A long-tracked bogie design for forestry machines on soft and rough terrain

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Abstract

A new design for a tracked forestry machine bogie (Long Track Bogie; LTB) on soft and rough terrain is investigated using nonsmooth multibody dynamics simulation. The new bogie has a big wheel that is connected to and aligned with the chassis main axis. A bogie frame is mounted on the wheel axis but left to rotate freely up to a maximum angle and smaller wheels that also rotate freely are mounted on the frame legs with axes plane parallel to the driving wheel. The wheels are covered by a single conventional forestry machine metal track. The new bogie is shown to have higher mobility and cause less ground damage than a conventional tracked bogie but requires larger torque to create the same traction force as a conventional bogie. The new bogie also gives less acceleration when passing obstacles than the conventional bogie. Additionally, due to the shape and size of the new bogie concept, it can pass wider ditches.

Keywords: Bogie; Forest machine; Ground damage; Mobility; Multibody dynamics simulation; Off-road; Rough terrain; Tracks; Traction;

1. Introduction

One of the challenging problems of the forest industry is to minimize the ground damage caused by heavy forestry machines during harvesting. The ground damage can lead to severe problems like soil compaction, deep ruts and damage to the roots of the remaining trees after thinning.

Roots increase soil stability and therefore damage on the roots causes reduction on soil stability [1]. Soil compaction can negatively change soil properties, such as drainage, and results in a growth reduction and increased surface run-off [2]. Subsoil compaction is a severe problem because the effects are long-lasting [3]. Compacted soil can cause up to a 50% reduction in the growth of trees remaining after thinning, and in the subsequent forest crop after final felling. This reduces the value of the wood when it is harvested [4]. Soil erosion and particle transport to streams are additional problems that can occur as a result of soil damage. When soil is laid bare in ruts or wheel tracks, rain and surface water erode the soil more easily, especially in steeply inclined areas. This erosion can also adversely affect aquatic ecosystems, primarily from suspended particles altering the light levels within water bodies, thus affecting photosynthesis. Other effects include

increased phosphorous and nitrogen levels in aquatic ecosystems, which can cause eutrophication [5] that can have serious consequences for productivity.

In order to reduce these negative impacts, different techniques like using larger tyres, tyres with lower pressure, bogies and bogie tracks have been introduced to the industry [4, 6, 7, 8, 9].

Despite the fact that wider and softer tyres can reduce the soil disturbances, wheeled machines have mobility limitation in rough terrains. There are limitations that rule out the possibility to overcome obstacles larger than the wheel's radius. They also need land with continuous surface to operate on [10].

As an alternative to the conventional wheeled machine, bogies can be a proper solution to the mobility problem. Bogies increase the traction and stability by basically maintaining the contact between the rough terrain and wheels. Additionally, the bogie systems provide more smoothness when the machine overcomes obstacles [7, 10, 11]. To increase the ground contact area and the traction, tracks are used over the bogie wheels. In this article, machines equipped with bogies with bogie tracks will be refered to as tracked machines.

The wheeled machine and tracked machine performance differs mainly in speed and soil damage. Herein, soil damage refers to soil displacement and soil compaction. Tracked machines have more contact with the ground which leads to less ground pressure. Moreover, due to the increased area, they can operate on softer and wetter soil without loosing traction compared to the wheeled machine. On the other hand, wheeled machines have higher top speed than tracked machines on flat firm ground. But the main drawback of the tracked machine is the amount of soil displacement when the machine is turning, particularly in a limited area [12].

The impacts of the wheeled machine and the tracked machine on the forest have been studied in Myhrman [13], Bygdèn et al. [7], Bygdén and Wästerlund [14], Sakai et al. [15], as well as investigation on the mobility of the machines in different conditions. Comparison between the wheeled and tracked machines emphasizes the fact that it is desirable to benefit from advantages of both types of machines.

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This study considers a new design of tracked bogie track, called the *long-tracked-bogie* (LTB). The idea behind the LTB is to combine features of wheels and tracks in a single bogie design. A big driving wheel is connected to and aligned with the chassis main axis. A bogie frame is mounted on the wheel axis but left to rotate freely up to a maximum angle. Smaller wheels are mounted on the frame legs with axes plane parallel to the driving wheel. The smaller wheels rotate freely. The three wheels are covered by a single conventional forestry machine metal track. The LTB is illustrated in Fig. 1 together with a conventional bogie, which we will denote *c-bogie*.

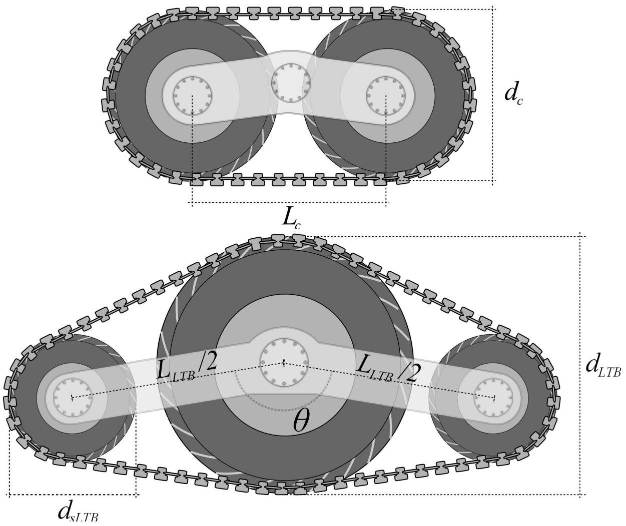

Figure 1: A conventional tracked bogie (top) and the proposed long-tracked bogie (bottom).

A suitable geometric shape of the LTB geometry is expected to provide a large contact surface (low contact pressure and good traction) on soft ground but minimal contact surface when turning on firm ground (small shear and soil displacement). We adjust the LTB geomtry with the frame angle, θ, and ratio, $d_{\rm sLTB}/d_{\rm LTB}$, between small and big wheel diameter. The free rotating frame aims to smooth the path of the center of gravity when the machine passes an obstacle just like a c-bogie. The frame rotation is mechanically restricted at some maximum angle to prevent the bogie from turning over instead of pulling the machine.

The LTB was proposed by Lars-Gunnar Nilsson and Johannes Nilsson at Vimek AB located in Vindeln, Sweden. Similar design ideas can be found for wheeled robots and smaller agriculture machines, e.g., the Galileo Wheel [16].

The purpose of this study is to critically investigate the proposed features of the LTB for heavy forestry machines by providing numerical evidence and finding the optimal choice of LTB frame angle and wheel diameters considering mobility and ground damage.

2. Model and computation method

The comparative study is carried out using computer simulation of two identical machines with the different bogies. The modeling and simulation framework is described in Sec. 2.1 and the particular machine, bogie and track models in Sec. 2.2.

2.1. Non-smooth rigid multibody dynamics

We use the framework of *non-smooth rigid multibody dynamics* [17, 18, 19]. A mechanical system is modeled as a collection of rigid bodies of various geometric shape and mass. Articulated mechanisms are modeled by imposing kinematic constraints on the bodies relative positions and velocities, e.g., hinge constraint for connecting wheel and wheel axes to the chassis. The constraints give rise to additional (constraint) forces to the Newton-Euler equations of motion. Secondary constraints can be imposed on the remaining degrees of freedom to model motors, joint limits and internal friction. Impacts and frictional contacts are introduced dynamically, triggered by the contact detection process, as additional constraints by assuming the Signorini-Coulomb law and the Newton impact law. Frictional contacts and seconday constraints brings complementarity conditions to the equations of motion and makes the velocity discontinuous in time, i.e., non-smooth.

We use a particular variational integrator scheme [20, 21] where constraint regularization and stabilization parameters can be mapped to physical viscoelasticity and enables stable time-integration at large time-steps. Each time-step involves solving a *mixed linear complementarity problem* (MLCP) with problem size usually 2-10 times the number of degrees of freedom (*dof*), depending on the number of contacts and drive-line complexity.

We apply a direct block-sparse solver that provides real-time interactive simulation for simple systems involving machine (roughly 100 *dof*) and contacting objects, e.g., rocks or logs, (1000 dof). The simulations were performed using *AgX Multiphysics Toolkit version 1.10* [22]. Gravity is set to 9.81 m/s^2, time step 0.01 s and direct solver settings on both normal and frictional forces.

2.2. Machine, bogie and track model

Two identical articulated machine configurations with different bogies are considered. The reference configuration uses conventional tracked bogies, referred to as *c-bogie* in short. In the other configuration, the c-bogies are replaces with the proposed LTB. Apart from the bogie the machine configurations are identical, see Fig. 2. The machine chassis is that of a small forwarder - a machine used for off-road transportation of logs between the felling place and transportation road in cut-to-length forestry systems. The mass of the front and rear chassis is 2 tons and 3.5 tons, respectively. The articulation is modeled as a universal joint with relative pitch motion constrained and torque motor for controlling the yaw motion when steering. The bogie main axes are centered on each chassis and modeled using hinge constraints. The distance between articulation joint center and front and rear axes is $l_{\rm f} = 1.1$ m and $l_{\rm r} = 2.0$ m, respectively. The transmission is modeled by individual efforts constraints (hinge motor) providing the c-bogie wheel axis and LTB main axis with torque to maintain a specified angular velocity. Differential unlocking is applied when turning to minimize contact slippage. The bogies have a maximum tilt angle set to $\theta_{\rm max} = 15$ degrees.

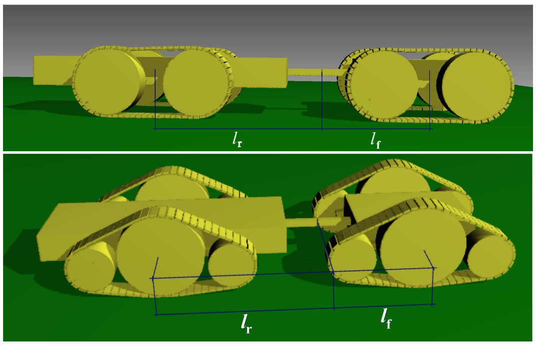

Figure 2: The conventional bogie (top) and the LTB (bottom) on a simple machine model.

Table 1: Geometrical and physical properties.

Notation	Value	Comment
θ	*variable*	LTB frame angle
d_c	0.874 m	c-bogie wheel diameter
d_{LTB}	1.2 m	LTB big wheel diameter
d_{sLTB}	*variable*	LTB small wheel diameter
l_f	1.1 m	articulation to front axis
l_r	2.0 m	articulation to rear axis
L_{LTB}	1.92 m	LTB frame length
L_c	1.17 m	c-bogie frame length
w	0.38 m	track and wheel width
M	10 t	machine weight

The c-bogie is modeled as a rigid body frame with two rigid wheels rotating at identical angular velocity. The LTB consists of a big center wheel at the main bogie axis. A rigid frame is also attached to the bogie axis but rotates freely up to the maximum tilt angle. At both ends of the frame there is a smaller wheel that is also free to rotate. All wheels are given cylindrical shape and identical width $w = 0.3$ m. The dimension and mass of the frames are, $L_c = 1.17$ m and $m_{bc} = 200$ kg for the c-bogie and $L_{LTB} = 1.92$ m and $m_{LTB} = 200$ kg for the LTB. The big wheels of the LTB have mass 200 kg. The wheels on the c-bogie have mass 100 kg. Each of the 20 track elements have mass 15 kg.

Each bogie is covered by a metal track that is modeled as rigid body elements linked together by hinge constraints. Each track element is also constrained to move in the co-moving symmetry plane of the bogie frame. The track and bogie wheels interact through contact forces with the friction coefficient set to infinity to model the strong grip between track elements and deep tyre tread. Dry friction in the links between the track elements is modeled using nonholonomic effort constraints. The link friction is important for numerical stability at large time-step integration.

The geometric dimensions and physical properties of the machine, bogies, wheels[2] and tracks are summarized in Table 1 with notations according to Fig. 1 and 2. The total machine weight is about 10 tonnes. This represents a small forwarder that is commonly used in thinning on softer (weaker) ground.

In the numerical simulations the machine is driven along a specified path at specified velocity. A motion tracker and motion planner is implemented to maintain that path by setting the wheel axis target velocity and target turning angle.

The terrain is represented as a static height field with variable mesh-size down to 0.01 m. The contact model between track and terrain includes friction coefficient set to $\mu = 1.0$ and elasticity[3] set to $Y = 40$ GPa. The elasticity is mapped to the regularization parameters of the numerical method and is assumed to include also the tyre elasticity. A rough terrain test course was constructed and specifically obstacles in the form of a step and square ditch.

3. Simulation study

Simulation of machine dynamics with the different bogies and different design variables (θ and d_{sLTB}) was performed for different test cases: *i)* driving over a step of height 0.1 m, 0.3 m and 0.4 m; *ii)* driving over a square ditch of width 1 m, 1.5 m and 1.9 m; *iii)* making a U-turn with 10 m radius. The machines where driven on both axes and at a target speed 1 m/s. Images from the simulations can be found in Fig. 10-14 in the Appendix.

The design variables are varied in different simulations as shown in Table 2. In total 125 simulations of roughly 6000 s are performed.

θ	148°	154°	160°	166°	172°	180°
d_{sLTB}	0.45 m	0.55 m	0.65 m			

Table 2: Design variables

Position, orientation, velocity, acceleration and torque of the chassis and the bogies are logged for each time-step during simulation. All contact data – position, penetration depth, normal and tangential velocities and forces – for track and ground interaction is also stored as time series. In post-processing measures for soil displacement, mobility, slippage, ground pressure and contact area are computed. These measures are used in study of the dependency of the design variables on *soil displacement* and *mobility*. Soil displacement is the volume of soil displaced by the tracks over time. Each contact point i contributes to the soil displacement rate by an amount proportional to the relative surface velocity v_i and the cross-section area associated with the contact point. The contact point cross-section area is estimated to $A_i = d_i w/2$, where d_i is the contact depth and w the width

[2]The small wheel diameters exist on market and may be put in parallel to match the width of the big wheel and the track. The big wheel diameter has been chosen as big as possible without inflicting the load space volume on a forwarder machine.

[3]In AgX 1.10 the elasticity value for contacts is purely a regularization parameter rather than physical elasticity

of the track element. Rather than choosing a soil model and geometric track shape we simply assume a constant track penetration depth $d_i = 0.01$ m (considering turning on firm forest ground). This is a reasonable model for studying the *relative* difference between the LTB and conventional bogie. The net soil displacement is thus computed

$$V = \int_{t_0}^{t_1} \left[\sum_{i=1}^{N_c(t)} A_i v_i \right] \mathrm{d}t \tag{1}$$

where $N_c(t)$ is the number of contact points between the track and ground at time t, ranging from start and end time of the simulations, t_0 and t_1.

The mobility is measured from the acceleration of one point in the front chassis, e.g., where a driver would be positioned. We use the root-mean-square acceleration

$$\langle a \rangle = \frac{1}{t_1 - t_0} \left[\int_{t_0}^{t_1} |\dot{v} + \dot{\omega} \times r|^2 \, \mathrm{d}t \right]^{1/2} \tag{2}$$

where v is the chassis center of mass velocity vector, ω the chassis angular velocity vector and r the measurement point relative position vector to the front chassis center of mass. We choose $r = [0, 0, 1]$ m. The contribution from rotational motion to the velocity and acceleration measurements turned out to be negligible compared to the linear motion, however. Good mobility is small $\langle a \rangle$ which means smooth motion over the obstacles. Poor mobility is large acceleration or inability to pass the obstacles.

The secondary measures are the slippage from the wheel to the ground, the contact pressure force from the machine on the ground and the contact area from the bogie. The slippage is computed as the mean relative velocity in the contact tangent plane of the contacting track elements. The ground contact pressure force, or simply pressure, is the total force (from all four bogies) from the machine acting on the ground. Also the contact area is summed for the four bogies.

4. Results

The main goal of the simulation is to find the optimal design for the LTB and then compare its performances with that of a c-bogie. As shown in Sec. 4.1, the simulation results suggest the values for the optimal design variables to be frame angle $\theta = 160$ degrees and small wheel diameter $d_{\mathrm{sLTB}} = 0.65$ m. The mean acceleration and soil displacement for this set of design parameter values of LTB and for the c-bogie are found in Table 3 and 4, respectively. From Table 3 we observe that the mean acceleration for the LTB is roughly 50% smaller than for the conventional bogie. For obvious reasons the LTB can pass wider ditches than the c-bogie. The LTB can pass ditches up to 1.9 m wide while the c-bogie can pass ditches that are maximally 1.6 m wide. The soil displacement for the optimal LTB in the U-turn test is 44% smaller than for the c-bogie.

Time series of the chassis longitudinal velocity, longitudinal acceleration, average torque, total slip, pressure and total contact area was compiled and analyzed. Sample time series for the optimal LTB passing the 0.4 m step obstacle are shown in Fig. 3 and 4 with time series for the c-bogie included for reference. A general trend is that acceleration peaks for the LTB are smaller than for the conventional and the LTB has smaller variations in contact area and thus better preserved traction. A critical point, however occurs for the LTB when climbing step obstacles. When the front bogie center wheel reaches the corner of the step the contact area become minimal and traction drops. This can be seen in the time series in Fig. 4 around the time point 16 s. Also a large torque is required to overcome the obstacle at the desired speed as seen in Fig. 3.

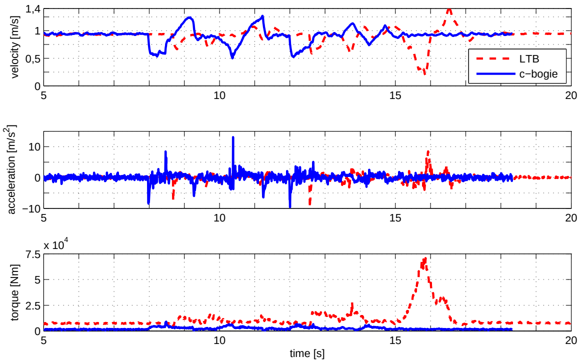

Figure 3: Time series of longitudinal velocity, longitudinal acceleration and torque for optimal LTB passing the 0.4 m step obstacle.

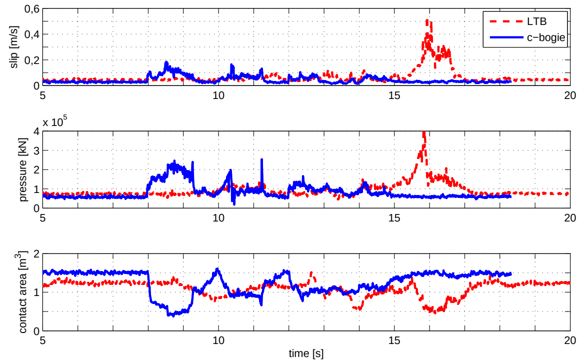

Figure 4: Time series of slip, pressure force and contact area for optimal LTB passing the 0.4 m step obstacle.

Sample time series of the optimal LTB for the U-turn are shown in Fig. 5 and 6. The data shows that the slip of the c-bogie is considerably larger than for the LTB at the turning points in the U-turn. The c-bogie frame is raised to one wheel at the turning point around $t = 25$ s by the involved forces but the resulting net soil displacement is still larger than for the LTB, as seen in Table 4.

	step 0.1	step 0.3	step 0.4	ditch 1	ditch 1.5	ditch 1.9
LTB	2.6	2.9	3.4	2.8	3.6	7.0
c-bogie	4.6	5.0	4.5	5.0	7.1	fail

Table 3: Mean acceleration $\langle a \rangle$ [m/s^2] for optimal LTB and conventional bogie.

	Soil displacement	Mean contact area	Mean acceleration
LTB	0.9 m^3	1.0 m^2	2.4 m/s^2
c-bogie	1.6 m^3	1.1 m^2	4.0 m/s^2

Table 4: Soil displacement, contact area and acceleration for optimal LTB and conventional bogie in a U-turn.

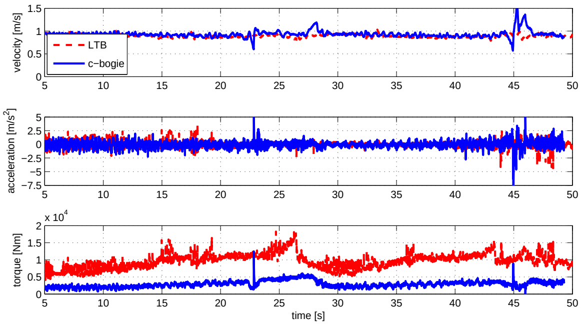

Figure 5: Time series of longitudinal velocity, longitudinal acceleration and torque for optimal LTB in U-turn.

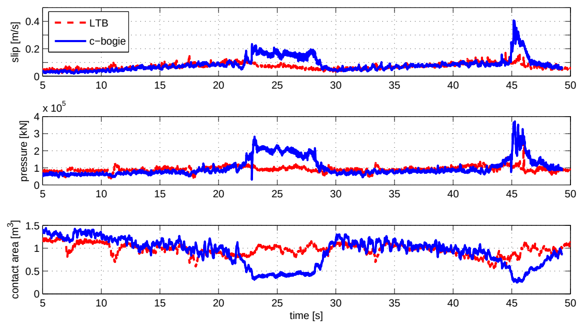

Figure 6: Time series of slip, pressure force and contact area for optimal LTB in U-turn.

4.1. *Dependency on design parameter values*

Figures 7 and 8 show how the mean acceleration for step and ditch crossing depends on the choice of frame angle θ and small wheel diameter d_{sLTB}. The corresponding value for the c-bogie is added for reference. From the step obstacle data in Figure 7 we conclude that the frame angle is best chosen between 160 and 175 degrees to minimize the acceleration. The ditch test results in Figure 8 suggests frame angles less than 175 degrees and the largest diameter of the small wheel. Only the bogie with largest small wheel diameter was able to pass the 1.9 m wide ditch.

The soil displacement depending on frame angle and small wheel diameter is shown in Figure 9, again with the corresponding value for the c-bogie included for reference. As can be expected large frame angle and small wheel diameter minimizes the soil displacement. But in the region between 155 and 170 degrees the soil displacement do not depend strongly on wheel diameter.

Altogether the dependency of acceleration and soil displacement on bogie design variables points to the choice of frame angle to 160 degrees and wheel size $d_{\mathrm{sLTB}} = 0.65$ m.

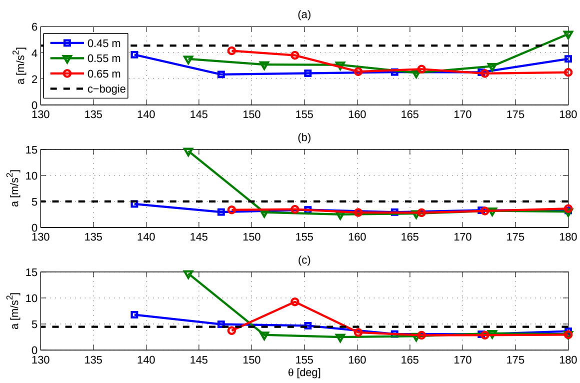

Figure 7: Mean acceleration for step obstacle as function of frame angle and for different small wheel diameter. The steps have height 0.1 m (a), 0.3 m (b), 0.4 m (c).

5. Discussion

The optimal LTB bogie has higher mobility than the conventional bogie in terms of smaller mean acceleration and ability

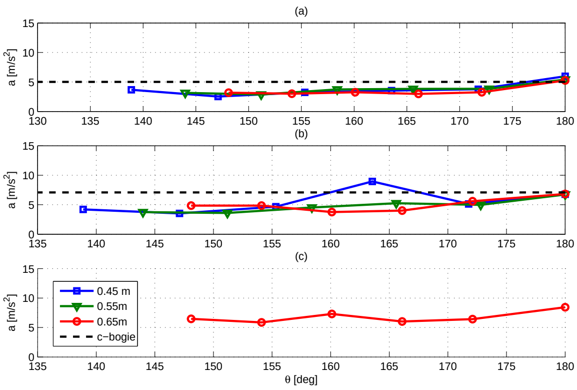

Figure 8: Mean acceleration for square ditch obstacle as function of frame angle and for different small wheel diameter. The ditches have width 1 m (a), 1.5 m (b) and 1.9 m (c).

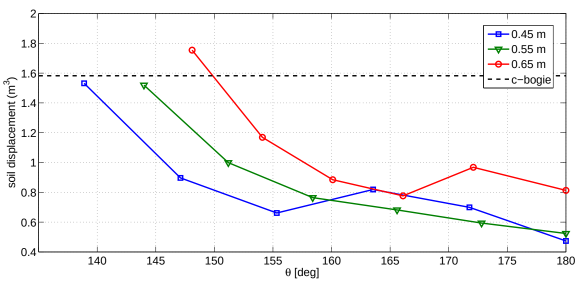

Figure 9: Soil displacement for U-turn as function of frame angle and for different small wheel diameter.

of passing wider ditches. The explanation is that the length and shape of the bogie results in a large contact area, and thereby traction, on uneven terrain and smooth climbing over obstacles thanks to its larger size and suitable shape. At the chosen angle the soil displacement is also smaller when turning on flat firm ground.

The maximum load for a given bearing capacity can also be increased using the LTB. The maximum ruth depth for avoiding ground damages are 0.1 m. For a c-bogie, the contact area on soft ground is roughly $(L_c+0.5d_c)\times w$ while for the optimal LTB it is roughly $(L_{\rm LTB}\sin(\theta/2) + 0.5d_{\rm sLTB}) \times w$ when it is sunken more than 0.1 m and the track is in contact with ground all the way to the small wheels. For the given bogie parameters the numerical values of these areas are 0.61 m^2 and 0.85 m^2, respectively, which means that the LTB can potentially carry up to 40% larger load because of larger contact area to distribute the load pressure on. The effect on the bogie alignment relative to the ground when applying torque is assumed negligible.

Increased mobility and maximum load can in turn shorten the routes and decrease the transportation time. This implies reduced fuel and personal costs. The amount depends strongly on the particular terrain characteristics. As an elucidating example, the forwarding costs in Sweden are estimated to 340 million EUR per year and a 5% decrease in cost would thus result in a gain of 17 million EUR. The estimation is based on yearly production volume of 73 million m^3 and forwarding costs ranging between 4.3 EUR/m^3 (harvesting) and 6.8 EUR/m^3 (thinning).

One drawback of the LTB bogie is that it requires larger torque than the c-bogie. To produce the same traction force the LTB require a torque that is a factor $d_{\rm LTB}/d_c$ times that for a c-bogie, i.e., 37% larger in this specific case. When climbing over high step obstacles, e.g., 0.4 m, the LTB bogie has a critical point where it looses large contact and traction. This will be pronounced for more smooth obstacles. The larger required torque might negatively affect the fuel economy.

The main uncertainty of the present study concerns the lack of empiric data for model validation and the choice of test cases. For simplification the vehicle and terrain have been modelled as a rigid multibody system. This includes the assumption that the tyres and terrain can be modelled as rigid bodies. Regularization of the tyre-terrain contact constraint is employed and brings some elasticity but is not based on any particular model for tyres and terrain. We observe that numerical dissipation in the track links are present but no validation against dissipation in real tracks has been made. This numerical dissipation is emphasized for the LTB, with higher radius of curvature, resulting in larger torque even on flat ground than for conventional bogie. All other errors associated with the model and simplifications are similar for both bogies and thus not alter the qualitative results that are based on comparison rather than on absolute values.

The test cases and search for optimal design also have limitations. A larger and more resolved parameter space for both test scenes and bogie design would have been preferable. The current simulation study was based on 125 simulations and totally 6000 seconds of dynamics. Each second taking roughly 10 seconds for computing on a conventional desktop computer. This gives a total computing time of 17 hours plus post-processing computations of a few hours. Extending to 4 design variables, each discretized by 10 values and 10 tests scenes in total would have changed this into roughly 10^5 simulations, 10^6 seconds of dynamics time and 10^7 seconds of computation time, i.e., roughly 100 CPU days. It would also have been preferable to run the tests with different vehicle speed.

A physical full-scale setup of vehicles with LTB and c-bogie as described in this paper is under construction and testing on step and ditch obstacles. Also, tyre and terrain deformation models are under development as well as parallelized solvers and simulation framework to handle more fine-grained models and test cases with good computational speed. Future research on bogies for increased mobility and reduced ground damage should include validation between simulation models and physical prototypes to give virtual tests higher predictive value. Also, the possibility of applying active control on the bogie frame rotation, frame angle and length should be considered. This way the bogie can be interactively shaped to the optimal design best fitting the current obstacle and terrain conditions.

6. Conclusions

The results show that the LTB combine the advantages from both a wheeled machine and a bogie machine. The LTB, with the selected parameters, has significantly less soil displacement than the conventional bogie when turning on firm ground. The LTB bogie can pass wider ditches, have better ride smoothness and can carry larger loads on soft ground. It does, however require larger torque.

7. Acknowledgements

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Supplementary material

Animations from simulations can be found on the web page `http://umit.cs.umu.se/wiki/LTB`.

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Appendix

Image sequences from simulations are shown in Fig. 10-14.

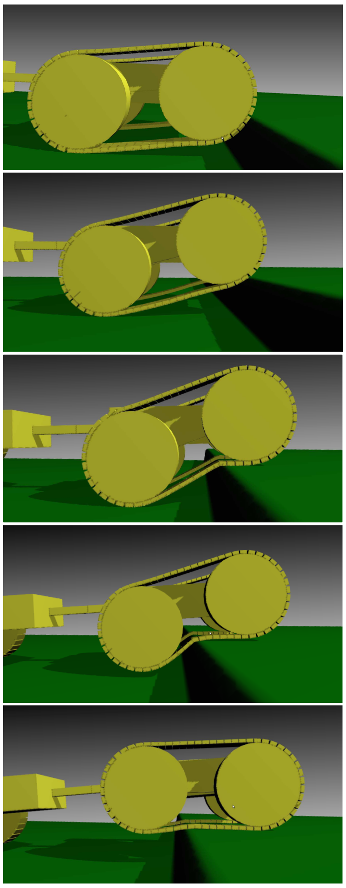

Figure 10: Image sequence of the conventional bogie taking the step obstacle of height 0.4 m.

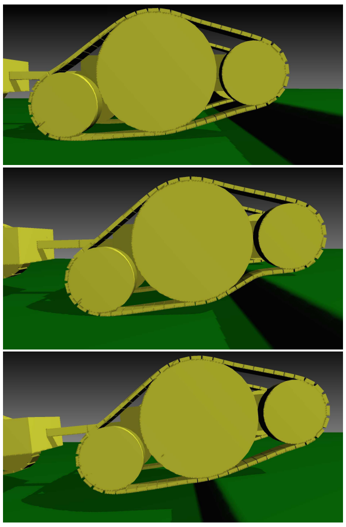

Figure 11: Image sequence of the LTB taking the step obstacle of height 0.4 m.

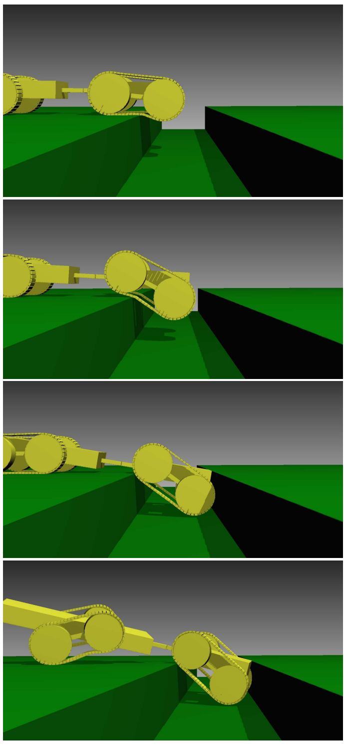

Figure 12: Image sequence of the conventional bogie taking the square ditch obstacle of width 1.9 m.

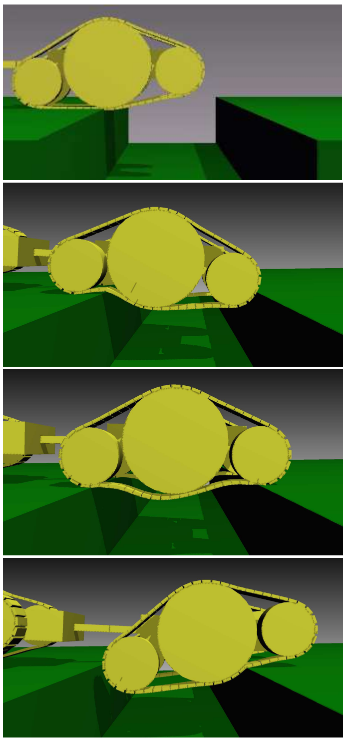

Figure 13: Image sequence of the LTB taking the square ditch obstacle of width 1.9 m.

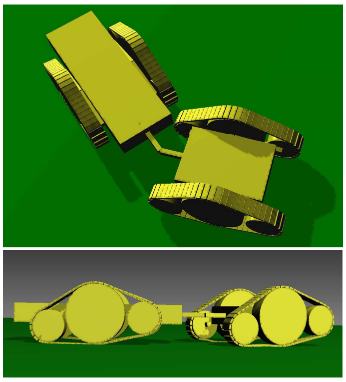

Figure 14: Image from U-turn of 10 m radius with the LTB.