UDC 621.8

Modeling the movement of a machine at an angle for transporting construction materials

Sergey Oryschenko [1], *Viktor Oryschenko* [2].

[1,2]Kyiv National University of Construction and Architecture,
31, Air Force Ave., Kyiv, Ukraine, 03037,
[1] oryschenko.sv@knuba.edu.ua, https://orcid.org/0000-0002-5359-5285 ,
[2] oryscshenko.vv@knuba.edu.ua, https://orcid.org/0000-0002-5081-1229

Received: 29.10.2024; Accepted: 29.11.2024
https://doi.org/10.32347/gbdmm.2024.104.0302

Abstract. During the working process of moving a machine for transporting building materials, forces act on it. The calculation of the longitudinal stability of a machine for transporting building materials is carried out from the condition of tipping forward, taking into account the fact that pneumatic tires are deformed if the machine is equipped with rubber wheels. The angle of additional forward tilt of the machine due to deformation of the supports is determined by the ratio of the gravity of the machine with the load, the stiffness of the soil under the front and rear rubber wheels. The distance between the center of gravity of the machine and the vertical axis passing through the tipping point is therefore taken into account when calculating the longitudinal stability of the machine. The machine has the smallest margin of longitudinal stability when moving down a slope with simultaneous braking of the machine and working equipment when lowering it.

Keywords: angle of inclination, mass of the car, resistance, gravitational force, inertial force, speed of movement, labor productivity, traction.

INTRODUCTION

The basics of vehicle theory include determining the traction capabilities of vehicles, as well as all possible resistances that arise during the operation of these vehicles. Let us dwell in more detail on the determination of resistances that arise during the operation of vehicles when overcoming a slope angle**.**

PURPOSE OF THE PAPER

The movement of the machine at an upward angle was studied, the machine model and the forces that need to be overcome were simulated. [10]. The studies confirmed the correctness of the adopted design solutions and calculations of the main parameters.

RESEARCH RESULTS

The first stage is determining the gravitational force from a loaded and empty vehicle, kN:

$$F_1 = (m_T + m_в) \cdot g \; ; \qquad (1)$$

$$F_2 = m_T \cdot g \; , \qquad (2)$$

Where:
m_T- vehicle weight, t; $m_в$- weight of the cargo being transported, t;
g- free fall speed, m/c^2

The second stage is the determination of all possible resistances during the operation of vehicles. Resistance to movement of a loaded and empty vehicle, kN:

$$W_{ПЕР\ ГР} = F_1 \cdot f \;\; ; \qquad (3)$$

$$W_{ПЕР\ ПОР} = F_2 \cdot f \; ; \qquad (4)$$

where:
f – coefficient of resistance to movement

The resistance to turning of wheeled vehicles moving on a solid base is usually very small and is not taken into account in the calculations. [1]. When driving on loose soil, kN:

$$W_{ПОВ} = (0{,}25 \ldots 0{,}5) \cdot W_{ПЕР}; \qquad (5)$$

Resistance to movement of a loaded and empty vehicle from the slope of the terrain, kN:

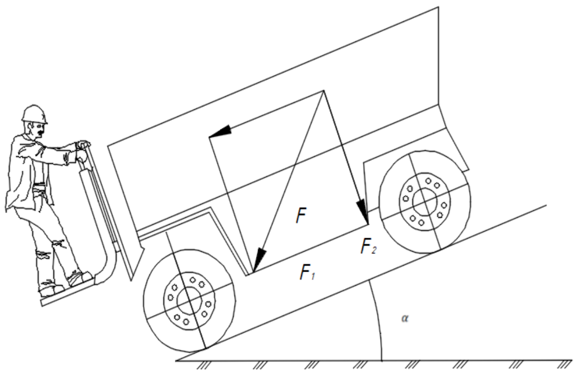

Fig. 1. Diagram of forces acting on the vehicle when tilted α:

$$W_{\text{УКЛ ГР}} = \pm F_1 \cdot \sin(\alpha)\ ; \qquad (6)$$

$$W_{\text{УКЛ ПОР}} = \mp F_2 \cdot \sin(\alpha)\ , \qquad (7)$$

Where:
α - The angle of elevation of the path, degrees. The sign (+) corresponds to the movement of the machine uphill, and (-) downhill.

Resistance to inertia forces during acceleration (+) and braking (-) for uniformly accelerated (uniformly decelerated) motion, kN:

$$W_{\text{ИН ГР}} = \pm \frac{(M_T + M_{\text{ГР}}) \cdot V_{\text{ГР}}}{t_{\text{РАЗГ(гал)}}}\ ; \qquad (8)$$

$$W_{\text{ИН ПОР}} = \pm \frac{M_T \cdot V_{\text{ПОР}}}{t_{\text{РАЗГ(гал)}}}\ ; \qquad (9)$$

Where:$V_{\text{ГР}}$($V_{\text{ПОР}}$) speed of the loaded (empty) car at the end of acceleration (beginning of braking) (m/s); $t_{\text{РАЗГ(гал)}}$ - duration of acceleration (braking) of the car, p.

The third stage is the determination of conditions for ensuring the movement of the vehicle. [1-2].

The movement of the car is possible if the maximum traction forces on the drive wheels (sprockets) of the vehicle for each gear T_{MAX} not less than the total resistance to movement$\sum W$:

$$T_{MAX} \geq \sum W\ ; \qquad (10)$$

Effort T_{MAX} limited by two factors: the drive power of the running equipment and the conditions of the coupling of the engine with the base:

$$\sum W \leq T_i = \frac{P \cdot \eta_{TP}}{V_{\text{ПЕР}}} \leq P_{\text{СЦ}i},\ ; \ (11)$$

Where: P - power of the engines of the movement mechanism, kW;
η_{TP}- Transmission efficiency of the movement mechanism;
$V_{\text{ПЕР}}$ - machine speed, m/s.

Traction force from the clutch of a loaded and empty vehicle, kN:

$$P_{\text{СЦ ГР}} = F_1 \cdot K\ ; \qquad (12)$$

$$P_{\text{СЦ ПОР}} = F_2 \cdot K\ ; \qquad (13)$$

Where K - vehicle's traction coefficient.Resistance to movement of a loaded and empty vehicle taking into account the track inclination angle - α, kN:

$$W_{\text{ПЕР ГР}}(\alpha) = F_1 \cdot (f_T \cdot \cos(\alpha) + \sin(\alpha))\ (14)$$

$$W_{\text{ПЕР ПОР}}(\alpha) = F_2 \cdot (f_T \cdot \cos(\alpha) + \sin(\alpha))(15)$$

Traction force from the clutch of a loaded and empty vehicle taking into account the track angle, kN:

$$P_{\text{СЦ ГР}}(\alpha) = F_1 \cdot \varphi \cdot \cos(\alpha); \qquad (16)$$

$$P_{\text{СЦ ПОР}}(\alpha) = F_2 \cdot \varphi \cdot \cos(\alpha). \qquad (17)$$

Next, we will consider the algorithm for determining the speed of the car at maximum elevation, m/s. [3]. The algorithm for calculating the speed includes determining the following vehicle parameters: Gravity forces from the car- G_T and loaded - $G_{\text{н}}$, kN:

$$G_T = M_T \cdot g\ ; \qquad (18)$$

$$G_{\text{н}} = M_{\text{н}} \cdot g\ ; \qquad (19)$$

Where M_T, M_{3T} - respectively, the mass of the machine and the load,т;

g - acceleration of free fall g =9,81$м/c^2$.

Tractor travel resistance W_T, cargo $W_н$ and the entire vehicle $W_{ПЕР}$ without taking into account the angle of inclination of the track, kN:

$$W_T = G_T \cdot f_T \; ; \qquad (20)$$

$$W_{3T} = G_{3T} \cdot f_{3T} \; ; \qquad (21)$$

$$W_{ПЕР} = W_Т + W_{3T} \; ; \qquad (22)$$

Where f_T, f_{3T} - respectively, the resistance of the machine and the load.

Traction force of the machine with clutch, kN:

$$P_{СЦ} = G_T \cdot \varphi \; ; \qquad (23)$$

Where φ – coefficient. Maximum speed of movement of the machine with clutch, m/s:

$$V_{MAX} = \frac{3{,}6 \cdot N \cdot \eta_{TP}}{P_{СЦ} \cdot (1-\delta)} \; ; \qquad (24)$$

Where: P - tractor engine power, kW;

η_{TP} - transmission efficiency;

δ - skid coefficient of the machine.

Resistance to movement for the machine and the load, taking into account the angle of inclination of the road, kN:

$$W_{3T}(\alpha) = G_{3T} \cdot (f_{3T} \cdot \cos(\alpha) + \sin(\alpha)) \; ; \qquad (25)$$

$$W_{ПЕР}(\alpha) = G_T \cdot (f_T \cdot \cos(\alpha) + \sin(\alpha)) + G_{3T} \cdot (f_{3T} \cdot \cos(\alpha) + \sin(\alpha)) = \cos(\alpha) \cdot (G_T \cdot f_T + G_{3T} + f_{3T}) + \sin(\alpha) \cdot (G_T + G_{3T}) \; ; \qquad (26)$$

Where α - road slope angle;

Traction force on the coupling from the reference angle of the track slope, kN:

$$P_{СЦ}(\alpha) = M_T \cdot g \cdot \varphi \cdot \cos(\alpha) \; ; \qquad (27)$$

Maximum track slope angle (degrees) from the condition of traction of the engine with the road[9]:

$$W_{ПЕР}(\alpha) = P_{СЦ}(\alpha) \; ; \qquad (28)$$

$$\cos(\alpha) \cdot (G_T \cdot f_T + G_{3T} \cdot f_{3T}) + \sin(\alpha) \cdot (G_T + G_{3T}) = M_T \cdot g \cdot \varphi \cdot \cos(\alpha) \; ; \qquad (29)$$

Let's divide both the left and right parts of this expression. [4]. We get the maximum permissible lifting angle of the machine:

$$tg(\alpha) = \frac{M_T \cdot g \cdot \varphi - (G_T \cdot f_T + G_{3T} \cdot f_{3T})}{G_T + G_{3T}} \; ; \qquad (30)$$

$$\alpha max = \frac{\alpha\tan(tg(\alpha)) \cdot 180}{\pi} \; ; \qquad (31)$$

Movement resistance for the machine taking into account the load at the lifting angle, kN:

$$W_{MAX} = W_{ПЕР}\left(\frac{\alpha max \cdot \pi}{180}\right) \; ; \qquad (32)$$

Machine travel speed at maximum lift, m/s:

$$V_{max} = \frac{3{,}6 \cdot N \cdot \eta_{TP}}{W_{max} \cdot (1-\delta)} \; ; \qquad (33)$$

Let us now consider the algorithm for determining vehicle performance[4-5]. It includes the determination of the following parameters: Vehicle cycle duration:

$$T_Ц = t_П + t_{ТГ} + t_{ТХ} + t_Р \; ; \qquad (34)$$

Where: $t_П$ - loading time, min;

$t_{ТГ}$ - loaded stroke time, min;

$t_{ТХ}$ - idle time, min;

$t_Р$ - unloading time, min.

Number of loads to load the vehicle:

$$n_{ПОГР} = g_{ТР} \cdot \frac{K_Г}{(g_{ПОГР} \cdot K_{ГР})} ; \qquad (35)$$

Where $g_{ПОГР}$ - load capacity of the truck,т; $K_{ГР}$ - coefficient of utilization of the load capacity of the freight vehicle[7-8].Vehicle loading time;

ISSN(online)2709-6149. Mining, constructional, road and melioration machines, 104, 2024, 38-42

$$t_{П} = t'_{П} + t_1 = t_{Ц} \cdot n_{ПОГР} + t_1 \; ; \qquad (36)$$

Where: $t'_{П}$ - vehicle loading time, min.;
t_1 - Loading point downtime due to vehicle change, min.;
$t_{Ц}$- truck operating cycle time, хв.
Time to load the vehicle, min:

$$t_{П} = \frac{t_{Ц} \cdot g \cdot K_{Г}}{(g_{ПОГР} \cdot K_{ГП}) + t_1} \; ; \qquad (37)$$

Time of loaded vehicle travel, min:

$$t_{ТГ} = \frac{60 \cdot l_{ТР}}{V_{ТГ}} \; ; \qquad (38)$$

Where: $l_{ТР}$ - distance of the transported cargo, km;
$V_{ТГ}$ - cargo transportation speed, km/h.
Time vehicle idling speed, min;

$$t_{ТХ} = \frac{60 \cdot l_{ТХ}}{V_{ТХ}} \; ; \; (39)$$

Where $l_{ТХ}$ - idling distance, km;
$V_{ТХ}$ - idle speed, km/h.
Gravity of a loaded and empty vehicle, kN.[6] :

$$G_{ТГ} = (M_T + M_{ТР}) \cdot g \; ; \qquad (40)$$

$$G_{ТХ} = (M_T \cdot g) \; ; \qquad (41)$$

Where: M_T - Vehicle weight, т;
$M_{ТР}$ - Weight of the cargo being transported, ton;
g - acceleration of free fall, m/s.
Resistance to movement of the loaded $W_{ТГ}$ and empty $W_{ТХ}$, vehicle, kN:

$$W_{ТГ} = G_{ТГ} \cdot f_T \; ; \qquad (42)$$

$$W_{ТХ} = G_{ТХ} \cdot f_T \; , \qquad (43)$$

Where f_T - Coefficient of resistance to movement;
Traction force with loaded clutch$P_{СЦГ}$ and empty$P_{СЦХ}$ vehicle speed, km/h:

$$P_{СЦГ} = (M_T + M_{ТР}) \cdot g \cdot \varphi \; ; \qquad (44)$$

$$P_{СЦХ} = (M_T) \cdot g \cdot \varphi \; ; \qquad (45)$$

Where φ - Coefficient of adhesion of the vehicle.The speed of movement of the loaded
$V_{ТГ}$ and empty $V_{ТХ}$ vehicle speed, km/h:

$$V_{ТГ} = \frac{3{,}6 \cdot N \cdot \eta_{TP}}{\min\{W_{ТГ}, P_{СЦГ} \cdot (1-\delta)\}} = \frac{3{,}6 \cdot N \cdot \eta_{TP}}{\min\{G_{ТГ} \cdot f_T, G_{ТГ} \cdot \varphi (1-\delta)\}} \; ; \; (46)$$

$$V_{ТХ} = \frac{3{,}6 \cdot N \cdot \eta_{TP}}{\min\{W_{ТХ}, P_{СЦХ} \cdot (1-\delta)\}} = \frac{3{,}6 \cdot N \cdot \eta_{TP}}{\min\{G_{ТХ} \cdot f_T, G_{ТХ} \cdot \varphi (1-\delta)\}} \; ; \; (47)$$

Where:
P - vehicle engine power, kW;
η_{TP} - Vehicle transmission efficiency;
δ - Slip coefficient.

CONCLUSIONS

The movement of the machine at an angle is investigated, the interaction of the wheel with the surface is also considered. The forces acting on the machine are considered. A method for calculating the main parameters of the machine for transporting building materials, as well as its ability to move with a load at an angle, is proposed.

REFERENCES

1. **Daniel Eriksson, Reza Ghabcheloo, Marcus Geimer.** (2024). Optimizing bucket-filling strategies for wheel loaders inside a dream environment Automation in Construction Volume 168, Part A, 1 December 2024, 105804 (https://doi.org/10.1016/j.autcon.2024.105804).
2. **Torstensson P. T., Pieringer A., Höjer M. , Nilsson R. , Simonsson V.** (2024). A case study of railway curve squeal radiated from both the outer and inner wheel Received 25 April 2024; Received in revised form 18 September 2024; Accepted 28 September 2024 (https://doi.org/10.1016/j.apacoust.2024.110327).
3. **Torstensson P. T., Nielsen J. C. O**. (2011). Simulation of dynamic vehicle–track interaction on small radius curves. Veh Syst Dyn 2011;49(11):1711–

32.(https://doi.org/10.1080/00423114.2010.499468).
4. **Sayyaadi H., Kouhi H., Salarieh H.** (2011). Control of car-like (wheeled) multi robots for following and hunting a moving target Scientia Iranica Volume 18, Issue 4, August 2011, 950-965. (https://doi.org/10.1016/j.scient.2011.07.006).
5. **Stasys Steišūnas, Ján Dižo, Gintautas Bureika, Vidas Žuraulis** (2017). Examination of Vertical Dynamics of Passenger Car with Wheel Flat Considering Suspension Parameters Procedia Engineering Volume 187, 2017, 235-241. https://doi.org/10.1016/j.proeng.2017.04.370.
6. **Muresan B., Cesbron J., Lumiere L., Gary V., Truong X.T., Kane M.** (2024). A study of the relationship between rear-of-wheel particle emissions and close-proximity tire/road noise of a passenger car Science of The Total Environment Volume 918, 25 March 2024, 170578https://doi.org/10.1016/j.scitotenv.2024.170578.
7. **Brucker Juricic B., Galic M., Marenjak S.**, (2021). Review of the construction labour demand and shortages in the EU, Buildings 11 (1) (2021) 17, http://dx.doi.org/10.3390/buildings11010017.
8. **Thompson D.** (2009). Chapter 9 - Curve Squeal Noise. In: Thompson D, editor. Railway Noise and Vibration. Oxford: Elsevier; 2009. p. 315–42. https://doi.org/10.1016/B978-0-08-045147-3.00009-8.
9. **Panko J., Kreider M., Unice K**. (2018). Chapter 7 - Review of Tire Wear Emissions: A Review of Tire Emission Measurement Studies: Identification of Gaps and Future Needs. Non-Exhaust Emissions, Academic Press, Eds. Fulvio Amato, 147–160. https://doi.org/10.1016/B978-0-12-811770-5.00007-8.
10. **Filla R., Frank B.** (2017). Towards finding the optimal bucket filling strategy through simulation, in: Proceedings of 15:Th Scandinavian International Conference on Fluid Power, 15th Scandinavian International Conference on Fluid Power, Fluid Power in the Digital Age, SICFP'17, June 7–9 2017 - Linköping, Sweden, in: Linköping Electronic Conference Proceedings, Linköping University Electronic Press, 2017, 402–417, http://dx.doi.org/10.3384/ecp17144402

Моделювання руху машини під кутом для перевезення будівельних матеріалів

Сергій Орищенко[1], *Віктор Орищенко*[2],

[1,2]*Київський національний університет будівництва і архітектури*

Анотація. Під час робочого процесу навантажувач перемішується на майже горизонтальних майданчиках, допустимий ухил яких. Розрахунок поздовжньої стійкості навантажувачів ведеться з умови перекидання вперед з урахуванням того, що деформуються пневматичні шини, якщо пневмоколісний хід. Кут додаткового нахилу навантажувача вперед внаслідок деформації опор визначається співвідношенням сили тяжкості навантажувача з вантажем жорсткість ґрунту під переднім та заднім котками гусеничного ходу або радіальна жорсткість передніх та задніх пневматичних шин навантажувача на пневмоколісному ході; відстань між центром ваги навантажувача та вертикальною віссю, що проходить через точку перекидання. Тому при розрахунку поздовжньої стійкості гусеничного та пневмоколісного навантажувачів. Найменший запас поздовжньої стійкості має навантажувач у разі руху під ухил з одночасним гальмуванням машини та робочого обладнання при його опусканні. Положення робочого обладнання відповідає максимальному вильоту.

Ключові слова: кут нахилу, маса авто, опір, сила тяжіння, сила інерції, швидкість руху, продуктивність праці, зчеплення.