

Power Consumption Analysis of a Flexible-Wheel Suspension Planetary Rover Operating upon Deformable Terrain

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ABSTRACT

This study analyzes the power consumption of a specific Planetary Exploration Vehicle (PEV) subsystem known as Flexible-Wheel (FW) suspension, more specifically the interaction between a FW and the deformable terrain upon which it traverses. To achieve this a systematic and analytical calculation procedure has been developed, which culminates in the definition of three dimensionless properties to capture the FW-soil interaction. Aimed towards the design engineer participating in concept evaluation, and the control engineer conducting initial analyses, this study has found that the resistance coefficient for the interaction between a FW and the deformable terrain can, in general, be several orders of magnitude higher than the rolling resistance of a pneumatic tire operating upon rigid terrain. The sensitivity analyses also unveiled certain design parameters which have much more effect on the resistance coefficient than others, enabling the engineer to concentrate their efforts more effectively to achieve greater design optimization.

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INTRODUCTION

Humanity's desire to understand the universe is a trait that has been unwavering for millennia [1]; the allure of exploring distant celestial bodies by pushing both human knowledge and scientific technology to their limits is something that many have been unable to resist. This fascination has been, and still is, an important catalyst in the development of the human race and has recently been focused on Earth's neighbors: Mars and the Moon. One useful category of equipment that has been employed to conduct this exploration is the unmanned Planetary Exploration Vehicle. Over the past twenty years several variants of PEV have come into and out of commission, with a noticeable trend being an increase in scientific capabilities and therefore vehicle mass. A good example of this trend is provided by the three generations of NASA rovers with Sojourner, Spirit and Curiosity weighing 10.5kg, 176.5kg and 899kg respectively [2-3]. Couple this with a desire to increase vehicle velocity, to enable more missions to be completed more quickly, and the result is a significant demand on power management.

One method of reducing power consumption, that is becoming favorable in current PEV design, is through the use of Flexible-Wheels. This novel and innovative concept is based around providing wheel deformation through the

integration of mechanical suspension members rather than an air filled rubber carcass seen on pneumatic tires [4]. For the air-filled tire equipped vehicle, the tire model is the most critical and difficult module in a full vehicle system model due to its complexity of nonlinear characteristics and interaction with terrain profile. Over the years, many tire models have been developed for vehicle dynamics and road loads simulation [12]. Wheel loads commonly measured by Wheel force transducers (WFT) are used in computer modeling analysis [12] and road test simulation [13, 14, 15]. An illustration of a typical flexible-wheel design is shown in Figure 1 [5]. This technology promises many significant benefits including improved packaging, traction, road friendliness and fuel economy [4, 16]. For these reasons this technology has also become very attractive for commercial applications.

To improve our understanding of the FW, and therefore allow for better design and power management to be achieved, it has been important to establish accurate and effective analytical modeling techniques. One such technique is a generalized model developed in [4], which models the flexible suspension members as spring-dampers. The technique is successful in deriving equations for power consumption, stiffness and damping, alongside three dimensionless properties which represent important

characteristics of the FW. However, the work is limited in the sense that it fails to include a deformable soil model, a consideration which cannot be neglected due to this being the only operating surface a PEV will typically encounter. A good example of a deformable soil model that is suited to an application such as that pursued in this paper is the semi-empirical calculation procedure derived by Wong and Reece in [6].



Figure 1. The Michelin® Tweel™, a commercial example of a flexible wheel [5]

The focus of this study is to therefore overcome the limitations experienced in [4] by incorporating a deformable soil model. To achieve this, a new calculation procedure has been derived, resulting in three dimensionless properties suitable for subjection to a sensitivity analysis. The structure of this paper therefore follows this development process. Initiating with an explanation and derivation of the calculation procedure and dimensionless properties, this paper then progresses to a statement of the nominal parameters used and graphical results of the sensitivity analysis before finishing on the conclusions of the work.

DERIVATION OF THE MODEL

The deformable soil model chosen to collaborate with the generalized FW model of Cao in [4] is an extension to the pressure-sinkage relationship derived by Wong and Reece in [6]. Figure 2 illustrates this combination, where a number of spring-damper units are connected to a central connection point. The units are evenly spaced around a rigid wheel rim that rolls upon a deformable terrain which offers stress in both radial and tangential directions. More specific details of the models will be provided in the forthcoming text, however, there are several main constraints that the reader should be aware of:

1. The techniques account for the single passage of a wheel over the terrain only.
2. The motion of the wheel is restricted to that of straight line travel.
3. Motion is over horizontal terrain only (i.e. resistance to slopes and obstacles is not taken into consideration).
4. The wheel is modeled as driven, and not towed.

5. The model is accurate for deflections of the central connection point that are $\leq 10\%$ of in the X-direction and $\leq 40\%$ of in the Y-direction.

where R is the Radius of the FW.

The Pressure-Sinkage Relationship

This fundamental relationship in terramechanics was first introduced in the 1960's by M.G. Bekker in [7] and later developed to greater degree of sophistication by Wong and Reece in [6]. The techniques center around the proposal that radial stress at a wheel-soil interface is proportional to the magnitude of wheel sinkage, for a given type of soil and size of contact area. The improvements made by Wong and Reece enabled the pressure-sinkage relationship to become applicable to a *driven* wheel, rather than just a towed wheel, by taking into account tangential as well as radial stresses. This is a much more accurate representation of the true failure mechanism of soil as discovered in [8] and can be argued to be of greater value to the PEV community as typically all wheels are driven.

Radial stress (σ)

It was found in [6] that the magnitude of radial stress at the wheel-soil interface is a function of wheel slip and varies over the length of the contact patch. More specifically the point of maximum radial stress exists at an intersection between two soil flows underneath the wheel, to the right of point "a" in Figure 2. Under the original pressure-sinkage relationship this simply did not occur as the point of maximum radial stress was deemed to be at the point of maximum sinkage i.e. exactly on point "a" of Figure 2. This new understanding therefore required manipulation of the pressure-sinkage relationship to improve accuracy, a manipulation which began with the definition of the angular position of maximum radial stress as seen in Eq.(1) [6] and Figure 3

$$\theta_M = (c_1 + c_2 i) \theta_1 \quad (1)$$

where θ_M is the angular position of maximum radial stress, c_1 and c_2 are soil property coefficients, i is the magnitude of slip and θ_1 is the entry angle of the wheel into the soil.

Slip can be specified by the engineer or calculated using Eq.(2) [9].

$$i = 1 - \frac{V}{R\omega} \quad (2)$$

where V is the forward velocity of the PEV, R is the radius of the FW and ω is the angular velocity of the wheel.

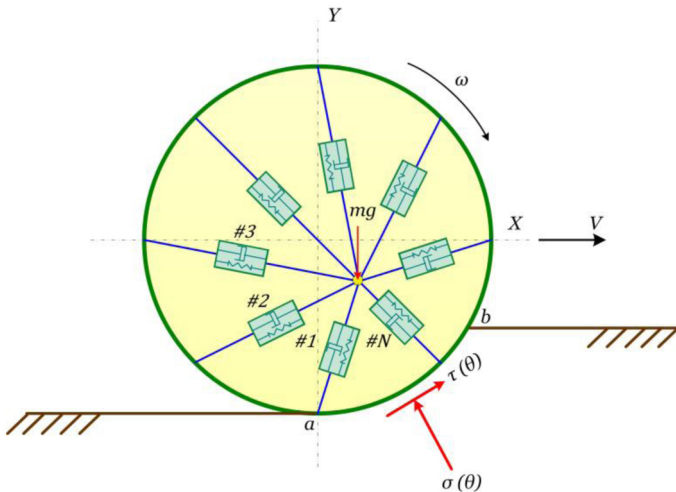


Figure 2. The Flexible-Wheel Deformable Soil Interaction Model

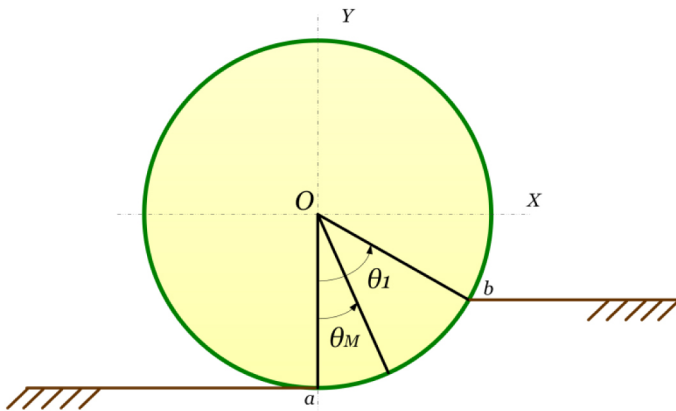


Figure 3. Simplified Wheel-Soil Interaction Diagram Illustrating the Angular Position of Maximum Radial Stress and Entry Angle

The recognition of the maximum radial stress point along the wheel soil interface prompted the distribution to be considered as two separate sections, these being front and rear which spanned from the point of maximum radial stress to the entry and exit points of the wheel into/from the soil, respectively. It was useful to split the distribution up into front and rear sections as it allowed individual equations to be created that more accurately represented the distribution. Wong recognized through experimentation in [6] that the front and rear distributions could be successfully represented by the pressure-sinkage relationship, leading to the creation of Eq's. (3), (4) and (5) [6]. Equations which enabled the radial stress to be determined at any angular θ position along the wheel soil interface.

$$Q = (k_c + k_\phi b) \left(\frac{R}{b} \right)^n \quad (3)$$

$$\sigma_1(\theta) = Q (\cos\theta - \cos\theta_1)^n \quad (4)$$

$$\sigma_2(\theta) = Q \left[\cos \left(\theta_1 - \frac{\theta}{\theta_M} (\theta_1 - \theta_M) \right) - \cos\theta_1 \right]^n \quad (5)$$

where σ_1 is the radial stress distribution over the front section of the contact patch, σ_2 is the radial stress distribution over the rear section of the contact patch k_c and k_ϕ are pressure-sinkage parameters, n is an exponent of sinkage and b is the smaller dimension of the projected contact patch. An illustration of this contact patch is shown here in Figure 4 as the shaded area.

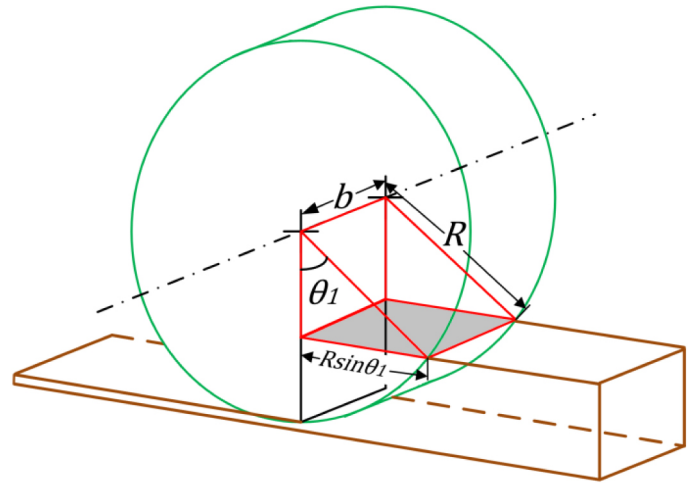


Figure 4. Illustration of the Projected Contact Patch

Tangential stress (τ)

The tangential stress, otherwise known as shear stress, can be defined as the stress which acts perpendicularly to the radial stress and it is essential to take account of when modeling a driven wheel, as its presence provides opposition to the thrust force of the wheel, and therefore ensures equilibrium.

The tangential stress, at any angular position of interest θ can be calculated using the well-established, simplified Bekker equation shown here as Eq.(6) [6]. An equation which states that shear stress is a function of shear deformation (displacement) of the soil and radial stress, among other factors.

$$\tau_{1,2}(\theta) = (c + \sigma_{1,2}(\theta) \tan\phi) (1 - e^{-j/K}) \quad (6)$$

where τ_1 is the tangential stress distribution over the front section of the contact patch, τ_2 is the tangential stress distribution over the rear section of the contact patch, c is the soil cohesion, ϕ is the angle of internal shearing resistance of the soil, K is the shear deformation parameter and j is shear deformation. Shear deformation (j) is dependent on the slip