

Mobility and Energy Efficiency Analysis of a Terrain Truck

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ABSTRACT

While much research has focused on improving terrain mobility, energy and fuel efficiency of terrain trucks, only a limited amount of investigation has gone into analysis of power distribution between the driving wheels. Distribution of power among the driving wheels has been shown to have a significant effect on vehicle operating characteristics for a given set of operating conditions and total power supplied to the wheels. Wheel power distribution is largely a function of the design of the driveline power dividing units (PDUs).

In this paper, 6×6/6×4 terrain truck models are analyzed with the focus on various combinations of PDUs and suspension systems. While these models were found to have some common features, they demonstrate several different approaches to driveline system design.

In order to further investigate the effect of wheel power split on mobility and energy/fuel efficiency in conjunction with suspension characteristics, this paper provides an analytical method and mathematical model for the entire truck, including driveline system (sets of PDUs), suspension, and tires under various typical operating conditions. Interactions between longitudinal and normal truck dynamics are first modeled and their influence on vehicle mobility and energy efficiency is analyzed. A computational algorithm is presented, which integrates equations for the driveline system with the suspension model (i.e., wheel normal reactions), along with a method for solving the integrated equations. Finally, an analysis of the effects of PDU combinations on truck mobility and energy efficiency is demonstrated.

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INTRODUCTION

While a lot of research has gone into improving mobility, energy efficiency and fuel consumption of terrain trucks by studying suspension-tire-soil transient and steady-state interaction and traction capabilities, engine-transmission configurations, various types of controls of traction performance and vehicle positioning, etc. [1,2,3,4,5,6,7,8], not much investigation has been done on the effects of the power distribution among the drive wheels. A technical literature analysis shows that less attention is devoted to wheel driving force analysis for terrain vehicles. As an example, paper [9] can be mentioned that presents such an analysis for an eight wheel vehicle; driveline systems in general are considered in [10], which covers theory and design issues, provides test results from actual production

vehicles to validate the theories, as well as a methodology for creating control algorithms and tests. Paper [11] presents an algorithm that controls wheel torques for a 4×4 truck such that wheel slippages are eliminated; it uses wheel normal loads and tire inflation pressure as inputs. The interactions between brake-based slip control and a limited-slip differential are investigated in [12].

Therefore, there is a need for an analytical method to concurrently analyze energy efficiency and mobility of a 6-wheel terrain truck as an effect of wheel power distribution in conjunction with normal and longitudinal dynamics of the vehicle. The latter allows a researcher to incorporate tire-suspension characteristics in the vehicle model; moreover, it allows to study suspension-driveline interaction and combined influence on vehicle normal and longitudinal

dynamics and thus on vehicle energy efficiency and mobility. This makes the subject-matter of the paper.

The paper presents an analytical method along with math models of the entire vehicle, including driveline, suspension, and tires. An algorithm is given which integrates equations for the driveline with the suspension model and provides a method for solving the integrated equations. Next a computer model (computational algorithm) of these equations is shown and analysis done on the effects of several driveline system configurations on energy efficiency and mobility.

MANUFACTURER MODEL ANALYSIS

A survey of driveline configurations was performed of major world manufactures of articulated trucks. The survey showed that all models allow all wheels to be driven: the wheel formulae of the trucks are either 6×6 or 4×4. Several companies allow for disengagement of one axle, that is change the wheel formula from 6×6 to 6×4. All 6-wheeled trucks have the two rear axles in tandem.

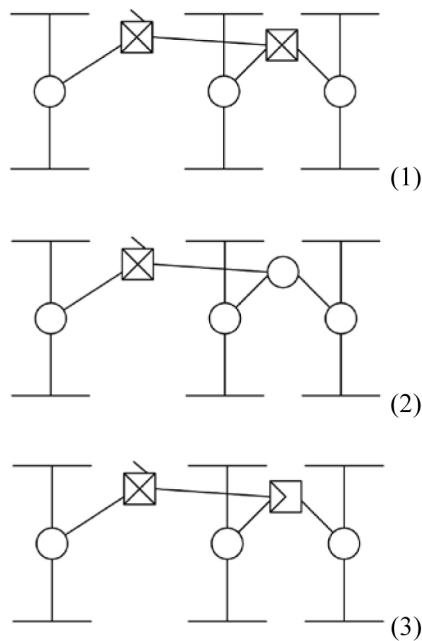


Fig. 1. Three Driveline System Configurations

Various manufacturers employ symmetrical open differentials and limited slip differentials as power dividing units between the left and right wheels. An asymmetrical locking differential and a positive engagement (a fully locked drive system) are used in the transfer case between the front axle and the rear axle tandem. The majority of 6×6 models have a symmetrical locking differential between axles 2 and 3. A few companies allow for disengagement of the rear axle of the tandem. One company has rear axle disengagement on its 4×4 models. Based on the analysis, three potential configurations of the driveline system of a 6×6/6×4 truck were included in consideration in this paper. The

configurations accepted for this study are given in Fig.1. Table 1 presents the designations for the power dividing units.

Table 1. Symbols of Mechanisms in Power Dividing Units

Mechanism	Designation
Symmetrical open (free-running) differential	
Constantly locking engagement of the output shafts	
Non-constant engagement of the output shafts with manual disengagement of one shaft	

TIRE / TRUCK MODELING

In order to evaluate the truck's energy efficiency and mobility, it is necessary to understand the forces and reactions exerted on the wheels in the driving mode during normal operation (see Fig. 2). Here the applied wheel torque T_w develops circumferential force F_x ; the useful portion of this force $F_{frame,x}$ is reduced by the value of the rolling resistance R_x resulting from tire-soil interaction.

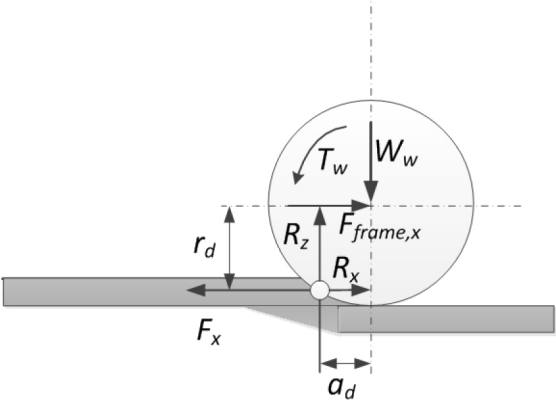


Fig. 2. Free Body Diagram of Wheel in Driving Mode

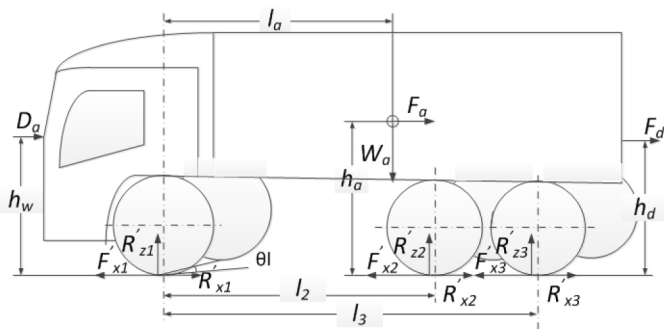


Fig. 3. Truck Free-Body Diagram

The free-body diagram of the truck in straight line motion in Fig. 3 demonstrates the vehicle total wheel circumferential

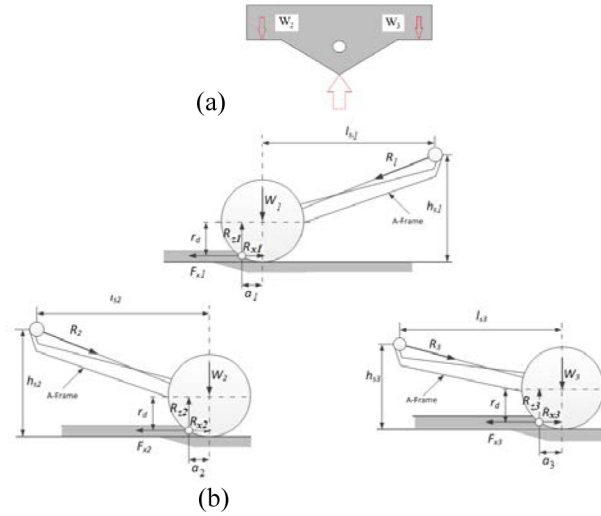


Fig. 4. Bogie diagram (a) and A-frame suspension geometry

force (force in the longitudinal direction) that is equal to the sum of the net longitudinal wheel forces:

$$F_{x\Sigma} = \sum_{i=1}^3 (F_{xi}^* + F_{xi}') \quad (1)$$

Total force $F_{x\Sigma}$ represents the total tractive effort. As shown in Fig.3, it consists of several components:

$$F_{x\Sigma} = \sum_{i=1}^3 R_{xi}^{(c)} \pm W_a \sin \theta_n + \frac{a_x W_a}{g} \delta_r + D_a + F_d \quad (2)$$

W_a is the gross weight of the vehicle; the grade resistance is represented by $W_a \sin \theta_n$, where θ_n is the assigned grade angle and sign is + for uphill, - for downhill motion; the inertia force F_a includes the mass in translational motion, W_a/g , the longitudinal acceleration a_x , the gravitational constant g , and δ_r that is the known mass factor for rotating inertia masses of the truck; F_d is the drawbar load (which in the case of the truck is 0). Aerodynamic drag D_a is only significant above 50 km/h. As top truck speed is approximately 36 km/h, the aero drag is not necessary to consider.

In this paper, an inverse dynamics approach is proposed to study the truck's motion. This means that the program of motion of the truck is given, i.e., the longitudinal acceleration and velocity are known at every moment of motion. Such an approach presents two advantages: (i) the truck's motion is modeled the same for all the driveline configurations, and (ii) the number of unknowns in Eq. (2) is reduced.

Eq. (2) can be solved for the total circumferential force $F_{x\Sigma}$ if the rolling resistance forces are first computed. The latter is possible if the tire normal reactions are known. Therefore, the normal dynamics of the truck that includes suspension consideration is to be discussed next.

Truck Normal Dynamics

Individual tire normal reactions are determined based on vehicle gross weight W_a . The majority of the weight is handled by the two rear axles, and under static conditions is distributed evenly between axles 2 and 3 by the bogie: $W_2 = W_3$ (see Fig. 4a).

The suspension incorporates an A-frame with spherical rubber bushing for each axle (Fig. 4b). The rear two axles (tandem) are mounted on a bogie on each side of the vehicle. This means the suspension is not independent (the wheels on an axle do not react to vertical deflection independently of each other); a special set of equations was derived to describe the axle normal reactions that include suspension geometry:

$$\begin{aligned} R_{z1} &= W_a \cos \theta_n - \frac{2A}{l_2 + l_3} - (F_{x1} - R_{x1}) \frac{h_{s1}}{l_{s1}} \\ R_{z2} &= \frac{Al_{s3} + Bl_3}{l_2 l_{s3} + l_3 l_{s2}} \\ R_{z3} &= \frac{Al_{s2} - Bl_2}{l_2 l_{s3} + l_3 l_{s2}} \end{aligned} \quad (3)$$

$$\begin{aligned} A &= W_a l_a \cos \theta_n \pm W_a h_a \sin \theta_n + F_a h_a \\ B &= A \frac{l_{s2} - l_{s3}}{l_2 + l_3} + (F_{x2} - R_{x2}) h_{s2} - (F_{x3} - R_{x3}) h_{s3} \end{aligned}$$

Next calculate individual tire reactions $R_{zi}^{(c)}$ at each i -axle: