi. Hisobiy parametrlarda tirkama bilan ko‘tarilishda boshqariluvchanlik saqlanadigan chegaraviy qiyalik 31,8° dan 21,0° gacha kamayishi, tirkama tormozining 0,4 s kechikishi ilmoqda 3,6 kN, 10 mm lyuft mavjud bo‘lganda esa 4,36 kN siquvchi kuch hosil qilishi (dinamiklik koeffitsienti 1,14–1,38) ko‘rsatildi. Tormoz kuchlarining moslik koeffitsienti $k_T \geq 1$ bo‘lishi va tirkama tormozi mashina tormozidan kechikmasligi agregat bo‘ylama dinamik turg‘unligining asosiy sharti ekanligi asoslandi.

Kalit so‘zlar: meva teruvchi mashina; ikki o‘qli tirkama; bo‘ylama turg‘unlik; dinamik turg‘unlik; ilmoq kuchi; normal reaksiya; tormozlanish; qiyalik; boshqariluvchanlik.

Аннотация. В статье теоретически исследована продольная динамическая устойчивость агрегата, состоящего из самоходной плодоуборочной машины для интенсивных садов и двухосной прицепной платформы для перевозки контейнеров. На основе метода свободного тела, второго закона Ньютона и уравнений равновесия моментов определены нормальные реакции на колёсах машины и прицепа, сила в сцепном устройстве для тягового и тормозного режимов на уклоне, а также критерии продольной устойчивости: предельные статические углы продольного опрокидывания и сползания, условие управляемости по коэффициенту нагрузки передней оси и условие отсутствия сжатия сцепки при торможении. С помощью двухмассовой модели, учитывающей кусочно-линейное нарастание тормозного момента и упруго-диссипативные свойства сцепки, оценена сила в сцепке в переходном процессе. Показано, что при расчётных параметрах прицеп снижает предельный угол подъёма по условию управляемости с 31,8° до 21,0°, а запаздывание тормозов прицепа на 0,4 с вызывает сжимающую силу 3,6 кН (4,36 кН при зазоре 10 мм; коэффициент динамичности 1,14–1,38). Обосновано, что основным условием продольной динамической устойчивости агрегата является коэффициент согласованности тормозных сил $k_T \geq 1$ при отсутствии запаздывания тормозов прицепа.

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Ключевые слова: плодоуборочная машина; двухосный прицеп; продольная устойчивость; динамическая устойчивость; сила в сцепке; нормальная реакция; торможение; уклон; управляемость.

Abstract. The paper presents a theoretical study of the longitudinal dynamic stability of an aggregate consisting of a self-propelled fruit-harvesting machine for intensive orchards and a two-axle container-carrying trailer platform. Using the free-body method, Newton's second law and moment-balance equations, the normal wheel reactions of the machine and trailer and the coupling force are derived for traction and braking on a slope, together with longitudinal stability criteria: the static limit angles of longitudinal overturning and sliding, a steerability condition based on the front-axle load coefficient, and a no-compression condition for the hitch during braking. A two-mass model accounting for the piecewise-linear rise of brake torque and the elastic-dissipative properties of the hitch is used to estimate the coupling force in the transient process. For the design parameters considered, the trailer reduces the steerability-limited climbing angle from 31.8° to 21.0°, and a 0.4 s trailer brake delay produces a compressive hitch force of 3.6 kN (4.36 kN with 10 mm free play; dynamic factor 1.14–1.38). It is shown that the main condition for longitudinal dynamic stability is a braking compatibility coefficient $k_T \geq 1$ with no delay of the trailer brakes relative to the machine brakes.

Keywords: fruit-harvesting machine; two-axle trailer; longitudinal stability; dynamic stability; coupling force; normal reaction; braking; slope; steerability.

The Republic has set the task of expanding the area of intensive orchards to 110 thousand hectares by 2030 [1]. In fruit-tree growing, harvesting is the most labour-intensive process, and the costs incurred on it amount to as much as 60 per cent of total costs [2]. In this connection, the Scientific Research Institute of Agricultural Mechanization, in cooperation with the Scientific and Practical Centre of the National Academy of Sciences of Belarus for Agricultural Mechanization, has proposed the design of a self-propelled fruit-harvesting machine for intensive orchards and of its two-axle container-carrying trailer platform [3]. The machine has a 4K2 wheel arrangement in which the rear driving wheels are powered by an electric motor; its turning properties and the traction dynamics of the aggregate have been studied in earlier works [3, 4].

During operation the fruit-harvesting aggregate moves over uneven, sloping surfaces between orchard rows and, in the transport mode, along field roads. The positioning of the fruit pickers on the machine platform and the stacking of containers in two tiers on the trailer raise the centres of mass considerably. As a result, during acceleration, braking and climbing or descending a slope, the redistribution of load between the axles is intensified, and there arises a risk of unloading of the front (steered) wheels, locking of the braked rear wheels and "jackknifing" of the units of the aggregate relative to one another.

The classical theoretical foundations of the braking dynamics and stability of wheeled machines and road trains are set out in the works of A.S. Litvinov and Ya.Ye. Farobin [5], and for tractors in those of V.V. Guskov and co-authors [6]. J. Ahokas and S. Kosonen studied experimentally the dynamic behaviour of a tractor–trailer combination during braking and showed that unbalanced operation of the brakes of the units reduces the stability of the combination [7]; H. Aykan and co-authors confirmed the relevance of this problem under various operating conditions [8]. R.Ye. Zheleznov [9] and B.V. Kisulenko [10] solved problems of improving the motion stability of a road train by selecting the parameters of the coupling device and the masses of the units. The standard values of braking distance are specified in the OST 37.001.016-70 standard [11].

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In works on the stability of self-propelled mobile vehicles, the main attention has been paid to lateral static stability: according to GOST 28523-90 the angle of lateral static stability must be no less than 20° [12], the transport speed of vehicles without sufficient lateral stability is limited to 17–18 km/h [13], and the lateral static stability indicators of row-crop tractors with various wheel arrangements have been studied by B. Qambarov and B. Xoliqov [14]. The use of machine-tractor aggregates and the greatest slope angle that a tractor train can negotiate are considered in the works of M. Toshboltayev [15, 16].

However, longitudinal dynamic stability criteria that jointly take into account the traction and braking modes, the slope, the imbalance and delay of the brakes of the units, and the elasticity and free play of the hitch have not been sufficiently developed specifically for an aggregate consisting of a fruit-harvesting machine and a two-axle trailer. The **aim** of this paper is to develop analytical dependences determining the longitudinal dynamic stability indicators of an aggregate consisting of a fruit-harvesting machine and a two-axle trailer and, on their basis, to substantiate the parameters of the aggregate and the requirements for the braking system. The **objectives** of the study are: 1) to determine the normal reactions of the wheels and the coupling force in the traction and braking modes on a slope; 2) to find the static limit angles of longitudinal overturning and sliding; 3) to determine the steerability limit in terms of the front-axle load; 4) to estimate the steady-state and transient values of the coupling force during braking.

2.1. Design scheme and assumptions adopted

The object of the study is an aggregate consisting of a self-propelled fruit-harvesting machine with a 4K2 wheel arrangement and a two-axle container-carrying trailer platform connected to it by a hinged drawbar (Fig. 1). The machine and the trailer were separated by the free-body method, and the longitudinal coupling force P_{il} acting at their connection point was introduced as an external force [5, 6]. The coupling force was taken as positive in tension (the trailer pulls the machine backwards) and negative in compression (the trailer pushes the machine forwards).

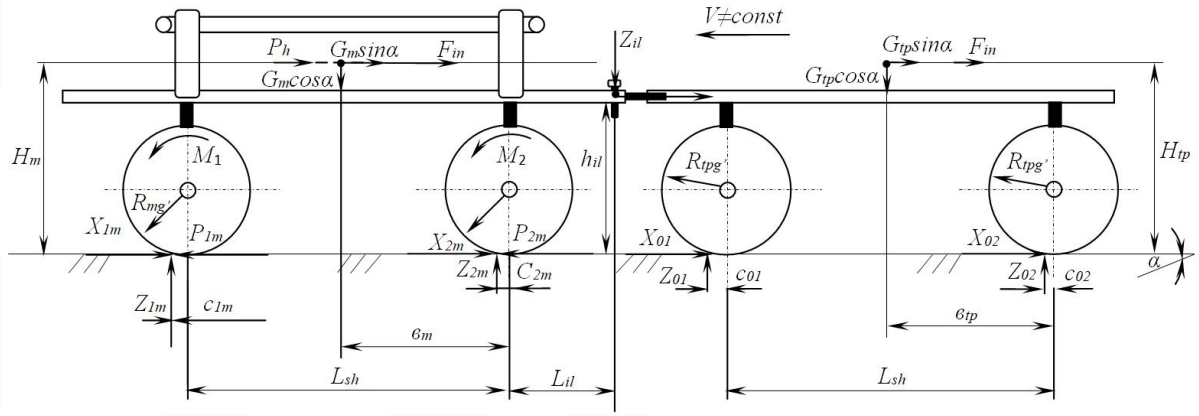


Fig. 1. Diagram of the forces acting on the aggregate consisting of a fruit-harvesting machine and a two-axle trailer platform

The following assumptions were adopted: the aggregate moves in a straight line, and the supporting surface is a plane slope with angle α (α > 0 when climbing); the machine and the trailer are treated as rigid bodies; since the drawbar of the two-axle trailer is hinged at both ends, the vertical force on the hitch $Z_{il} \approx 0$; since the travel speed does not exceed 5 m/s, air resistance is neglected (for the aggregate, according to the expression $P_h = 3.5(1 + 0.25n)V^2$ [3], it is less than 110 N); since braking forces account for 96–98 per cent of the decelerating forces during braking [5, 6], rolling resistance is neglected; the adhesion coefficient φ between the wheel and the supporting surface is constant.

Longitudinal dynamic stability was assessed by the following four criteria: (i) the static limit angle of longitudinal overturning α_{ag}; (ii) the limit angle at which the braked machine slides down the

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slope α_φ (sliding is required to occur before overturning: $\alpha_\varphi < \alpha_{ag}$); (iii) the load coefficient of the front steered axle $\lambda_1 \geq [\lambda_1] = 0.2$ [6]; (iv) no compression of the hitch during braking $P_{il} \geq 0$ [7, 10].

2.2. Normal reactions of the wheels

The normal reactions of the machine axles are found from the equation of moments of the forces acting on the machine about the contact point of the rear wheel and from the sum of the vertical projections (Fig. 1):

$$Z_{1m}=\frac{G_m(b_m\cos\alpha-H_m\sin\alpha)-m_mWH_m-P_{il}h_{il}}{L_{sh}} \quad (1)$$

$$Z_{2m}=G_m\cos\alpha-Z_{1m} \quad (2)$$

where $G_m = m_m g$ is the weight of the machine; L_{sh} is the wheelbase of the machine; b_m is the distance from the centre of mass to the rear axle; $a_m = L_{sh} - b_m$; H_m is the height of the centre of mass; h_{il} is the hitch height; W is the acceleration of the aggregate (during braking $W = -j$). Similarly, for the trailer:

$$Z_{01}=\frac{G_{tp}(b_{tp}\cos\alpha-H_{tp}\sin\alpha)-m_{tp}WH_{tp}+P_{il}h_{il}}{L_{tp}} \quad (3)$$

$$Z_{02}=G_{tp}\cos\alpha-Z_{01} \quad (4)$$

where G_{tp}, m_{tp} are the weight and mass of the loaded trailer; L_{tp} is the wheelbase of the trailer; b_{tp} is the distance from the centre of mass to the rear axle of the trailer; H_{tp} is the height of the trailer's centre of mass. Expressions (1)–(4) are a special case, with $Z_{il} \approx 0$, of the equilibrium equations obtained in [3], and they are common to the traction ($W \geq 0$) and braking ($W < 0$) modes.

2.3. Coupling force

In the traction mode, the trailer's resistance to motion determines the coupling force:

$$P_{il}=G_{tp}(f\cos\alpha+\sin\alpha)+m_{tp}W \quad (5)$$

where f is the rolling resistance coefficient. In the braking mode, the equations of motion of the machine and the trailer take the form

$$m_m j=F_m+P_{il}+G_m\sin\alpha, \quad m_{tp}j=F_{tp}-P_{il}+G_{tp}\sin\alpha \quad (6)$$

(F_m, F_{tp} being the total braking forces at the wheels of the machine and the trailer); eliminating the common deceleration j from them gives the coupling force [5]:

$$P_{il}=\mu(\delta_{tp}-\delta_m)=\mu\delta_m(k_T-1), \quad \mu=\frac{m_m m_{tp}}{m_m+m_{tp}} \quad (7)$$

where $\delta_m = F_m/m_m$, $\delta_{tp} = F_{tp}/m_{tp}$ are the partial specific decelerations of the machine and the trailer; $k_T = \delta_{tp}/\delta_m$ is the braking-force compatibility coefficient; μ is the reduced mass of the aggregate. It follows from (7) that, since the weight components due to the slope act on both units in proportion to their masses, they do not directly affect the coupling force; the slope manifests itself only through the reactions (1)–(4). When n identical trailers are coupled to the aggregate, the force in the i-th hitch is given by

$$P_{il,i}=\frac{(n-i+1)m_{tp}m_m}{m_m+nm_{tp}}(\delta_{tp}-\delta_m), \quad i=1,\ldots,n \quad (8)$$

and the greatest force arises between the machine and the first trailer.

2.4. Transient process model

In the initial stage of braking, the braking systems of the units come into action at different times. If the hitch is treated as an element with stiffness c and damping coefficient k, i.e. as an elastic-dissipative element, the following equation is obtained from (6) for its deformation $u = x_m - x_{tp}$:

$$\mu\ddot{u}+k\dot{u}+cu=\mu[\delta_{tp}(t)-\delta_m(t)], \quad P_{il}=cu+k\dot{u} \quad (9)$$

When free play Δ is present, within the range $|u| \leq \Delta/2$ it is assumed that $P_{il} = 0$. The partial decelerations of the units are expressed by a piecewise-linear approximation of the rise of brake torque with time:

$$\delta_i(t)=0 \ (t<t_{\delta i}); \quad \delta_i(t)=\delta_{i\,max}\frac{t-t_{\delta i}}{t_{0i}} \ (t_{\delta i}\leq t\leq t_{\delta i}+t_{0i}); \quad \delta_i(t)=\delta_{i\,max} \ (t>t_{\delta i}+t_{0i}) \quad (10)$$

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where $t_{\delta i}$ is the actuation (response) time of the braking system of the i-th unit, and t_{0i} is the duration of the brake torque rise. Equations (9)–(10) were integrated numerically by the Euler method with a step of 10^{-4} s. The effect of the transient process was assessed by the dynamic factor $K_d = |P_{il}|_{max}/(\mu\delta_{m\,max})$.

2.5. Design parameters

The calculations were carried out for design parameters adopted on the basis of the structural layout given in [3, 4] (Table 1). The trailer mass is the sum of four full containers (≈ 400 kg each) and the structural mass of the platform.

Table 1. Design parameters of the aggregate

Parameter	Symbol	Value
Operating mass of the machine	m_m	2600 kg
Wheelbase of the machine	L_{sh}	2.50 m
Distance from the centre of mass to the rear axle	b_m	1.15 m
Height of the machine's centre of mass	H_m	1.05 m
Hitch height	h_{il}	0.55 m
Mass of the loaded trailer	m_{tp}	2400 kg
Wheelbase of the trailer / b_{tp}	L_{tp} / b_{tp}	2.60 / 1.30 m
Height of the trailer's centre of mass	H_{tp}	1.20 m
Rolling resistance coefficient (orchard surface)	f	0.10
Adhesion coefficient	φ	0.5…0.7 (baseline 0.6)
Braking system times (machine)	$t_{\delta m}$ / t_{0m}	0.2 / 0.3 s
Rise time of the trailer brake	t_{0tp}	0.4 s
Hitch stiffness / relative damping	c / ζ	$4\cdot10^5$ N/m / 0.15

3.1. Static longitudinal stability

Setting W = 0, P_{il} = 0 and Z_{1m} = 0 (or Z_{2m} = 0) in (1)–(4) gives the static limit angles of the machine overturning backwards when climbing and forwards when descending:

$$tg\alpha_{ag}^{\uparrow}=\frac{b_m}{H_m},\qquad tg\alpha_{ag}^{\downarrow}=\frac{a_m}{H_m},\qquad tg\alpha_{ag}^{tp}=\frac{b_{tp}}{H_{tp}} \tag{11}$$

In a machine with the 4K2 arrangement, brake mechanisms are fitted only to the rear wheels. From the condition that the braked machine does not slide on the slope, $\varphi Z_{2m} \geq G_m \sin\alpha$, taking (2) into account, the limit sliding angles are obtained:

$$tg\alpha_{\varphi}^{\downarrow}=\frac{\varphi a_m}{L_{sh}+\varphi H_m},\qquad tg\alpha_{\varphi}^{\uparrow}=\frac{\varphi a_m}{L_{sh}-\varphi H_m} \tag{12}$$

where "↓" denotes the machine facing downhill and "↑" the machine facing uphill. The calculation results are given in Table 2.

Table 2. Limit angles of longitudinal static stability, deg

Indicator	φ = 0.5	φ = 0.6	φ = 0.7
Backward overturning of the machine, $\alpha_{ag}^{\uparrow}$	47.6	47.6	47.6
Forward overturning of the machine, $\alpha_{ag}^{\downarrow}$	52.1	52.1	52.1

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Indicator	φ = 0.5	φ = 0.6	φ = 0.7
Overturning of the trailer, α_{ag}^{tp}	47.3	47.3	47.3
Sliding (facing downhill), $\alpha_{\varphi}^{\downarrow}$	12.6	14.5	16.3
Sliding (facing uphill), $\alpha_{\varphi}^{\uparrow}$	18.9	23.4	28.2

In all cases the condition $\alpha_{\varphi} < \alpha_{ag}$ is satisfied, i.e. in the static state the limiting condition is governed by sliding rather than overturning. The holding ability of a machine facing downhill with only its rear wheels braked is limited to 14.5° at φ = 0.6.

3.2. Steerability in the traction mode

Substituting (5) into (1) gives an expression for the load coefficient of the front axle $\lambda_1 = Z_{1m}/(G_m\cos\alpha)$. In steady motion (W = 0), the condition $\lambda_1 = [\lambda_1]$ yields, in closed form, the limit climbing angle at which steerability is retained:

$$\operatorname{tg}\alpha_{\lambda} = \frac{b_m - [\lambda_1]L_{sh} - f h_{il} m_{tp}/m_m}{H_m + h_{il} m_{tp}/m_m} \qquad (13)$$

It follows from (13) that the ratio of the trailer mass to the machine mass and the hitch height are the main factors reducing the limit angle. The dependence of λ_1 on slope and acceleration is shown in Fig. 2.

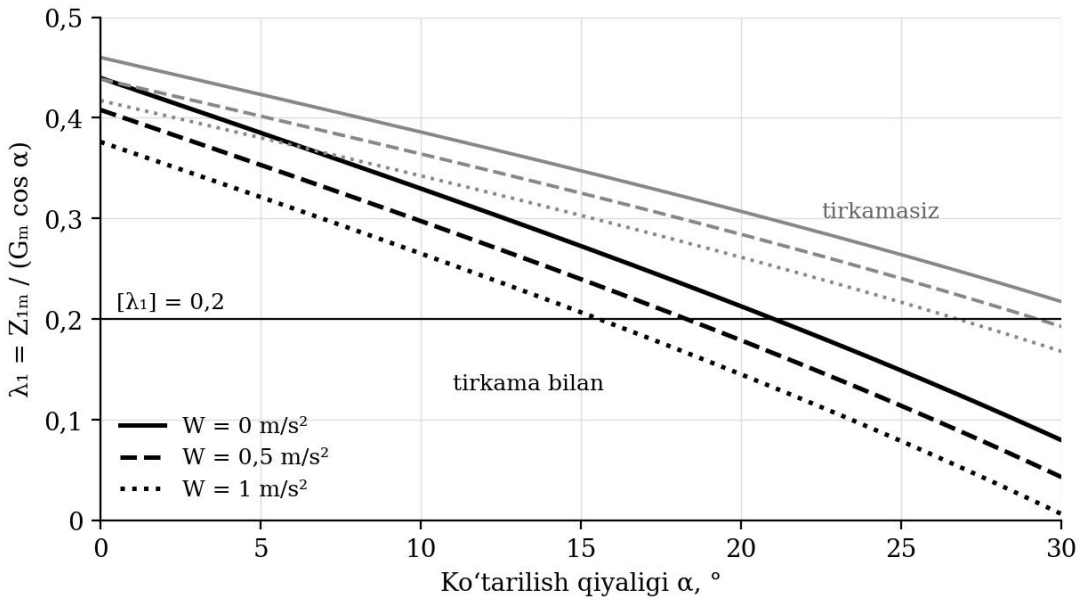


Fig. 2. Dependence of the load coefficient of the machine's front axle on the climbing slope (grey – without trailer, black – with trailer)

For the machine without a trailer, the condition $[\lambda_1] = 0.2$ is satisfied up to $\alpha \le 31.8°$, whereas with the trailer coupled the limit slope decreases to 21.0°. When moving on a slope with acceleration (W = 0.5 and 1.0 m/s²), the limit angle is 18.3° and 15.6° respectively with the trailer, and 29.3° and 26.8° without it. Thus, the effect of the trailer reduces the limit slope by 10–11°, and this effect persists as acceleration increases.

3.3. Steady-state mode during braking

For a machine with only its rear wheels braked, the condition $F_m \le \varphi Z_{2m}$, taking (1), (2) and (7) into account, gives the maximum partial deceleration attainable by the machine on a horizontal road:

$$\delta_{m\,max} = \frac{\varphi g a_m}{L_{sh} + \varphi\left[H_m \frac{m_m + k_T m_{tp}}{m_m + m_{tp}} - h_{il}\frac{m_{tp}(k_T - 1)}{m_m + m_{tp}}\right]}, \qquad j = \delta_{m\,max}\frac{m_m + k_T m_{tp}}{m_m + m_{tp}} \qquad (14)$$

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The braking distance of the aggregate was estimated by a simplified expression that accounts for the piecewise-linear rise of brake torque:

$$S_T = \frac{V_0^2}{2j} + V_0\left(t_{\delta m} + \frac{t_{0m}}{2}\right) \quad (15)$$

and compared with the standard value according to OST 37.001.016-70, $[S_T] = 0.18V_0 + V_0^2/143$ [11]. The calculation results for $V_0 = 20$ km/h, $\varphi = 0.6$ are given in Table 3.

Table 3. Effect of the braking-force compatibility coefficient on the performance of the aggregate ($V_0 = 20$ km/h, $\varphi = 0.6$)

k_T	$\delta_{m\ max}$, m/s²	j, m/s²	P_{il}, kN	Z_{2m}, kN	Z_{02}, kN	S_T, m
0	2.66	1.38	–3.32	11.53	10.94	13.1
0.5	2.60	1.97	–1.62	11.26	9.93	9.8
0.8	2.56	2.32	–0.64	11.10	9.34	8.6
1.0	2.54	2.54	0	11.00	8.96	8.0
1.5	2.48	3.08	+1.55	10.75	8.04	7.0
2.0	2.43	3.59	+3.03	10.52	7.15	6.2
1.0[1]	5.89	5.89	0	—	—	4.6

Note: [1] – the case in which all wheels of the machine are braked; $[S_T] = 6.4$ m.

The table shows that when $k_T < 1$ the hitch is compressed: when the trailer brake does not operate ($k_T = 0$) the compressive force reaches 3.32 kN, while the deceleration falls to 1.38 m/s². As k_T increases, the hitch is put in tension and the deceleration of the aggregate rises, but the rear axle of the trailer is unloaded (Z_{02} decreases from 10.94 kN to 7.15 kN). For a machine with only its rear wheels braked, the standard braking distance $[S_T] = 6.4$ m is achieved only at $k_T \geq 1.8$; in this case the partial deceleration of the trailer is 4.4 m/s², which is below the limit $\varphi g = 5.9$ m/s².

3.4. Transient process of braking

The effect on the coupling force of the delay of the trailer brake relative to the machine brake, $\Delta t = t_{\delta tp} - t_{\delta m}$, was studied using model (9)–(10) for $k_T = 1$ (Fig. 3, Table 4). For the design parameters, the natural frequency of the hitch is $\omega_0 = (c/\mu)^{1/2} = 17.9$ rad/s (2.85 Hz), and the reduced mass is $\mu = 1248$ kg.

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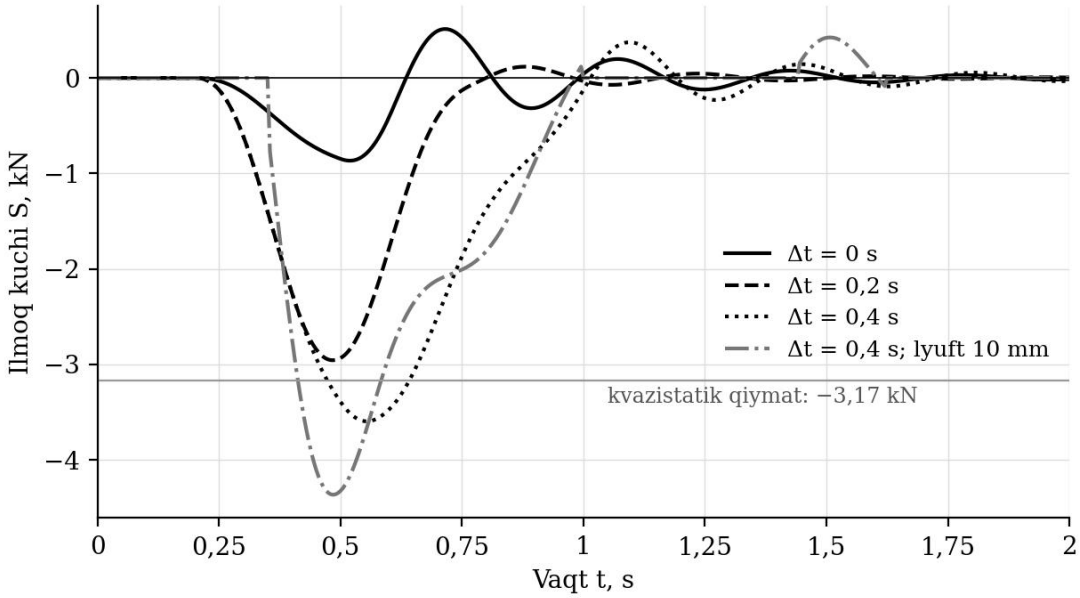


Fig. 3. Variation of the coupling force in the transient process of braking ($k_T = 1$, $\zeta = 0.15$)

Table 4. Maximum compressive force in the hitch and dynamic factor in the transient process

Trailer brake delay Δt, s	**0**	**0.2**	**0.4**	**0.6**
$\|P_{il}\|_{max}$, kN ($\Delta = 0$)	0.87	2.96	3.60	3.60
K_d ($\Delta = 0$)	0.27	0.93	1.14	1.14
$\|P_{il}\|_{max}$, kN (Δ = 10 mm)	1.85	4.12	4.36	4.36
K_d (Δ = 10 mm)	0.58	1.30	1.38	1.38

When the delay $\Delta t \geq t_{0m}$, a period arises in which the machine is fully braked while the trailer is not yet braked, and the coupling force exceeds the quasi-static value $\mu\delta_{m\,max} = 3.17$ kN by 14 %. With 10 mm free play, the force rises to 4.36 kN because the units collide after taking up the gap ($K_d = 1.38$). Even without a delay, the difference in the rise times of the brake torques (0.3 and 0.4 s) produces a compressive impulse of 0.87 kN. Varying the damping coefficient from $\zeta = 0.05$ to 0.30 reduces the maximum force only from 3.63 to 3.57 kN, i.e. the main factors are the synchronism of the brakes and the magnitude of the free play.

According to (1), the compressive force $|P_{il}| = 3.6$ kN transfers a load of $P_{il}h_{il}/L_{sh} \approx 0.79$ kN from the rear axle of the machine to the front axle, which reduces the normal reaction of the braked rear wheels by ≈ 7 % and increases the likelihood of their locking.

The results obtained show that the longitudinal stability of the fruit-harvesting aggregate is governed by constraints of two different kinds. The static overturning angles (47–52°) are far greater than the actual slopes found in orchard conditions and are of no practical significance; the real constraints are determined by the sliding (12.6–16.3°) and steerability (15.6–21.0° with the trailer) criteria. This conclusion agrees with the results obtained for tractor trains in [6] and [15, 16]: when a trailer is coupled, the slope that the aggregate can negotiate is limited not by overturning but by adhesion and by unloading of the front axle. Expression (13), in turn, makes it possible to assess this limit for the fruit-harvesting aggregate directly at the design stage.

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During braking, the coupling force given by expression (7) is proportional to the difference between the partial decelerations of the units, and when $k_T < 1$ the hitch is compressed. This is a quantitative description of the situation recorded in the experiments of J. Ahokas and S. Kosonen [7] – destabilisation of the combination when the trailer brakes are insufficient. As shown by B.V. Kisulenko [10] for road trains, a compressed hitch makes the units prone to "jackknifing"; in the fruit-harvesting aggregate this risk is further increased, because in the 4K2 machine only the rear wheels are braked and compression unloads precisely those wheels. For this reason, the condition $k_T \geq 1$ should be adopted as a necessary condition for longitudinal dynamic stability.

On the other hand, increasing k_T excessively unloads the rear axle of the trailer (Table 3) and may cause the trailer to swing sideways if its rear wheels lock first. It is therefore advisable to distribute the braking forces between the trailer axles close to the ratio Z_{01}/G_{tp} (0.62–0.68 for the design parameters). The fact that a machine with only its rear wheels braked requires $k_T \approx 1.8$ to achieve the standard braking distance indicates the need to consider fitting the front wheels of the machine with brake mechanisms as well: in a machine with all wheels braked, even at $k_T = 1$, $S_T = 4.6$ m $< [S_T] = 6.4$ m.

The transient-process results confirm, for the fruit-harvesting aggregate, the need to select the parameters of the coupling device (stiffness, free play) substantiated in the work of R.Ye. Zheleznov [9]: a free play of 10 mm increases the dynamic force by 21 %. In calculating the strength of the hitch, it is recommended to apply to the quasi-static force (7) a dynamic factor of at least $K_d = 1.4$.

Limitations of the study: the model is built for straight-line motion and a plane slope; tyre elasticity, wheel slip, the variability of the mass of the fruit pickers on the platform and road irregularities are not taken into account; the parameters are design values and need to be refined for the experimental prototype. In further research it is planned to verify the dependences obtained in field tests and to develop a model combined with the lateral dynamic stability of the aggregate when turning.

1. Dependences (1)–(8) were obtained that determine the normal reactions of the wheels of the fruit-harvesting machine and the two-axle trailer and the coupling force for the traction and braking modes on a slope; they make it possible to assess four criteria of the longitudinal dynamic stability of the aggregate – overturning, sliding, steerability and no compression of the hitch.

2. For the design parameters, the static limit angles of longitudinal overturning are 47–52°, far greater than the sliding angles (12.6–28.2°); hence the static longitudinal stability of the aggregate is limited by the adhesion condition.

3. Coupling the trailer reduces the limit climbing angle under the steerability condition ($\lambda_1 \geq 0.2$) from 31.8° to 21.0°, and to 15.6° at an acceleration of 1 m/s^2; this limit is determined by the closed-form expression (13).

4. The condition for longitudinal dynamic stability during braking is a braking-force compatibility coefficient $k_T \geq 1$; for a machine with only its rear wheels braked, $k_T \approx 1.8$ is required to achieve the standard braking distance according to OST 37.001.016-70, which shows that it is advisable to brake the front wheels of the machine as well.

5. A 0.4 s delay of the trailer brake produces a compressive force in the hitch of 3.6 kN, and of 4.36 kN with 10 mm free play ($K_d = 1.14$–1.38). It is recommended that the trailer brake should not lag behind the machine brake, that the free play of the hitch be minimal, and that the strength of the hitch be calculated with $K_d \geq 1.4$.

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