OPTIMISATION OF THE LOCOMOTION PERFORMANCE FOR EXPLORATION ROVER SYSTEMS USING A HIGHLY VARIABLE CHASSIS PLATFORM

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ABSTRACT

A growing number of planetary robotic exploration rover systems accessed the surfaces of celestial bodies of the Solar System within the last decades, providing countless of scientific data indispensable for humanity exploring the universe. Even more surface exploration missions using robotic systems shall follow in the next decades. Amongst numerous factors for successful planetary surface exploration with rover systems, one key driver are the mobility capabilities of the system. Since building up multiple counts of breadboard chassis models to evaluate and verify the mobility aspects of different configurations results in a costly development process, as well as in high hardware costs for the practical realisation, we are investigating the implementation of an efficient, reconfigurable concept for the design of a rover locomotion system. Within this paper a general approach for a locomotion system design development shall be discussed, as the design of hardware models is still required for the verification of modern simulation tools.

1. INTRODUCTION AND MOTIVATION

In order to analyse a variety of locomotion performance factors, without excessively driving the development and hardware costs, a design approach using a reconfigurable concept was chosen. A high degree of reconfigurability of the chassis is achieved by a possible variation of the type of suspension system itself, including a further adaptability of its geometric parameters in particular. Besides the kinematic parameters, also size and type of wheels, as well as a variability of the chassis centre of mass is implemented. For a first evaluation, two types of suspension systems are adaptable to the chassis: a rocker-bogie suspension, comparable to the ones developed for the NASA Mars rover systems, and a 3-bogie suspension system, as used for the ESA ExoMars rover. Both concepts were chosen due to their moderate complexity while providing reliable mobility capabilities.

Due to the described adaptability of the suspension system, the chassis is used as a highly variable development and research platform for in-depth analysis, identifying kinematic and mechanical design drivers for efficient mobility performance of a robotic chassis locomotion system. This approach allows the analysis of several different mission profiles due to the reconfigurability of the chassis and its performance drivers. Besides the analysis of maximum stability and trafficability boundaries, the concept of a variable and reliable chassis also serves a research platform of future scientific and robotic payloads and rover subsystems. For the first performance analysis of the locomotion system, design requirements for the operational capabilities have been formulated, such as driving slopes of at least 20 degrees under varying surface and soil conditions, as well as fundamental capabilities of obstacle overcoming. The requirement for the overall chassis mass is set to a maximum of up to 15 kg to maintain a comparative manageability of the system.

Based on these requirements a hardware model using commercial off the shelf components shall be realised for the functional verification. This hardware model providing the above described flexibility is used to optimise the performance of the chassis for the defined requirements. The optimisation process shall be furthermore supported by setting up a simulation environment providing information on the capabilities of the locomotion system in dependence of the variable parameters.

2. GROUND INTERACTION AND WHEEL DESIGN MODEL

The basic approach and aspects commonly used to predict and estimate the locomotion performance of a mobile surface vehicle in rough terrain have been formulated by Bekker [1][2] and Wong [3] and are generally described by the Drawbar Pull (DP) metrics. As the performance is strongly dependent on the interaction between the rover and its environment, affected by the overall chassis design as well as the soil properties on the surface, the implementation of a maximum of adaptability shall be achieved by a generic approach for the wheel design applicable to the chassis.

Table 1: Characteristic soil properties for dry sand [3]

Soil cohesion	C_0	1,04	kPa
Soil friction angle	φ	28	$\circ$
Modulus of cohesion	k_c	0,99	kN/m^{n+1}
Modulus of friction	k_φ	1528,43	kN/m^{n+2}
Deformation coefficient	n	1,1	

For operating under terrestrial conditions the soil characteristics for loose sand according to [3] are found in Tab. 1 and shall be used for the calculation of the ground interaction metrics. A variation of the characteristic soil parameters effects the mobile capabilities of the chassis respectively, wherefore a reconfiguration of different wheel shapes and sizes shall be realised.

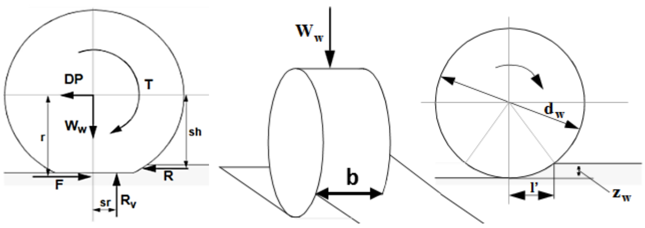

Figure 1: Wheel soil interaction model [4]

The equations for the wheel-soil interaction as formulated by Bekker and analysed by Apostolopoulos [4] shall be derived for setting up the model. For calculating the drawbar pull performance of a (rigid) powered wheel a model as shown in Fig. 1 is used. Depending on the wheel geometric dimensions and the soil characteristics the maximum static sinkage (z_W) of a wheel in soft soil can be described by Eq. 1.

$$z_W = \left(\frac{3W_W}{(3-n)(k_c + b_w k_\varphi)\sqrt{d_W}} \right)^{\frac{2}{2n+1}} \tag{1}$$

Analyses of the effects of wheel sinkage in loose soil are mandatory for estimating the maximum tractive force, as well as the motion resistance of the soil. Compaction resistance (R_C) due to deformation of the soil beneath the wheel is formulated in Eq. 2. As the rover shall traverse slopes the gravitational resistance (R_g) depending on the inclining terrain need to be taken into account.

$$R_C = \frac{z_W^{n+1}}{n+1}(k_c + bk_\varphi) \tag{2}$$

$$R_g = W_W \sin\theta \tag{3}$$

Limited by the mechanical and geochemical properties of the soil, the individual load and contact area between wheel and soil, the soil thrust (H) describes the maximum tractive force that can be sustained. For a positive drawbar pull and therefore effective forward motion of the rover system, this traction needs to exceed motion resistance. Eqs. 1-5 are used for calculation of the drawbar pull parameters.

$$H = AC_0 + W\mu = AC_0 + W\tan\varphi \tag{4}$$

$$A = b_W\sqrt{(d_W - z_W)z_W} \tag{5}$$

$$DP = H - \sum R \tag{6}$$

By applying the parameters for a design point respecting the formulated requirements an envelope for possible realistic wheel sizes as shown in Fig. 2 can be derived.

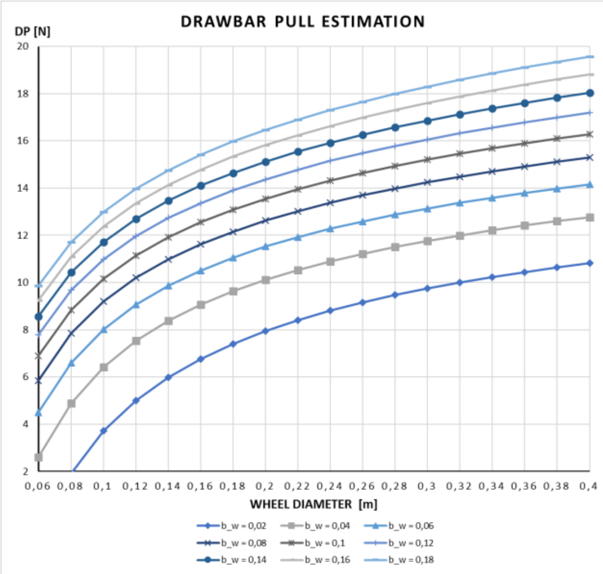

Figure 2: Wheel design impact on drawbar pull

Due to the aspects of reconfigurability, for an optimised locomotion performance wheel sizes from diameters of at least 60 mm shall be applicable to the cassis. The lower limit can be determined by the estimated slope gradeability depending on a specific drawbar pull performance as given in Eq. 7. The upper limit is compromised from reasonable estimations which come along with aspects regarding to wheel and overall system mass, as well as stowability concerns due to increasing the rover footprint. Therefore, wheel diameters above 300 mm doesn`t seem reasonable.

$$\varphi_{max} = \tan^{-1}\left(\frac{DP}{W_W}\right) \qquad (7)$$

The requirement of reconfiguration implies a constant iteration process for the concept design. The application of smaller wheels with less mass results in lower individual wheel loads, as well as the possibility of a general smaller sizing of the overall rover footprint. In contrast especially the drawbar pull performance is decreased.

3. LOCOMOTION SYSTEM CONFIGURATION DESIGN

To analyse the performance driver of the suspension system the design variables for the kinematic configuration were formulated as shown in Fig. 3. A six-wheeled locomotion system is chosen as the favoured initial design point for a basic chassis configuration. Nevertheless, the adaption to an n-wheeled locomotion system can also be realised due to the high degree of reconfiguration the described concept allows. In terms of complexity the initial design will consists out of a passive suspension system and rigid linkages without additional actuators or springs.

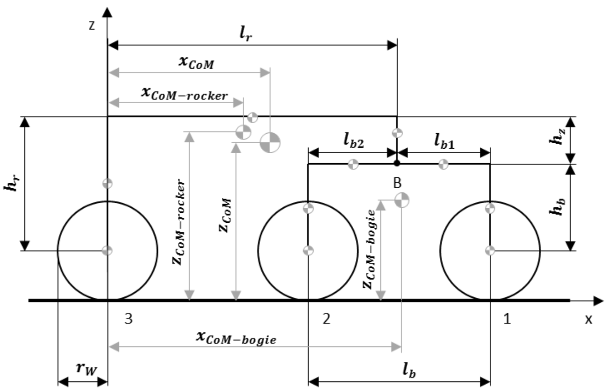

Figure 3: Design variables suspension system

The focus of the suspension system analyses is to determine impact factors of changing geometric proportions on the performance metrics of the chassis, specified by the wheel load equalisation capabilities while traversing. Varying wheel loads significantly change mobile capabilities on loose soil and while climbing obstacles. Fig. 3 shows the exemplary definition of the design variables for a rocker-bogie configuration, assuming a locomotion concept with a 6 x 6 x 4 wheel formula (“total number of wheels” x “number of powered wheels” x “number or steerable wheels”) and respectively point mass assumptions for each powered wheel as well as for the linkages. Within this paper only the analytical implementation of the rocker-bogie system into the chassis design model is described as showcase, though an implementation of different suspension system models will be provided.

3.1. Static stability model

To estimate the resulting wheel loads a general analytical approach assessing quasi-static conditions for low speeds while traversing and climbing obstacles is used. Fig. 4 describes the free body diagram of the rocker-bogie system. All design variables of the kinematic structure along with additional subsystem mass variables (mass of electronic components, battery mass, body structure mass, etc.) are implemented into the model to define the estimated coordinates of the overall Centre of Mass (CoM) for calculation of the reacting forces.

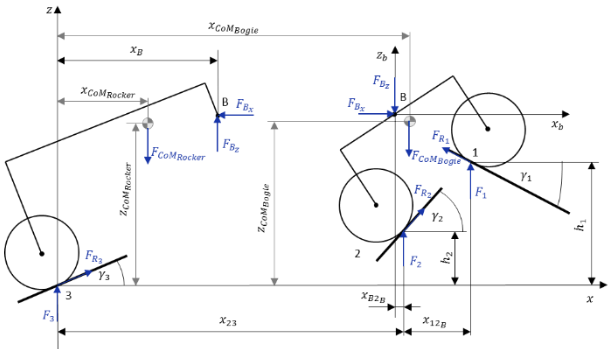

Figure 4: Design variables of suspension system for force analyses

To implement the changing wheel-surface conditions as well as the shift of the respective CoM while traversing on rough terrain an individual surface contact angle γ_i as well as the wheel height h_i needs to be determined. This approach for analysing one single side of the kinematic only, is limited to a two dimensional force analyses, whereas effects of a roll and yaw motion of the chassis are not respected.

Formulating the equations for the static force equilibrium to estimate the normal forces under stationary conditions the respecting γ_i and h_i equal zero. The equations for force equilibrium of the rocker system follow as defined:

$$F_x = 0 \qquad (8)$$

$$F_{CoM_{Rocker}} = F_3 + F_{B_z} \qquad (9)$$

$$F_{CoM_{Rocker}} x_{CoM_{Rocker}} = F_{B_z} x_B \qquad (10)$$

For the bogie accordingly:

$$F_x = 0 \qquad (11)$$

$$F_{B_z} + F_{CoM_{Bogie}} = F_1 + F_2 \tag{12}$$

$$-F_{B_z} x_{B2_B} + F_{CoM_{Bogie}} \left(x_{CoM_{Bogie}} - x_{23}\right) = F_1 x_{12_B} \tag{13}$$

As all variables are in relative relation to the geometric properties of the overall locomotion system design. The respective forces can be calculated after Eqs. 14-16.

$$F_1 = \frac{1}{x_{12_B}} \left(F_{CoM_{Rocker}} \frac{x_{CoM_{Rocker}}}{x_B} \left(-x_{B2_B}\right) + F_{CoM_{Bogie}} \left(x_{CoM_{Bogie}} - x_{23}\right) \right) \tag{14}$$

$$F_2 = F_{CoM_{Rocker}} \frac{x_{CoM_{Rocker}}}{x_B} + F_{CoM_{Bogie}} - F_1 \tag{15}$$

$$F_3 = F_{CoM_{Rocker}} \left(1 - \frac{x_{CoM_{Rocker}}}{x_B}\right) \tag{16}$$

One major parameter derived from the normal force analysis provided in the equations above is the definition of the static stability angles of a specific configuration. A general approach to define the maximum static stability for an n-wheeled system is described by the loss of normal force on one wheel, although due to the articulated linkages between kinematic connections a loss of ground contact for a single wheel on sloped terrain can occur before a general tip over of the whole system takes place. As most suspension configurations do not describe a symmetric design a stability angle for uphill and downhill slopes need to be calculated respectively.

Table 2: Design variable input (exemplary)

	Design variables for geometric properties			
Body	Length of body	l_{Body}	0,8	m
	Height of body	h_{Body}	0,2	m
	Width of body	b_{Body}	0,5	m
Rocker	Length of rocker	l_r	0,6	m
	Height of rocker linkage	h_r	0,18	m
Bogie	Length of bogie	l_B	0,4	m
	Height of bogie linkage	h_B	0,1	m
	Position of articulation	l_{B1}	0,2	m
Wheel	Radius of wheel	r_W	0,1	m
	Width of wheel	b_W	0,1	m
	Design variables for CoM estimations			
Point mass	Mass of wheel	m_W	0,3	kg
	Mass of motor and gear	b_W	0,25	kg
	Mass of linkages	m_{link}	0,5	kg/m
Payload #1	Mass of design payload	m_{pl}	2,0	kg
	Longitudinal coordinate	x_{pl}	0,4	m
	Lateral coordinate	y_{pl}	0,35	m
	Altitudinal coordinate	z_{pl}	0,28	m
	Design variables for terrain simulation			
Slope	Inclination angle, wheel i	γ_i	20	°
Obstacle	Obstacle height, wheel i	h_i	0,2	m
Gravitation	Conditions on Earth	g	9,81	m/s^2

For setting up a reference model a symmetric configuration regarding wheel base and an equal wheel load distribution on flat terrain was chosen for a favourite design point suited for the defined requirements. Some exemplary input-variables for a specific chassis configuration are shown in Table 2. For a simplified design optimisation specific parameters can be put into iterative calculation loops. Due to the requirement of using commercial off the shelf components the mass estimations of the subsystem components, linkages etc. for the following static analyses are preliminary.

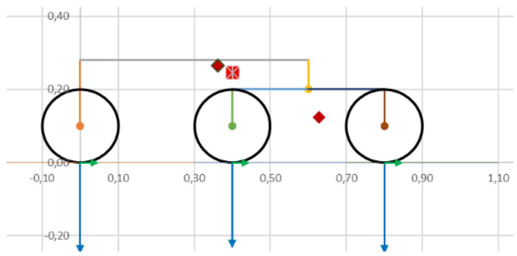

Figure 5: Visualised symmetric rocker-bogie design

Fig. 5 shows the visualised design output for a favorited kinematic configuration. Within the schematic diagram the respective forces resulting from the kinematic linkages and mass distribution on even terrain assuming a rigid surface are shown. The visualisation was implemented to demonstrate the impact of a reconfiguration by changing specific input parameters.

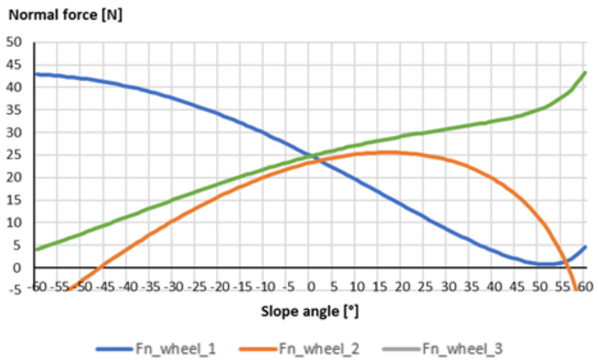

Figure 6: Normal force analyses for symmetric rocker-bogie configuration

As individual wheel contact angles are implemented the output for a representative normal force analyses in sloped terrain as shown in Fig. 6 can be calculated. Using these calculations allows to determine the static stability angles of a chosen configuration.

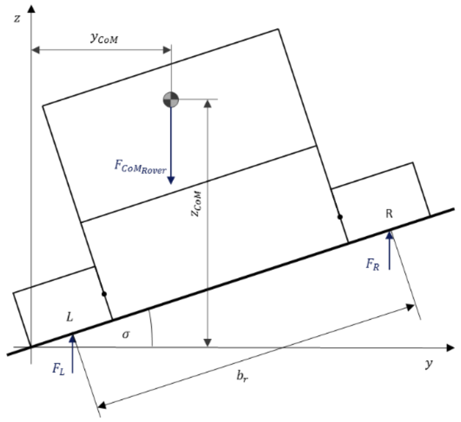

Figure 7: Left and right side stability analyses

The maximum lateral stability angles are calculated from left- and right side normal force analyses shown in Fig. 7. As kinematic effects of the suspension system in lateral direction can be neglected for a passive suspension chassis design, the maximum stability is therefore calculated with respect to the overall rover width as well as the CoM properties as formulated in Eqs. 17-19.

$$F_R b_r \cos\sigma = F_{CoM_{Rover}} y_{CoM} \quad (17)$$

$$F_R = \frac{F_{CoM_{Rover}} y_{CoM}}{b_r \cos\sigma} \quad (18)$$

$$F_L = F_{CoM_{Rover}} - F_R \quad (19)$$

Depending on the design variables the right and left side stability angles can be plotted as seen in Fig. 8 for an exemplary configuration.

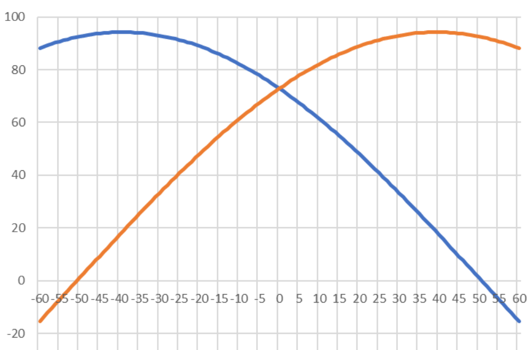

Figure 8: Normal force analyses result for calculation of lateral stability angle

3.2. Variability of Centre of Mass

In order to maintain a maximum of stability for varying chassis configurations, the influence of a changing centre of mass due to relocation or exchange of individual subsystems, as well as the implementation of additional payloads (camera mast, robotic arm, etc.) requires an iterative adjustment of the locomotion subsystem design. The implemented calculation model offers the possibility of a simple adaptation of rocker and bogie length and height, as well as the geometric dimensions of the connecting structure, so that favorited configurations can be analysed.

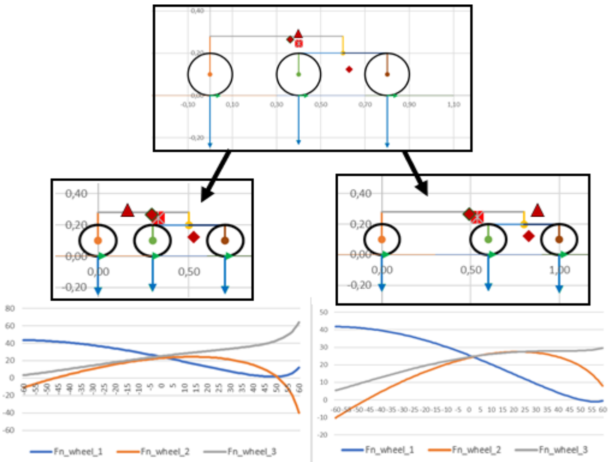

Figure 9: Impact of latitudinal CoM variations on locomotion system design for load equalisation

As shown in Fig. 9 an exemplary variation of the longitudinal coordinates of a specific payload mass (red triangle) can be counterbalanced due to length variations of the suspension systems properties, especially by adapting the rocker and bogie length respectively. By adapting the configuration an equalisation of the wheel load on flat surface, as well as representative static stability angles for up- and downward movement on sloped terrain are realised.

A shift of the CoM within its altitude is a critical aspect in relation to the maximum slope gradability and climbing capability. A combined variation of longitudinal and altitudinal CoM variation results in an overall redesign of the configuration properties in order to maintain the static stability angle in a reasonable range. Being able to reconfigure major design properties is required to sustain equal load distribution on all wheels for changing chassis configurations out of varying mission profiles. As shown in Fig. 10 the respective rocker and bogie length need to be reconfigured in order to optimise locomotion performance parameters for an implemented payload. The generated output for individual configurations can be analysed in order to define a simplified model of the overall chassis configuration.

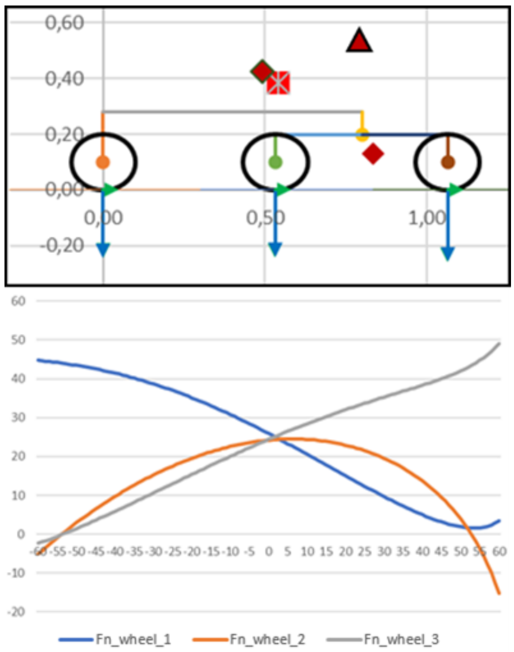

Figure 10: Impact of altitudinal CoM variations on locomotion system design for load equalisation

As each reconfiguration results in changing mass for the locomotion system components the model can be further utilised to estimate and compare the mass variations for changing configurations.

3.3. Obstacle climbing performance

The requirements for climbing an obstacle with a specific height (h_s) are depending on the geometric constraints affected by the wheel size and dimensions of the rocker-bogie configuration, as well as the friction requirements derived from the individual wheel loads.

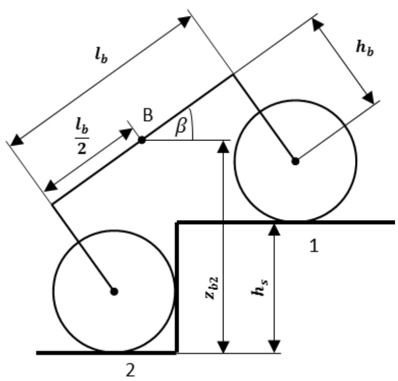

Figure 11: Geometric requirements of a bogie climbing a vertical obstacle

The geometric requirement for a bogie climbing a vertical obstacle as shown in Fig. 11 depends on the properties of the bogie length, wheel radius and obstacle height and can be formulated in Eq. 20.

$$z_{b2} = \frac{l_b}{2} \sin\beta + h_b \cos\beta + r_W \tag{20}$$

As a inclination angle (β) of 45° is most critical for interference for the relation of the design variables follows:

$$\frac{l_b}{2} \sin\beta + h_b \cos\beta + r_W > h_s \tag{21}$$

Besides the geometric constraints while climbing an obstacle higher than the wheel radius, a certain amount of friction needs to be maintained at each individual wheel. There have been plenty of investigations comparing the friction requirements for different chassis suspension systems in order to define performance characteristics of a specific configuration [5]. As the simulation effort of analyses and determination of climbing performances of a rover in soft soil is quite high, the friction requirements shall be derived for solid surfaces for static conditions within the developed model.

Due to the kinematic linkages of a six wheeled rocker-bogie concept, the friction requirements for each single wheel vary depending on the individual wheel load. Therefore, the static conditions for the front, middle and rear wheel need to be calculated separately as shown in Fig. 12.

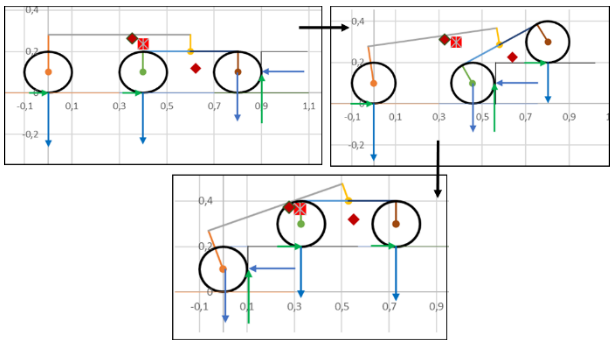

Figure 12: Visualisation of a respective wheel climbing a vertical obstacle

For the analyses the requirements need to be distinguished between forward and backward movement. In order to calculate the progression of the required friction with a changing height of the climbing wheel, for all static analyses it was distinguished between the wheel at the beginning of the step and at the final step height.

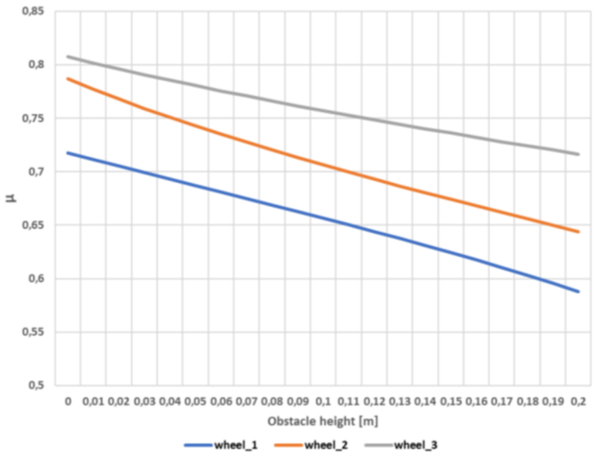

Figure 13: Required friction for each wheel (obstacle height: $h = 0{,}2\ m$)

The friction requirements were basically derived from the shifting wheel loads resulting from the kinematic impact of a constantly changing centre of mass during climbing. To enable a specific wheel to climb a vertical obstacle, the friction force between wheels and surface need to overcome the specific wheel load of the climbing wheel. The relation between friction force and normal force component implemented in the model is calculated as shown in Eq. 22. The respective F_{n_i} are derived from the static calculation model.

$$F_{r_i} = F_{n_i}\mu \tag{22}$$

As shown in Fig. 13, the required friction is varying for each wheel respectively and therefore highly dependent on the individual wheel loads of a specific chassis configuration. Although the model is quite simplified in a two dimensional plane, and especially the effect of grousers on the wheel interaction with the obstacle surface are not respected in the calculation, friction requirements for static wheel load conditions can be derived.

3.4. Drive unit torque definition

From the calculated forces derived of the static analyses the definition of the required wheel torque to enable mobile capabilities needs to be calculated in order to design requirements for the drive unit and an ongoing component selection. To enable forward locomotion on loose soil the defined motion resistance of the soil needs to be overcome by the drive units' torque [4]. The maximum torque needed for a single drive unit (T_{drw}) depends on the respective wheel size as shown in Eq. 23 (applicable for rigid wheels).

$$T_{drw} = \left(\sum R\right) r_W \tag{23}$$

The required torque for climbing a vertical obstacle is derived from the required friction force [5] shown in Eq. 24.

$$T_{obs} = \mu F_n r_W \tag{24}$$

As the rover shall be able to drive in sloped terrain and changing surface conditions, the required torque for various load cases was analysed as shown in Table 3. In order to determine the most demanding torque requirement, the loads for each single wheel torque need to be distinguished.

Table 3: Required drive unit torque (exemplary)

Solid surface ($\mu = 0{,}8$), 20° slope	T_{drw_ss20}	1,145	Nm
Loose soil (dry sand), 0° slope	T_{drw_ds}	0,436	Nm
Loose soil (dry sand), 20° slope	T_{drw_ds20}	1,647	Nm
Obstacle climbing ($h = 0{,}2\ m$)	T_{obs_02}	2,471	Nm

The calculated drive torques are highly dependent of the specific chassis configuration, for operating on loose soil especially the wheel proportions are one main driver for the drive unit design.

4. CHASSIS CONCEPT DESIGN

Within the previous sections the derivation of fundamental design parameters for the locomotion system in order to realise a functional chassis have been described. The static analyses model is used for the definition of individual chassis configurations, which are built up at the robotic laboratory at the University of Stuttgart for further analyses.

4.1. Reconfiguration concept design and component selection

Major ongoing investigations for implementing the reconfigurable chassis are compared. To enable an efficient adaptation of the chassis to a chosen configuration, especially the locomotion system subcomponents are designed as modular units. As passive articulation for all suspension systems shall be realised, all rigid components are designed as Aluminium profiles. Required subsystems and additional payloads can be attached via struts to the structural frame of the body, and progressively relocated. The structural components of the locomotion system (linkages etc.) are also realised as rigid Aluminium profiles. For reconfiguration and to be conform with the calculated driving locomotion system parameters, all corresponding profile struts are designed interchangeable. Replacing only desired structural components enables a cost effective and straightforward possibility of implementing a highly variable system. The drive units connected to each wheel shall be applicable as integral systems to the locomotion system structure. For optimising the locomotion performance of a specific configuration exchangeable DC-Motor and gearhead combinations commercially available are implemented.

4.2. Application concepts

With realising a functional breadboard chassis from the derived design parameters the system shall be used for comprehensive verification of simulation analyses model. In order to verify specific locomotion and performance parameters for varying chassis configurations, the implementation of additional sensory elements, as well as the design of a representative test environment is a major part of ongoing investigations. The set-up of a rover testbed allows to simulate changing terrain conditions (uphill and downhill slopes), surface variations (loose soil), and obstacle traverse capabilities of specific chassis configurations, and therefore an in-depth verification of individual performance parameters. Within the verification process the optimisation of these parameters is intended to be realised.

Another prospective objective is the realisation and ongoing development of the rover control metrics, which shall successively be implemented to enable advanced mobile capabilities of the system. Furthermore, with the functional chassis the concept of a universal research platform will be further investigated. As a promising outlook the application field of visual data capture and processing for navigation and advanced autonomous capabilities is to be mentioned. Therefore, interfaculty cooperation projects are intended at the University of Stuttgart as well as cooperation with industrial partners.

5. CONCLUSION

This paper presents the analytic method that was used to design a reconfigurable locomotion suspension system. By being able to maintain a high degree of adaptability of driving design parameters the developed chassis shall be used as a multifunctional platform to verify developed simulation models as well as to analyse varying rover mission design concepts. The requirement to reconfigure major locomotion system parameters lead to a generic approach for calculating the fundamental locomotion metrics. A calculation model was developed, in which major parameters of ground interaction for changing wheel-soil conditions, as well as a model of a simplified two dimensional kinematic force analyses for quasi-static conditions was implemented. Besides the realisation of a simplified visualisation the output of the model can be used to determine the required parameter to reconfigure the chassis for a changing mission profile, especially in terms of interchanging subsystems and additional payloads. Ongoing developments at the University of Stuttgart for the practical implementation of the calculated chassis design parameters to verify the simulation model and to optimise the locomotion performance are investigated.

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