Six by Six Terrain Vehicle for Optimal Mass, Geometric Configuration and Tractive Efficiency

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Abstract

This paper presents a new design methodology for determining the optimal mass, geometric configuration and wheel power distribution of a 6x6 terrain vehicle, in order to provide the vehicle with optimized traction. The educational aspect of this paper is to motivate and integrate student learning process through mathematical models and equations for design.

The methodology is realized using a mathematical model of a 6x6 wheel drive tractor satisfying the off-road profile. The methodology is used to find the optimum co-ordinates for the center of gravity and optimum arrangements of the middle axle wheels along the tractor. The optimal tractive efficiency is then achieved by optimizing the geometric coordinates.

Educational value of the paper

This paper is basically an analysis-based learning approach to motivate and integrate student learning process. This approach will increase student’s motivation because it introduces the engineering content early in the curriculum and helps them understand the concepts practically.

This analysis is initially performed as an independent study at a graduate curriculum. A graduate student worked upon this lesson-study allowed him to better understand the concepts like moment of inertia, damping and center of gravity in real-time applications. So, this would definitely be helpful for undergraduate students taking mechanical vibration, dynamics and vehicle design classes. The students could study the location of various components and understand how the selection of spring-stiffness, damping, axle coefficients, and distance between axles could have impact on the performance characteristics of the vehicle. This work would definitely serve as a guideline for students interested in doing vehicle design projects. The work could be used as a reference for the students of the Mini-Baja and Formula-SAE projects in the Mechanical Engineering Department.

This paper delivers a mathematical model with analytical equations that provides a various options for designing acceptable vehicle models. Further research and simulations need to be performed for a detailed study. This approach will give undergraduate and graduate engineering students better scope on designing concepts and they will have more knowledge on the fundamentals. This approach will also allow students to: i) tolerate ambiguity that shows up in viewing design as inquiry or as an iterative loop of divergent-convergent thinking, ii) maintain sight of the big picture by including systems thinking and systems design, iii) handle uncertainty, iv) make decisions, v) think as part of a team in a social process, and vi) think and communicate in the several languages of design.

As far as the educational value of the paper is concerned, the students get to know more about the modeling of a mechanical system. In this study engineering students will come to know how to frame the equations of motion for six wheel drive terrain vehicles and other vehicles with similar characteristics. The equations for various forces acting on the system, stiffness, and moments of inertia could be realized by the students. This approach will give them a practical view of solving a vibration and dynamics oriented problem. Doing so will help them understand better and solve various problems involved in engineering design. The paper will also teach the students how choosing the values for stiffness and framing the equations of motion could affect the stability of the system.

Introduction

Many definitions are given in the automobile literature for the off road multiple wheel drive vehicles [6, 7, 8]. Basically, the performance of off road vehicles is defined as the property that enables a vehicle to move successfully and reliably on varying (macro and micro) road conditions by overcoming various obstacles. A 6 wheel drive off -road vehicle's performance may be improved by controlling individual wheel torques [1]. Individual control of wheel torques such as traction control and anti-lock braking improve vehicle handling and stability in cornering. Multidisciplinary Design Optimisation (MDO) is a very useful technique for solving any multi-body dynamics problem [20]. MDO is a body of methods and techniques for performing the optimization so as to balance the design considerations at the system and detail levels. Another approach to improving vehicle performance is to integrate both the vehicle operational properties and adaptive vehicle dynamics [2, 3, 4]. However, analysis of mass and geometric parameters may provide a basis needed to better understand vehicle performance.

Several parameters which affect the fuel consumption, performance, and top speed of a vehicle are vehicle weight, tire rolling resistance, aerodynamic drag and driveline configuration [28]. Studies have indicated that it is possible to change each of these parameters to reduce the fuel consumption of a vehicle. The basic mass and geometric parameters of all-wheel drive off-road vehicles, such as coordinates of the center of gravity and wheel arrangements, are usually achieved to reduce the dynamic normal loads on the running gear system while satisfying the off road profile [5].

The geometric vehicle characteristics such as front and rear trafficability angles, and the longitudinal and cross radii of trafficability should satisfy the trafficability requirements. However less attention is given for studying the influence of the center of gravity placement and wheel arrangement on traction and velocity properties for basic off road mobility of vehicles. Therefore, off road mobility may be understood as the ability of the vehicle to move outside of roads with or without load [9]. Furthermore, basic off road mobility is a complex vehicle performance that interconnects traction, velocity, and surface grip properties of vehicles. These properties are fundamental to estimating basic on/off road mobility. For tractors, traction and velocity properties together provide adequate drawbar to pull various implements and machinery.

Heavy vehicles with high centers of gravity are prone to rollover accidents. For a given height and width, stability is affected by suspension design. In articulated vehicles, the matching of the tractor and trailer suspensions, as well as the degree of coupling between tractor and trailer, affects the stability of the combination. The effects of suspensions, couplings, tires and chassis on the rollover limit are also to be considered [25]. The traction and velocity properties of vehicles are characterized by their ability to move under an action of the circumferential forces exerted on the drive wheels [5, 6]. These properties are usually examined with fuel consumption. Taking the generalized parameters for traction as the average vehicle speed, V_{mid} and the average fuel consumption, Q_{mid} we may determine the vehicle's productivity, *SP*, as follows [4]:

$$SP = G_l V_{mid} / (g Q_{mid}) \tag{1}$$

where G_i is the weight of the handling load (payload) and g is the gravitational acceleration.

The vehicle's fuel consumption per unit of productivity, SP_o is defined [11] as:

$$SP_0 = 1/SP \tag{2}$$

Understanding the effects of tire and vehicle properties on the rollover propensity of tractor semi-trailer trucks is essential. A simplified computational tool can be used to understand and predict the effects of various tire characteristics and truck design parameters on rollover under steady cornering and non-tripped conditions [29]. The traction and grip properties define the interaction between the drive wheels and a surface on which the tractor moves. These properties are typical in the tractor engineering and provide a measure for the tractor's ability to tow trailers and to work with agricultural machines. The tractor transport efficiency, η_{tr} provides the overall efficiency of a tractor and may be defined as:

$$\eta_{tr} = G_l V_r / N_e, \tag{3}$$

Where N_ε the power is input to the transmission and V_r is the actual forward speed of the vehicle. The transport efficiency is a product of three components [8]: The lift/drag ratio C_{ld} (or the ratio of the total vehicle weight to the total motion resistance), structural efficiency η_{st} (or the ratio of the payload to the vehicle total weight), and propulsive efficiency η_p.

Equations (1) and (3) show the structure of the specific productivity is similar to the efficiency coefficient defined with Wong's transformations [8]. The Rolling resistance is a very important factor to be considered in obtaining increased tractive forces [30]. In estimating the tractive efficiency, and transport efficiency of tractors in basic off road mobility, we will be taking the advantage of the running gear efficiency. General mathematical formulas to account for running gear efficiency as a function of transport and tractive efficiency are provided in [10]. Therefore, in view of these dependencies, we generate a formula to calculate the tractive efficiency of a 6 wheel drive tractor moving on a horizontal soft terrain with a drawbar pull.

In this paper we develop a new design based methodology, subject to optimization of the tractive efficiency constraint function, for determining the optimal mass and geometric parameters that provide a vehicle with the optimized traction and velocity properties for off road mobility. The method uses a mathematical model of a six-wheel-drive tractor on soft terrain. The underlying idea is that the optimal parameters provide the tractor with optimized tractive efficiency on its running gear system.

Methodology: One sixth tractor model

A common representation of the interaction between a vehicle system and the road surface in the vertical direction is the two degree-of-freedom system shown in Fig.1.

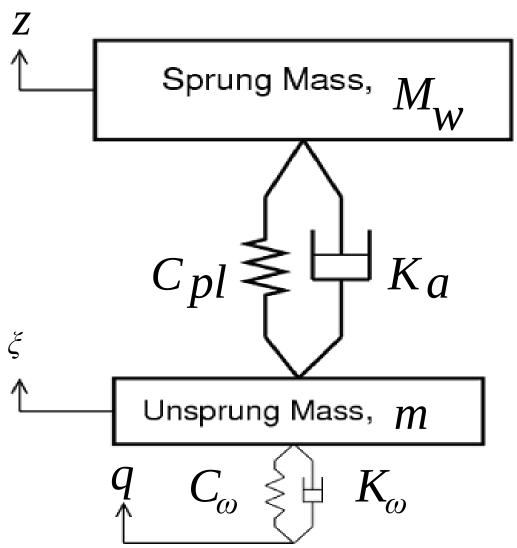


Fig. 1. One - sixth tractor model

This basic model is used in a number of different circumstances to describe the behavior of a vehicle suspension system. Mathematically, the relationships embodied in the model are given in Eq. (4).

$$\begin{aligned} &M_w \cdot \ddot{z} = C_{pl}(\xi - z) + K_a(\dot{\xi} - \dot{z}) \\ &m \cdot \ddot{\xi} = C_\varpi(q - \xi) + K_\varpi(\dot{q} - \dot{\xi}) - C_{pl}(\xi - z) \\ &-K_a(\dot{\xi} - \dot{z}) \end{aligned} \tag{4}$$

Where M_w is the one sixth tractor model effective sprung mass. This “one-sixth tractor” model can be implemented readily in a variety of simulation packages such as Matlab Simulink. Road input that matches the natural frequencies of the vehicle will cause suspension deflection which exceeds the magnitude of the input. The one-sixth tractor model is characterized by two natural frequencies, one for each mass in the system. The natural frequencies of the sprung and unsprung masses can be approximated with Eq. (5) and Eq. (6).

$$f_{n-sprung} = \frac{1}{2\pi}\sqrt{\frac{C_{pl}C_\varpi/(C_{pl}+C_\varpi)}{M_w}} \tag{5}$$

$$f_{n-unsprung} = \frac{1}{2\pi}\sqrt{\frac{C_{pl}+C_\varpi}{m}} \tag{6}$$

The one-sixth tractor model provides a stepping stone to a more complex model incorporating vehicle pitch and roll. A representation of a six wheel drive full tractor model derived for multi axles is shown in Fig. 2.

Full tractor model

A schematic of the tractor is shown in Fig.2. The tractive efficiency of the tractor in a running gear can be determined as follows:

$$\eta_x^t = N_{xp}/N_x \, , \tag{7}$$

where N_{xp} is the drawbar power corresponding to the drawbar pull (F_d), and N_x is the power delivered to the drive wheels. Powers N_{xp} and N_x are equal to, respectively,

$$N_{xp} = F_d V_r \tag{8a}$$

$$N_x = \sum_{i=1}^{3} F_{xi} V_{ti} = V_r \sum_{i=1}^{3} F_{xi} /(1 - s_{\delta ai}) \tag{8b}$$

where V_{ti} is the theoretical wheel speed of an axle with index *i* (*i*=1,3), F_{xi} is the tractive force of the i^{th} wheel and $S_{\delta ai}$ is the wheel slip coefficient of the i^{th} axle. Thus, the tractive efficiency is determined as follows:

$$\eta_x^t = F_d /(\sum_{i=1}^{3} F_{xi} /(1 - s_{\delta ai})) \tag{9}$$

To calculate the tractive efficiency, we need a function between the tractive force, F_{xi} and the wheel slip coefficient $S_{\delta ai}$. For this purpose, we use the most common function:

$$F_{xi} = R_{zi} \mu_{pxi} (1 - exp(-k_{moi} s_{\delta ai})) \, , \tag{10}$$

where R_{zi} is the i^{th} wheel normal reaction, the factor K_{moi} and the cohesion (grip) coefficient μ_{pxi} are both dependent on properties of the tire and ground contact. The approximate values for μ_{pxi} and K_{moi} are usually determined using experiment data. In addition, these values may be different for the wheels of the three axles moving in the same track on a soft surface.

It is necessary to emphasize here that the placement of center of gravity and axle arrangement along the tractor base may influence the normal axle loads and subsequently tractive forces and tractive efficiency of the tractor's running gears. Thus, by determining the normal load as dependent on coordinates of the gravity center and placement of the middle wheels, it is possible to find a combination of these coordinates appropriate for maximum tractive efficiency. The geometric tractor parameters found in this way provide not only the most feasible engineering solution but also the most economical tractor for basic off road mobility. In order to achieve these parameter values, we make a mathematical model of the tractor with an individual wheel suspension as shown in Fig.1. To determine the axle normal reaction values, we use force and moment balance equations acting on the tractor as follows:

$$R_{z1} + R_{z2} + R_{z3} - R_{z4} = W_a \, , \tag{11}$$

taking the moments about point O_1 of Fig.1:

$$W_a l_a - \sum_{i=1}^{3}(F_{xi} - R_{xi})a_i + F_d h_d - \sum_{i=2}^{3} R_{zi} l_i + R_{z4} l_4 = 0 \tag{12}$$

where W_a is the weight of the tractor and R_{z4} is the vertical downward trailer hitch force. The displacements, a_i's (i= 1, 3), are usually found from tire-ground interaction.

It is clear from above that the number of unknown axle normal reactions exceeds the number of equations (11) and (12). To solve this problem, we take advantage of the equations of elastic travel, Z_i of the tractor frame points above the three axles having three individual suspensions. These equations are as follows.

$$R_{zi} = C_{ri} Z_i \tag{13}$$

$$C_{ri} = \frac{C_{si} C_{ti} C_{gi}}{C_{gi} C_{si} + C_{ti} C_{gi} + C_{si} C_{ti}}, \ i = 1, 3, \tag{14}$$

Where C_{ti}, C_{gi}, C_{si} are the tire, ground, and suspension stiffness, respectively. The C_{ri} is called reduced rigidity stiffness of the suspension - tire - terrain system. To transform the travel of the points above the second and third axles; through the travel of the point above the first axle, the longitudinal axle coordinates and the inclination angle, θ of the frame relative to the surface of motion, we obtain the following formulas for the axle normal reactions:

$$Z_i = Z_1 - l_i \tan\theta, \ i = 2, 3, \tag{15}$$

Where l_i's is given in Fig.2. Combining equations (12), (14) and (15), and solving for Z_1, we get:

$$Z_1 = (W_{a+} \tan\theta \sum_{i=1}^{3} C_{ri} l_i) / \sum_{i=1}^{3} C_{ri} \tag{16}$$

Solving Eq.(16) for $\tan\theta$ and substituting into Eq. (15), we get the axle normal reactions R_{zi} from Eq.(13):

$$R_{zi} = C_{ri}(((W_a(C_{r2}l_2^2 + C_{r3}l_3^2 - l_a\sum_{i=1}^{3} C_{ri}l_i) + \sum_{i=1}^{3} C_{ri}l_i \sum_{i=1}^{3}(F_{xi} - R_{xi})a_1 - \sum_{i=1}^{3} C_{ri}l_i F_d(h_d - a_1))/$$

$$(\sum_{i=1}^{3} C_{ri}(C_{r2}l_2^2 + C_{r3}l_3^2) - \sum_{i=1}^{3} C_{ri}l_i(C_{r2}l_2 + C_{r3}l_3)) - l_i((Z_i\sum_{i=1}^{3} C_{ri} - W_a)/\sum_{i=1}^{3} C_{ri}l_i)) \tag{17}$$

where i =1, 2 and 3 with $l_1 = 0$;

Using Eq. (10) along with equation (17), we can find the tractive forces, F_{xi}'s. Therefore, the equation of motion of the tractor at a constant speed may be written as follows:

$$F_{x\Sigma} = \sum_{i=1}^{3} F_{xi} = \sum_{i=1}^{3} F_{fi}^{0} + F_d + \sum_{j=1}^{s} F_j \tag{18}$$

where $\sum_{j=1}^{3} F_{fi}^{0}$ is the sum of the rolling resistance forces, $\sum_{j=1}^{s} F_j$ is the algebraic sum of external forces acting on the tractor; including air resistance force and the inertial forces. Other forces may be defined by a research problem of interest. In our case of steady motion of the tractor: $\sum_{j=1}^{s} F_j$ =0.

The total rolling resistance forces of axles may be computed by using formulas available. On changing terrain, the front wheels move on non-tamped terrain, the middle wheels move in the track of the front wheels and the rear wheels move in the track of the middle wheels. After calculating the total tractive force, $F_{x\Sigma}$, we may optimize all the tractive axle forces F_{xi}'s, to obtain the maximum tractive efficiency. In this paper I have solved this problem for various combinations of wheel arrangements, different placements of the center of gravity, and for various values of F_d subject to the constraints:

$$\eta_x^t \to max, \tag{19}$$

$$0 < F_{xi}^* < R_{zi}\mu_{pxi}, \; i = 1, 3, \tag{20}$$

$$F_{x\Sigma} = \sum_{i=1}^{3} F_{xi}^* \tag{21}$$

The above analysis provides a mathematical model for the optimization of the basic geometric parameters of a six wheel drive tractor having maximum running gear efficiency. The model was developed with the assumption that the tractor's drawbar pull is horizontal with R_{z4} equals to zero. Figure 2 shows the free body diagram of a six wheel drive independent suspension tractor model on a deformable soil.

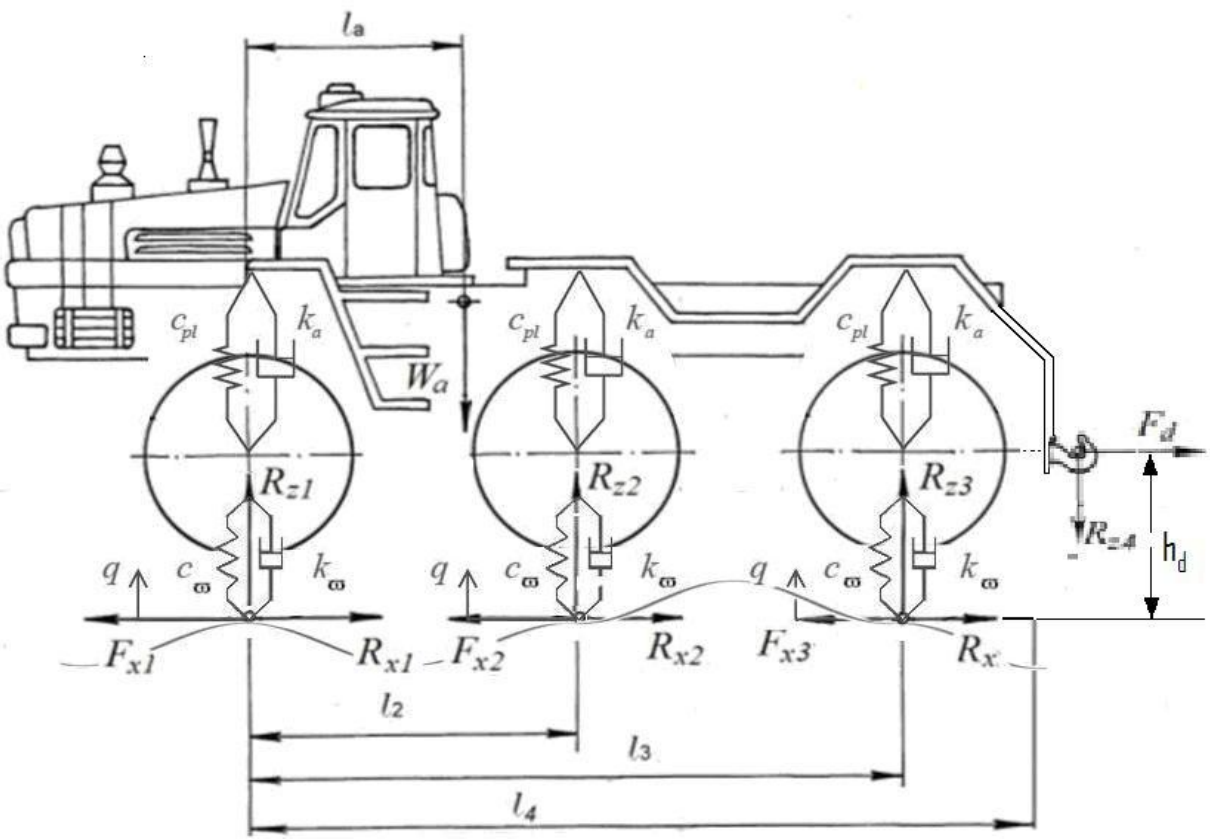


Fig.2. A six wheel drive tractor: free-body diagram

Table.1 Parameters for normal reaction calculation

Parameters/Units	Notation	Value
Weight, kN	W_a	90
Front axle tire normal stiffness, N/m	C_{t1}	840,000
Middle axle tire normal stiffness, N/m	C_{t2}	613,200
Rear axle tire normal stiffness, N/m	C_{t3}	562,800
Suspension stiffness, N/m	$C_{t1} = C_{t2} = C_{t3}$	1,600,000
Soil stiffness under the front axle, N/m	C_{g1}	800,000
Soil stiffness under the middle axle, N/m	C_{g2}	1,800,000
Soil stiffness under the rear axle, N/m	C_{g3}	3,680,000
Front axle rolling resistance coefficient	f_1	0.1
Middle axle rolling resistance coefficient	f_2	0.08

Rear axle rolling resistance coefficient	f_3	0.065
Height of the draw-bar pull from ground, m	h_d	3.5
Radius of the wheels, m	$a_1 = a_2 = a_3$	0.3

The mathematical model of optimization of the distribution of power between the axles for these travel conditions can be rewritten as:

$$\eta_x^t = \frac{F_d}{\sum_{i=1}^{3} F_{xi} + \sum_{i=1}^{3} \frac{F_{xi} s_{\delta ai}}{1 - s_{\delta ai}}} \tag{22a}$$

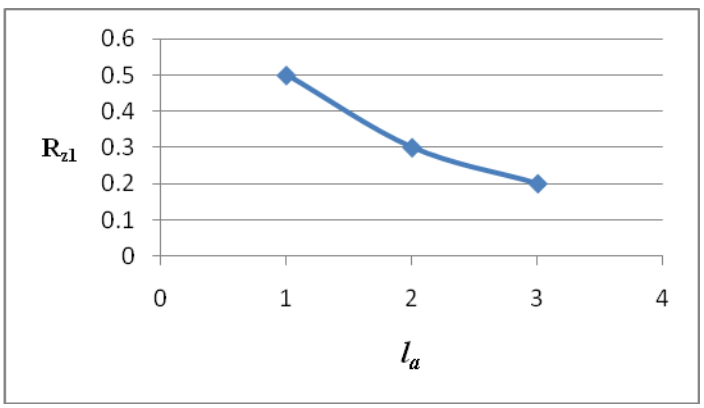
$$\sum_{i=1}^{3} F_{xi} = F_{x\Sigma} = \sum_{i=1}^{3} R_{xi} + F_d \tag{22b}$$

$$0 < F_{xi}^* < \mu_{pxi} R_{zi} \tag{22c}$$

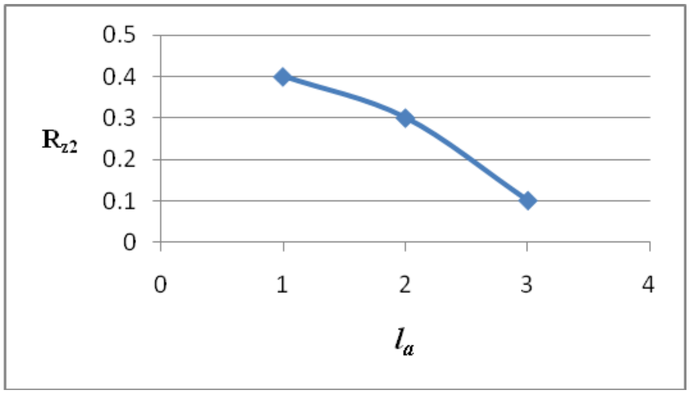
$$F_{xi}^* = R_{zi} \mu_{pxi} \left(1 - exp\left(-k_i s_{\delta ai}^*\right)\right) \tag{22d}$$

Results

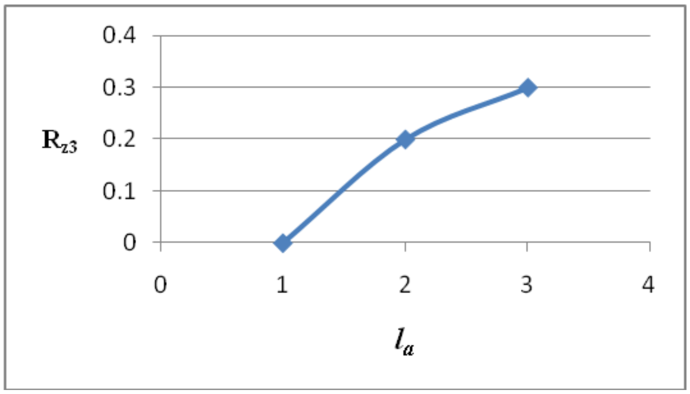
The results of a 9-ton 6×6 tractor with 16.9R30 tires moving over a plowed field with caked soil are as follows. Figure 3 shows the distribution of normal reactions between the axles as a function of the location of center of gravity l_a and for three forms of middle wheel arrangements as close as possible to : i) the front wheels, ii) the middle of the wheelbase, and iii) the rear wheels.



R_{z1} Versus l_a for l_2 =1.5



R_{z2} Versus l_a for l_2 =1.5

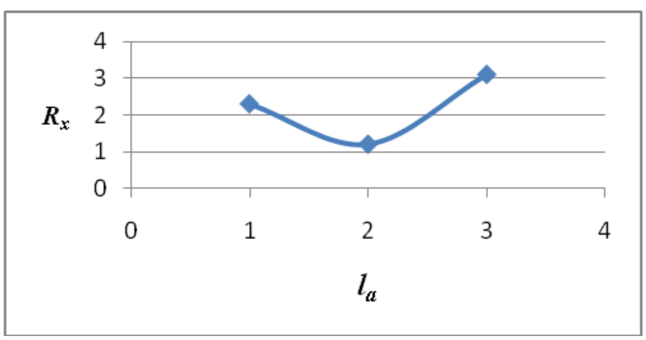


R_{z3} Versus l_a for l_2 =1.5

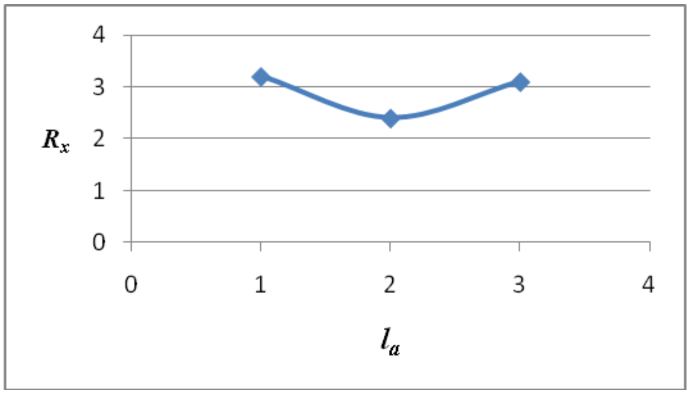
Fig3. Normal reactions of the axles when l_2 =1.5

R_x Plots

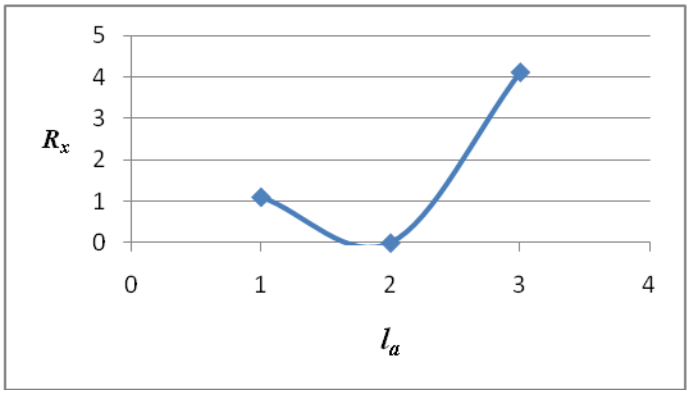
The total rolling resistance curves of the total resistance to the rolling of the wheels, $R_{x\Sigma} = \sum_{i=1}^{3} R_{xi}$, for these three arrangements are plotted in Fig. 4.



R_x versus l_a for l_2=1.5



R_x versus l_a for l_2=2

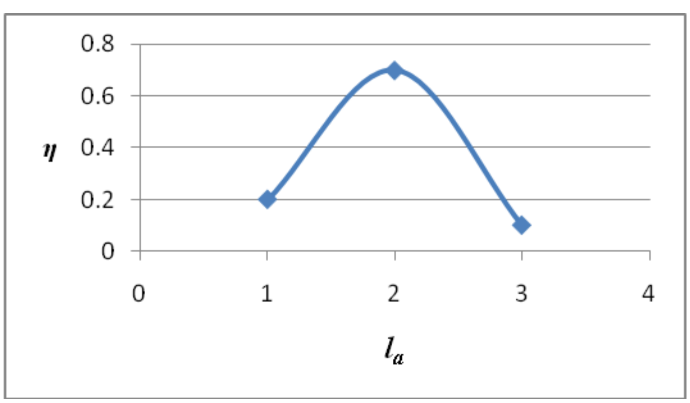


R_x versus l_a for l_2=3

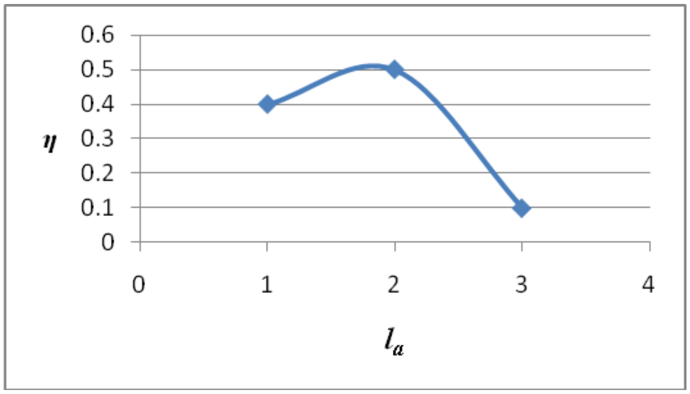
Fig4. Total rolling resistance $R_{x\Sigma}$ for l_2=1.5, 2, and 3.

Efficiency without acceleration

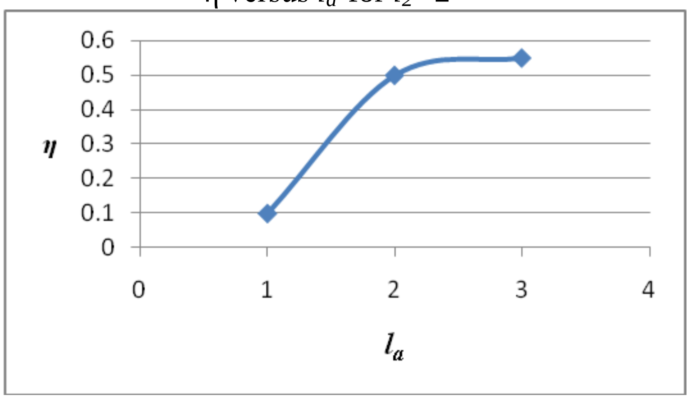
The maximum possible tractive efficiencies on a running gear are shown in Fig.5 provided by the pertinent distribution of power between the axles, where the l_a is along the x-axis and the η is along y-axis.



η versus l_a for l_2=1.5



η versus l_a for l_2=2



η versus l_a for l_2=3

Fig.5. Maximum tractive efficiency of the tractor on a running gear for l_2=1.5, 2, and 3.

The distribution of the normal loads between the axles has a remarkable effect on the rolling resistance $R_{x\Sigma}$. In all the wheel arrangements the value of $R_{x\Sigma}$ is at minimum in regions with uniform distribution of normal reaction R_{zi}. The minimum values of $R_{x\Sigma}$ are observed with increasing F_d at lower values of l_a.

It follows from Fig. 5 that the middle-wheel location arrangements discussed here have, under different traction loads, such locations of the center of gravity l_a at which η_x attains the maximum value of the maximum possible. When F_d increases, the values of l_a decrease for all the three arrangements of the middle wheels.

The maximum possible values of η_x correspond to the minimum of rolling resistance and uniform distribution of the loads between the axles, respectively as shown Figs. 4 and 3. When the middle wheels are shifted from the front wheels to the rear wheels, the maximum efficiency of the maximum possible changes somewhat as depicted in Fig. 5. It is hence of interest to investigate in more detail the effect of parameter l_2 on the traction performance of the vehicle.

Determination of the reaction forces including acceleration

The addition of the acceleration force to the momentum equation shows us a clearer picture of the behavior of the reaction force and also the efficiency. By D'Alembert's Principle we have,

$$F - ma = 0 \tag{23}$$

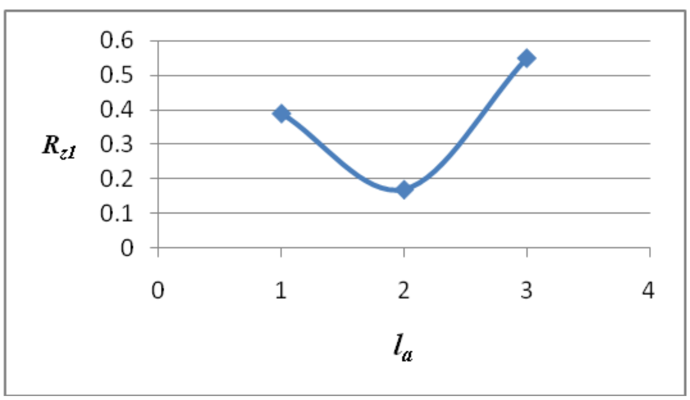
Where 'a' is the acceleration of the vehicle. Now, by adding the acceleration term to momentum equation, Eqn (12) is revised as,

$$W_a l_a - \sum_{i=1}^{3}(F_{xi} - R_{xi})a_1 + F_d(h_d - a_1) - R_{z2} l_2 - R_{z3} l_3 + (W_a / g)a = 0 \tag{24}$$

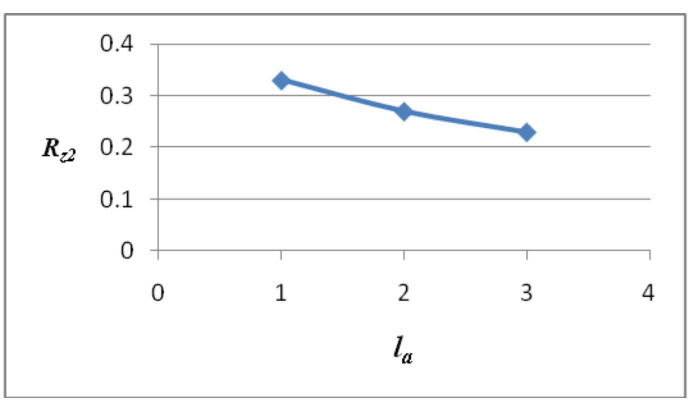
Solving for R_{zi}, we have

$$\begin{aligned}\mathrm{Rzi} = {} & C_{ri}(((W_a(C_{r2} l_2^2 + C_{r3} l_3^2 - l_a \sum_{i=1}^{3} C_{ri} l_i - (a/g)\sum_{i=1}^{3} C_{ri} l_i) + \sum_{i=1}^{3} C_{ri} l_i \sum_{i=1}^{3}(F_{xi} - R_{xi})a_1 - \\ & \sum_{i=1}^{3} C_{ri} l_i F_d (h_d - a_1))/(\sum_{i=1}^{3} C_{ri}(C_{r2} l_2^2 + C_{r3} l_3^2) - \sum_{i=1}^{3} C_{ri} l_i (C_{r2} l_2 + C_{r3} l_3)) - \\ & l_i((Z_i \sum_{i=1}^{3} C_{ri} - W_a)/\sum_{i=1}^{3} C_{ri} l_i))\end{aligned} \tag{25}$$

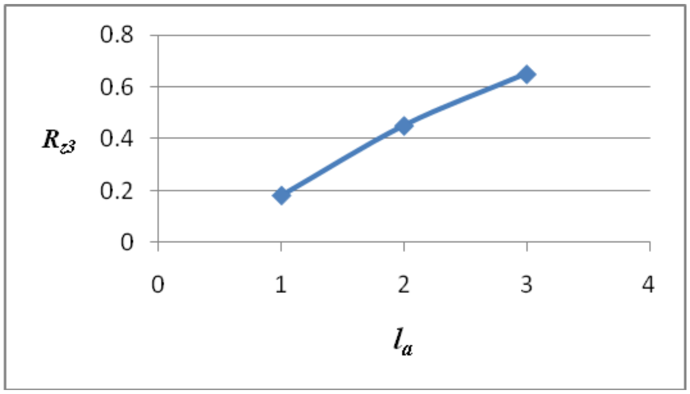
Results



R_{z1} versus l_a for l_2=1.5



R_{z2} versus l_a for l_2=1.5



R_{z3} versus l_a for l_2= 1.5

Fig6. Normal reactions of the axles when l_2=1.5

R_x Plots

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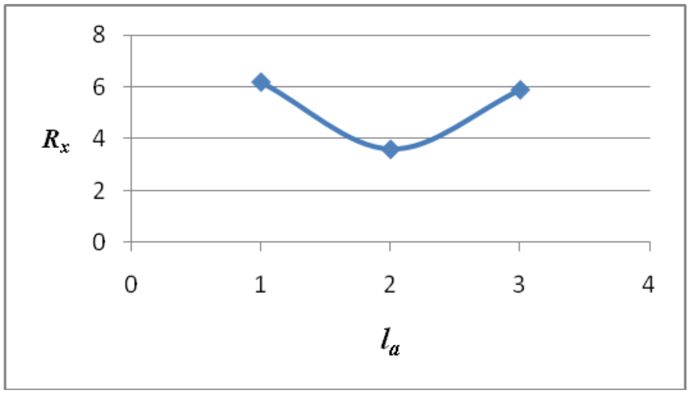
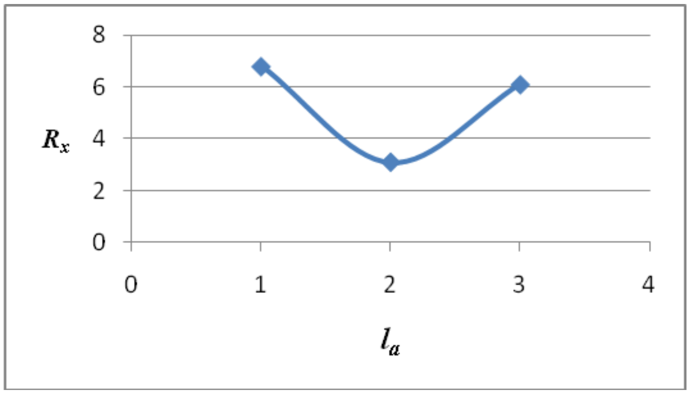
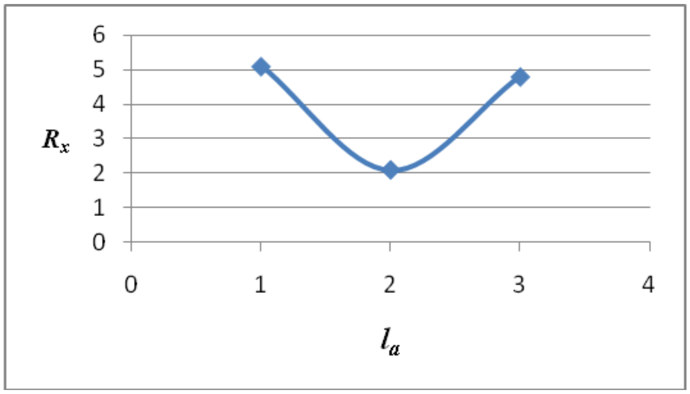


R_x versus l_a for l_2=1.5



R_x versus l_a for l_2=2

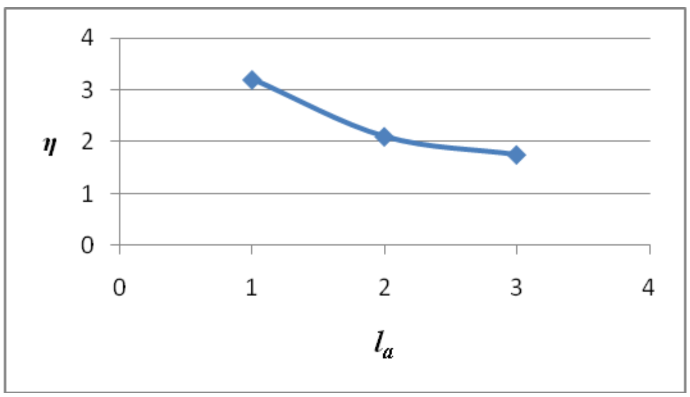


R_x versus l_a for l_2=3

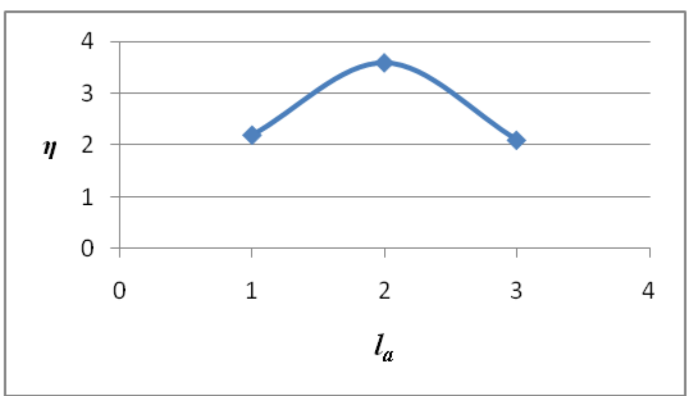
Fig7. Total rolling resistance $R_{x\Sigma}$ for l_2=1.5, 2, and 3.

Efficiency with acceleration

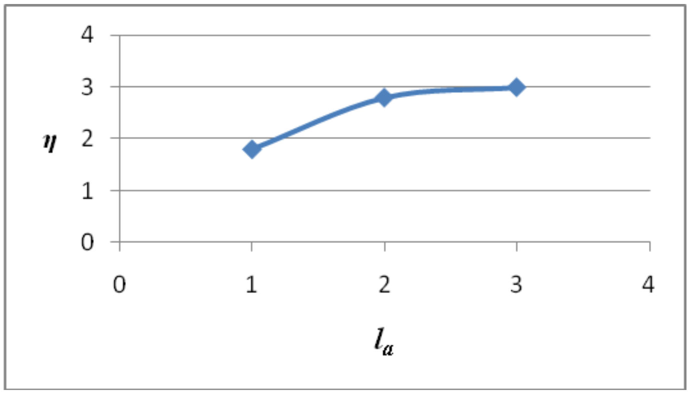
The maximum possible tractive efficiencies on a running gear are shown in Fig.8 provided by the pertinent distribution of power between the axles where η is along y-axis and l_a is along x-axis.



η versus l_a for l_2=1.5



η versus l_a for l_2=2



η versus l_a for l_2=3

Fig.8. Maximum tractive efficiency of the tractor on a running gear for l_2=1.5, 2, and 3.

Conclusion

The results show the influence of the driveline system parameter arrangement of a six wheel drive tractor on rolling resistance forces, power needed to overcome these forces, and power lost due to drive wheel slippage all of which determine the efficiency of the running gear system. The developed theoretical statements and mathematical model have been used to solve the following problems: a) determination of optimum coordinates of the center of gravity under the given arrangement of the middle axle wheels along the tractor base; b) determination of optimum arrangement of the middle axle wheels along the base under the given coordinates of the center of gravity; c) determination of optimum coordinates of the center of gravity and arrangement of the middle axle wheels along the tractor base. We were also able to see how the consideration of acceleration influenced the various plots. The curves got shifted indicating the increase in the tractive force of the vehicle. Similarly, we could generate the plots for different values of l_2 for maximum traction.

Recently, various devices have been developed to replace the vehicle's center of gravity position for changing drawbar pull. The paper can be further extended by also considering other dynamical factors acting on the vehicle. The result of this paper can be used for the development of optimum algorithms to control the replacing process.

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