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MATHEMATICAL MODELING OF THE KINEMATIC AND DYNAMIC PARAMETERS OF THE RUNNING GEAR OF A THREE-WHEELED MINI-TRACTOR

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Abstract:

Compact three-wheeled mini-tractors — comprising a single steered front wheel and two driven rear wheels — are increasingly used on small private (dekhkan) farms, in orchards and in greenhouse rows because of their small turning radius and high manoeuvrability in confined plots. Systematic, self-contained mathematical descriptions of the coupled kinematics and dynamics of this specific running-gear layout are, however, less commonly available in an accessible form than models developed for four-wheel or articulated tractor-implement systems. This paper develops a combined kinematic–dynamic model of the running gear of a three-wheeled mini-tractor. The kinematic sub-model, based on the classical tricycle (Ackermann-equivalent) formulation, relates the front-wheel steering angle to the turning radius, the yaw rate of the machine and the individual angular velocities of the two rear wheels. The dynamic sub-model, derived from Newton's second law, expresses the longitudinal equation of motion in terms of the adhesion-limited tractive force, the rolling-resistance force and the draft resistance of a towed implement, and is complemented by a lateral-stability (rollover) criterion based on the static stability factor and a critical cornering speed. Because no physical prototype or experimental data set was available at the time of the study, every numerical input is explicitly declared as a design assumption, calibrated against published data for comparable mini-tractor classes. The model was evaluated numerically for a front-wheel steering-angle range of 5–35° and for three draft-resistance scenarios (unloaded travel, light cultivation and mouldboard ploughing). The results show the turning radius decreasing from about 16.0 m to 2.0 m while the critical rollover speed falls from about 43 km/h to 15 km/h over the same steering range, revealing a direct manoeuvrability–stability trade-off inherent to the three-wheeled layout; the longitudinal-dynamics simulation shows the time needed to reach 90% of the steady travel speed increasing from 2.7 s (unloaded) to 9.0 s (ploughing) as the draft resistance approaches the adhesion limit of the driven wheels. On this basis, practical recommendations are proposed for the choice of track width, centre-of-gravity height and maximum permissible steering angle in the design of three-wheeled mini-tractor running gear.

Keywords: three-wheeled mini-tractor; running gear; kinematic model; dynamic model; Newton's second law; turning radius; rollover stability; tractive force; rolling resistance.

Introduction

The mechanization of small private and dekhkan farms is one of the priority directions of agrarian reform in Uzbekistan and in Central Asia more broadly, where the average size of individual farm plots is considerably smaller than in large mechanized agricultural enterprises. On such plots — as well as in orchards, vineyards, greenhouses and vegetable rows — the manoeuvrability of a power unit is often as important as its traction capacity, because the machine must repeatedly turn at row ends (headlands) within a limited working width. Three-wheeled mini-tractors, in which a single front wheel provides steering while two rear wheels supply traction, are one of the layouts developed specifically to meet this requirement: because only one wheel needs to be turned and the wheelbase can be kept short, this configuration typically achieves a substantially smaller minimum turning radius than an equivalent four-wheeled machine with two steered wheels[1-3].

The turning ability of wheeled tractors and machine-tractor aggregates has received considerable theoretical attention. Bulgakov et al. built an equivalent kinematic scheme for a wheeled tractor together with a trailed implement and derived analytical expressions for the turning radius and the coordinates of the turning centre as functions of the steering angle. Kambarov et al. analysed design measures for minimizing the turning radius of 4K2 and 4K4 wheeled tractors with reference to the relevant state standard for the determination of minimum turning radius, GOST 30752–2001. Maitra et al. investigated an increase of the steering-wheel deflection angle from 35° to 55° through a modified four-bar steering linkage in order to further reduce the turning radius of a tractor assembly. Nalobina et al. systematized the design and operating parameters of mini-tractors of different structural weight classes, while Holtkamp reviewed the role of small, low-power wheeled tractors in the mechanization of smallholder agriculture in developing regions[4].

A separate body of work addresses the dynamics of tractors coupled to towed or mounted implements. Nosrati et al. simplified a six-wheel tractor-and-implement system to an equivalent three-wheel dynamic model in order to describe the lateral and draft forces transmitted through the hitch, and several authors have developed kinematic and dynamic models of articulated tractor–trailer systems for automatic guidance and path-tracking applications. In parallel, the mobile-robotics literature provides a compact and rigorous kinematic description of wheeled platforms with one steered wheel — the so-called tricycle and Ackermann models — which is directly transferable to the running gear considered here. The dynamic side of wheel–soil interaction, rolling resistance and traction on deformable terrain is treated in depth in the classical terramechanics and vehicle-dynamics texts of Wong , and Gillespie, including the static-stability and rollover criteria used below; the underlying rigid-body mechanics follow standard engineering-dynamics treatments such as, and the general robot kinematics/dynamics formalism follows. Complementary work on the traction devices and driving wheels of conventional agricultural tractors confirms the practical importance of the adhesion-limited traction force used in Section II[5].

Despite this extensive literature, a compact derivation that combines, within a single consistent notation, (i) the tricycle kinematics of a three-wheeled mini-tractor, (ii) the longitudinal dynamics of its driven rear axle under variable implement draft, and (iii) an explicit lateral rollover-stability criterion referred to the same geometric parameters, is not commonly presented for this specific vehicle class. The purpose of the present paper is therefore to develop such a model, to evaluate it numerically for a representative set of design parameters, and to draw practical conclusions for the choice of the main running-gear dimensions — wheelbase, track width and centre-of-gravity height — of a three-wheeled mini-tractor[6].

Materials and Methods

A. Object of Study and Design Scheme

The object of the study is the running gear of a three-wheeled mini-tractor with a single, free-steered front wheel and two mechanically driven rear wheels mounted on a common rigid axle (no differential is assumed, so that the running gear is treated, for the purpose of the kinematic model, as an ideal rolling-without-slipping tricycle). The reference point of the machine is taken at the midpoint of the rear axle; the vehicle pose is described by the planar coordinates (x, y) of this point and by the heading angle θ

measured from the x-axis, while δ denotes the steering angle of the front wheel measured from the longitudinal axis of the machine and v is the forward speed of the reference point[7].

Because no physical prototype of the mini-tractor and no experimental measurement campaign were available at the time of this study, the numerical values of the design and operating parameters used below (Table I) are explicitly adopted as design assumptions. They were chosen, in each case, to be representative of the structural-weight class of mini-tractors described in the systematization of Nalobina et al. and of typical off-road wheel–soil interaction coefficients reported by Wong; wherever the exact value would in practice require experimental identification on a physical unit, this is stated explicitly rather than presented as a measured result[8].

Table 1. Baseline design and operating parameters of the three-wheeled mini-tractor (all values adopted as design assumptions unless otherwise noted)

Symbol	Parameter	Value	Unit
m	Total operating mass (tractor + operator)	650	kg
L	Wheelbase (front axle – rear axle)	1.40	m
B	Rear track width	1.00	m
r	Rolling radius of rear (driven) wheels	0.30	m
h_m	Height of centre of gravity above ground	0.55	m
a	Distance from CG to rear axle	0.60	m
b	Distance from CG to front axle	0.80	m
δmax	Maximum front-wheel steering angle	35	deg
f	Rolling-resistance coefficient (tilled loam soil)	0.08	–
φ	Wheel–soil adhesion (traction) coefficient	0.60	–
Pe	Assumed rated drawbar power	9.0	kW
vmax	Transmission/governor-limited top travel speed	25 (6.94)	km/h (m/s)

The static loads on the front wheel, Gf, and on the combined rear axle, Gr, follow from the longitudinal position of the centre of gravity:

$$G_f = \frac{a}{L} m g , \qquad G_r = \frac{b}{L} m g$$

B. Kinematic Model of the Running Gear

Under the standard no-slip, planar-motion assumption used for low-speed wheeled platforms, the tricycle running gear obeys the kinematic equations

$$\dot{x} = v\cos\theta, \qquad \dot{y} = v\sin\theta, \qquad \dot{\theta} = \frac{v}{L}\tan\delta$$

$$R = \frac{L}{\tan\delta} , \qquad \omega_w = \frac{v}{r}$$

where L is the wheelbase, R is the instantaneous turning radius referred to the midpoint of the rear axle, and ωw is the angular velocity of a rear wheel of rolling radius r rotating without slip. During a steady turn at radius R and yaw rate θ̇ = v/R, the two rear wheels — separated by the track width B — must roll at different linear speeds:

$$v_{in} = \dot{\theta}\left(R - \frac{B}{2}\right), \qquad v_{out} = \dot{\theta}\left(R + \frac{B}{2}\right)$$

with the inner wheel (vin) tracing a smaller radius, R − B/2, and the outer wheel (vout) a larger radius, R + B/2. Equations (1)–(3) allow the required angular velocity of each rear wheel, ωin = vin/r and ωout = vout/r, to be computed for any combination of forward speed and steering angle, and are used directly to generate the numerical results of Section III-A[9].

C. Longitudinal Dynamic Model

The longitudinal motion of the mini-tractor along its path is governed by Newton's second law applied to the machine treated as a single rigid body of mass m:

$$m\frac{dv}{dt} = F_k - F_f - F_p - F_w$$

$$F_f = f m g \cos\alpha, \qquad F_k^{\max} = \varphi G_r$$

where Fk is the tractive (driving) force generated at the rear wheels, Ff is the rolling-resistance force, Fp is the draft resistance of a towed or mounted implement (equal to zero for unloaded travel), and Fw is the aerodynamic drag, which is negligible at the low working speeds of a mini-tractor and is dropped from the numerical evaluation below. The rolling-resistance force is proportional to the total weight component normal to the ground, with rolling-resistance coefficient f, and the tractive force that can actually be transmitted to the soil is bounded by the adhesion (traction) limit at the driven rear wheels, expressed through the adhesion coefficient φ and the static rear-axle load Gr. Because the engine–transmission system delivers a limited rated power Pe, the tractive force available at a given forward speed is additionally constrained by Fk(v) ≤ Pe/v, so that at low speed the motion is adhesion-limited (Fk = Fk max) while at higher speed it becomes power-limited (Fk = Pe/v); the smaller of the two values is used at every instant in the numerical integration of (4).

D. Lateral Dynamics and Rollover-Stability Criterion

When the mini-tractor negotiates a turn of radius R at forward speed v, its centre of gravity experiences a centripetal (lateral) acceleration ay = v²/R. Following the classical rollover analysis of wheeled vehicles [5], the outer wheels experience an increased normal load and the inner wheels a reduced normal load as ay grows; the theoretical threshold at which the inner-wheel load falls to zero and rollover about the outer wheel track becomes imminent defines the static stability factor, SSF = (B/2)/h_m, and the corresponding critical (rollover) speed for a given turning radius R:

$$a_y = \frac{v^2}{R}, \qquad SSF = \frac{B/2}{h_g}$$

$$v_{cr} = \sqrt{\frac{g R B}{2 h_g}}$$

Equation (6) is used below to evaluate, for each steering angle (and hence each turning radius R computed from (2)), the forward speed above which the three-wheeled mini-tractor would theoretically be at risk of lateral rollover on a level, firm surface. Because a three-wheeled layout has no lateral wheel on the side opposite the single front wheel, the relevant track width for the SSF calculation is that of the driven rear axle, B, which is normally the narrower of the two axle track widths and therefore governs the stability margin of the whole machine.

E. Numerical Implementation

Equations (1)–(6) were implemented and evaluated numerically using the parameters of Table I. Two sets of numerical experiments were performed. First, the cornering kinematics and the lateral-stability limit (Section III-A) were evaluated for front-wheel steering angles δ ∈ {5°, 10°, …, 35°} at a fixed reference working speed v = 1.8 m/s. Second, the longitudinal equation of motion (4) was integrated forward in time with an explicit first-order (Euler) scheme, time step Δt = 0.01 s, from a standing start (v = 0) for three draft-resistance scenarios representative of typical field operations: unloaded transport (Fp = 0 N), light secondary cultivation (Fp = 1000 N) and mouldboard ploughing (Fp = 1500 N) (Section III-B). All computations were carried out in Python 3 (NumPy); no experimental data were used to calibrate or validate the numerical results, which should accordingly be interpreted as a theoretical evaluation of the proposed model rather than as a validated performance prediction for a physical machine[10].

Results

A. Cornering Kinematics and Lateral Stability

Table II summarizes the turning radius, yaw rate, lateral acceleration and critical rollover speed computed from (2), (3) and (6) for the seven steering angles considered, at the reference working speed v = 1.8 m/s. As expected from (2), the turning radius decreases monotonically and non-linearly with increasing steering angle, from R = 16.0 m at δ = 5° to R = 2.0 m at δ = 35°. Over the same range the lateral acceleration at the fixed reference speed increases from 0.20 m/s² to 1.62 m/s², while the critical rollover speed — which depends on R and therefore falls as the machine is steered more sharply — decreases from about 43.0 km/h to 15.2 km/h. Because a working speed of 1.8 m/s (6.5 km/h) used for Table II is well below every computed critical speed, the machine is not predicted to be at rollover risk under the assumed baseline geometry at this speed for any steering angle up to 35°; the margin, however, narrows sharply as δ increases, from a factor of about 6.6 at δ = 5° to a factor of only about 2.3 at δ = 35°[11].

Table 2. Computed cornering kinematics and lateral-stability limit versus front-wheel steering angle (v = 1.8 m/s, b = 1.00 m, h_m = 0.55 m)

δ (deg)	R (m)	θ̇ (rad/s)	ay (m/s²)	vcr (km/h)
5	16.00	0.113	0.20	43.0
10	7.94	0.227	0.41	30.3
15	5.23	0.345	0.62	24.6
20	3.85	0.468	0.84	21.1
25	3.00	0.600	1.08	18.6
30	2.43	0.742	1.34	16.7
35	2.00	0.900	1.62	15.2

Fig 1. Turning radius and lateral-stability limit versus steering angle (B = 1.0 m, h_g = 0.55 m)

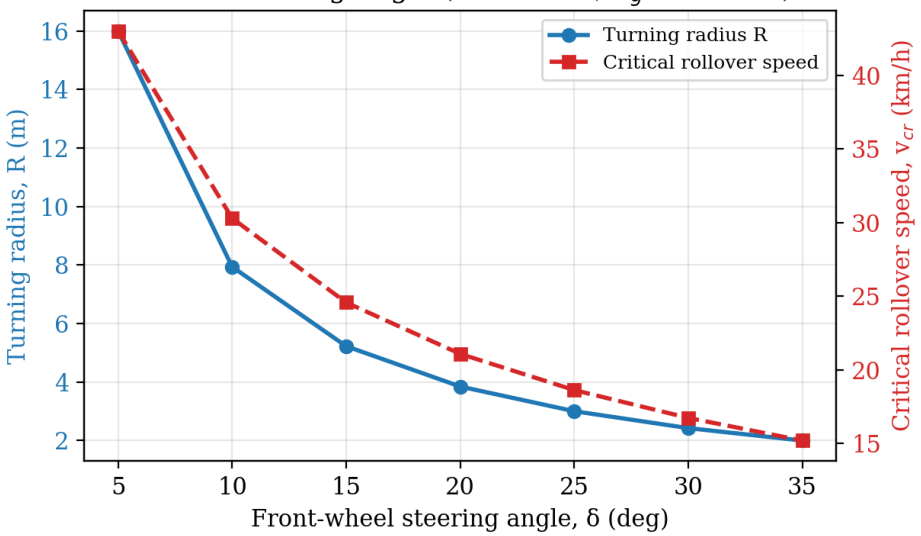


Figure 1. Turning radius and critical rollover speed as functions of the front-wheel steering angle.

At δ = 35° the individual rear-wheel speeds computed from (3) are vin = 1.35 m/s (inner wheel) and vout = 2.25 m/s (outer wheel) for a common vehicle speed of v = 1.8 m/s — a difference of 67% — which confirms that, in the absence of a differential, sharp turning of the three-wheeled layout necessarily involves considerable slip at one or both rear wheels unless a differential or an overrunning clutch mechanism is provided; this is discussed further in Section IV.

B. Longitudinal Dynamics Under Variable Draft Resistance

With the baseline parameters of Table I, the static rear-axle load is Gr = 3644 N (57.1% of the total weight, reflecting the assumed forward position of the engine and the rearward bias of the load-bearing

structure typical of tricycle mini-tractors), giving an adhesion-limited maximum tractive force Fk max = φ·Gr ≈ 2186 N and a rolling-resistance force Ff = f·m·g ≈ 510 N. The corresponding maximum instantaneous acceleration available from a standing start, before the power limitation becomes active, is amax = (Fk max − Ff)/m ≈ 2.58 m/s²[12].

Fig. 1 shows the forward-speed response v(t) obtained by integrating (4) for the three draft-resistance scenarios defined in Section II-E. For unloaded travel the machine reaches the assumed transmission-limited top speed of 25 km/h (6.94 m/s) after about 3.2 s, with 90% of this steady speed reached at t90 ≈ 2.7 s. Under the light-cultivation draft load (Fp = 1000 N) the achievable steady speed falls to about 21.0 km/h (t90 ≈ 5.9 s), because the traction available above the power-limited threshold speed is no longer sufficient to reach the transmission-limited maximum. Under the mouldboard-ploughing draft load (Fp = 1500 N) — which brings the combined resistance Ff + Fp to within about 3% of the adhesion-limited tractive force Fk max — the steady-state working speed falls sharply to about 9.8 km/h (2.71 m/s), with a markedly longer transient (t90 ≈ 9.0 s), illustrating that as the draft load approaches the adhesion limit the running gear becomes increasingly traction-limited rather than power-limited, and small further increases in draft resistance would be expected to produce disproportionately large reductions in attainable working speed, and eventually wheel spin or stalling once Ff + Fp exceeds Fk max[13].

Fig 2. Simulated forward-speed response for three draft-resistance scenarios (m = 650 kg)

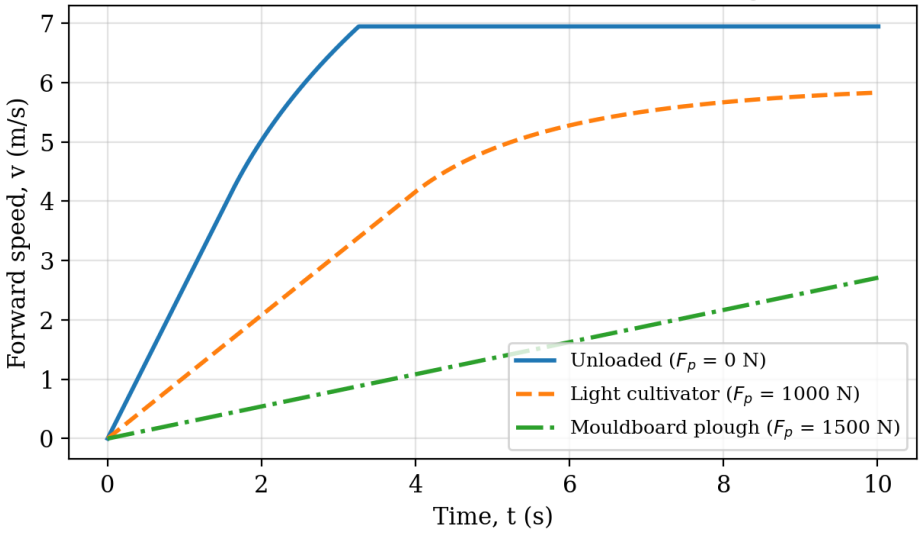


Figure 2. Simulated forward-speed response of the mini-tractor for three draft-resistance scenarios.

Table 3. Simulated longitudinal-dynamics results for three draft-resistance scenarios

Scenario	Fp (N)	Ff+Fp (N)	Steady v (km/h)	t90 (s)
Unloaded transport	0	510	25.0	2.7
Light cultivation	1000	1510	21.0	5.9
Mouldboard ploughing	1500	2010	9.8	9.0

Discussion

The numerical results of Section III quantify a trade-off that is qualitatively well known in the tractor-design literature but is here expressed within a single, explicit model: reducing the turning radius of a three-wheeled mini-tractor by increasing the front-wheel steering angle — as pursued, for instance, through improved steering linkages in or through optimized 4K2/4K4 steering geometry in— simultaneously reduces the theoretical critical rollover speed for a given track width and centre-of-gravity height, because a smaller turning radius increases the lateral acceleration experienced at any given forward speed. For the assumed baseline geometry (B = 1.00 m, h_m = 0.55 m, SSF ≈ 0.91), the critical speed at the tightest modelled steering angle (35°, R = 2.0 m) is about 15 km/h, comfortably above the low working speeds typical of tillage operations but low enough that an operator turning at the headland

at transport speed, or on a cross-slope, could approach the stability limit. This is consistent with the general treatment of rollover in [5] and underlines that, for a three-wheeled layout in particular — which lacks the additional lateral wheel base of a four-wheel machine on the side of the steered wheel — the rear track width B and the centre-of-gravity height h_m are the two design parameters with the most direct influence on lateral stability, and should be reviewed jointly with any decision to increase the maximum steering angle beyond the 35° assumed here.

The absence of a differential between the two rear wheels, assumed for simplicity in the kinematic model of Section II-B, is shown by the results of Section III-A to become significant at sharp steering angles: a 67% speed difference between the inner and outer rear wheels at $\delta = 35°$ implies that a solid (non-differential) rear axle would force one wheel to slip during any sharp turn, which would increase rolling resistance locally, accelerate tyre wear, and reduce the effective tractive force available for a following implement, in line with the wheel–soil slip effects documented in the terramechanics literature [3], [4], [14]. This suggests that, for three-wheeled mini-tractors intended for frequent headland turning in narrow rows, a low-cost overrunning clutch or a limited-slip differential on the rear axle — rather than the fully rigid axle assumed for the baseline model — would be a worthwhile design refinement[14].

The longitudinal-dynamics results of Section III-B illustrate the practical consequence of the adhesion limit at the driven rear wheels: because the assumed adhesion coefficient ($\varphi = 0.60$) and rear-axle load together bound the maximum tractive force to about 2186 N, a draft resistance approaching this value (as for the assumed mouldboard-plough scenario) leaves only a narrow margin above the rolling-resistance force, so that the steady working speed becomes highly sensitive to further increases in draft resistance or to any reduction in adhesion — for example on wet or loosely tilled soil, where φ would be expected to fall below the 0.60 assumed here. This is consistent with the general finding in the tractor-implement dynamics literature that draft-load variation is a first-order influence on achievable field speed, and indicates that, for heavier tillage implements, increasing the rear-axle static load (through ballasting or a more rearward centre-of-gravity position) or increasing tyre-soil adhesion would be more effective design responses than increasing rated engine power alone, since the latter is only exploitable once the machine is already travelling above the adhesion-limited speed threshold identified in Section II-C.

Several simplifications limit the scope of the present model and should be addressed in follow-up work. First, the kinematic model assumes pure rolling without slip; a more complete description would couple (1)–(3) with a wheel-slip and side-slip model of the kind used in terramechanics analyses of rigid wheels on deformable terrain. Second, the rolling-resistance and adhesion coefficients (f, φ) were adopted as single representative values rather than measured for a specific soil and tyre combination; in practice these coefficients vary with soil moisture, tillage state and tyre inflation pressure, and should be identified experimentally for any specific application. Third, the rollover criterion of Section II-D is a static, quasi-steady approximation that neglects transient weight transfer during rapid steering inputs and neglects the compliance of the tyres and any suspension; a fully dynamic, multi-body rollover analysis, of the type applied to tractor–trailer systems in and, would be required before the numerical rollover-speed values reported here could be used for a safety-critical design decision. Because no physical prototype was available, none of the numerical results presented in this paper have been experimentally validated; they should be read as a theoretical, assumption-based evaluation of the proposed model, intended to guide subsequent prototype design and experimental testing rather than to substitute for it[15].

Conclusion

A combined kinematic–dynamic model of the running gear of a three-wheeled mini-tractor with one steered front wheel and two driven rear wheels has been developed from first principles, using the tricycle kinematic formulation together with Newton's second law for the longitudinal dynamics and a static-stability-factor criterion for lateral rollover. Numerical evaluation of the model, using explicitly declared assumed design parameters representative of a small mini-tractor class, shows that (i) the turning radius falls from about 16.0 m to 2.0 m as the front-wheel steering angle increases from 5° to 35°, while the theoretical critical rollover speed falls correspondingly from about 43 km/h to 15 km/h, confirming a direct manoeuvrability–stability trade-off governed principally by the rear track width and the centre-of-

gravity height; (ii) at sharp steering angles the two rear wheels are required to roll at markedly different speeds (a 67% difference at 35°), indicating that a rigid, non-differential rear axle is likely to introduce significant wheel slip during headland turns; and (iii) the achievable working speed under draft load is governed by the adhesion-limited tractive force at the rear wheels, falling from a transmission-limited 25 km/h when unloaded to about 9.8 km/h when the draft resistance of a mouldboard plough brings the total resistance close to the adhesion limit. On this basis it is recommended that the design of three-wheeled mini-tractor running gear treat the rear track width, the centre-of-gravity height and the maximum steering angle as a coupled design decision rather than optimizing turning radius in isolation, and that a differential or overrunning-clutch rear axle be considered for machines intended for frequent sharp turning. As the model and its numerical results are based on assumed rather than experimentally identified parameters, the priority for future work is the construction of an instrumented physical prototype to measure the rolling-resistance and adhesion coefficients, the actual centre-of-gravity position, and the dynamic (rather than static) rollover threshold, so that the theoretical results obtained here can be validated and refined.

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