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Modeling the Influence of Internal Factors on Vehicular Mobility Performance

Aziz Mdimi[1†], Geoffrey R John[2], Cuthbert Mhilu[2], Joseph Kihedu[2], and Hannibal Bwire[3]

[1]National Institute of Transport, Dar es Salaam, Tanzania
[2]Department of Mechanical and Industrial Engineering, College of Engineering Technology, University of Dar es Salaam, Tanzania
[3]Department of Department of Transportation and Geotechnical Engineering, College of Engineering and Technology, University of Dar es Salaam, Tanzania
[†]Corresponding author: mdimiaziz@gmail.com; ORCID: 0009-0005-7677-535X

ABSTRACT

Traffic mobility is significantly influenced by road conditions and the internal factors governing vehicular performance. Internal factors play a crucial role in vehicular mobility, influencing vehicles operate and performance. While existing vehicular mobility models primarily address external factors, external factors such as governor, engine, gear train and differential unit remain underexplored. This study formulates a comprehensive model to address this gap, focusing on the internal components that affect vehicular mobility. The proposed model employs both analytical and numerical methods to derive transfer functions for these subsystems using Matrix Laboratory (MATLAB), aiming to capture the dynamic behavior of vehicles under various conditions. The model was evaluated by examining vehicle transaction times across different road surface qualities, measured by the International Roughness Index (IRI) values of 3.5, 3.0, and 2.0, and tractive forces ranging from 4500 N to 17750 N, with applied pedal forces of 50 N, 100 N, 150 N and 200 N. Results indicated that higher tractive forces lead to reduced transaction times, with IRI values of 3.5 m/km showing a decrease from 122.0 seconds to 31.0 seconds as tractive forces increased from 4500 N to 17750 N. Similarly, for IRI values of 2.0 m/km, the transaction times reduced from 67.5 seconds to 7.5 seconds under the same conditions. The analysis further demonstrated that increased applied pedal forces correspond to higher tractive forces, thereby enhancing vehicular mobility and performance. These findings highlight the critical role of internal factors in optimizing vehicular mobility and performance, suggesting that internal subsystem dynamics should be integrated into future mobility models for more accurate and comprehensive assessments.

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INTRODUCTION

The interaction between the wheel tyre and the road surface determines vehicular mobility characteristics, which affect tractive effort and vehicle speed (Belousov and Popov, 2014). Dynamic wheel torque being supplied to the wheel by the vehicle's driveline system causes on or progressive mobility. The dynamics depend on the characteristics of the vehicle's driveline system (Vantsevich, 2015). Automotive companies have introduced torque-bias coupling, torque-vectoring and torque management devices to control and improve vehicle mobility based on individual wheel power distribution, as mentioned by Lenzo *et al*. (2019); Lenzo *et al*. (2017) and Parker *et al.* (2015). These driveline systems need to be flexible and proactive in controlling power delivery to the front and rear axles Krajnik *et al*. (2021) so as to distribute the engine power to the driving wheels converted to tractive force.

Vehicular mobility depends on road conditions occurring and comprises of off-road and on-road conditions. The off-road is the condition of road which provides the ability for a vehicle to operate on particular terrain Ragheb (2014). The mechanics of vehicle mobility for off-road conditions depends on the terrain where, soil strength, surface shape such as climb, bend and stench are involved Pakowski and Cao (2013). On-road conditions interaction of the vehicle tyre and road surface factors such as grip surface, the vehicle speed, engine and mechanical characteristics play a role. The factors may deteriorate mobility and possibly immobilize the vehicle on the road. Even though many studies on vehicular mobility modeling exist for off-road external factors, little exists on on-road internal factors. The motivation to the present study is to formulate a model for investigating the effects of selected internal factors on on-road vehicle affecting vehicle mobility.

The existing literature on off-road factors such as wheel power management, factors affecting field mobility of off-road vehicles, tractive efficiency of off-road vehicles mobility and mobility of multi-wheel drive vehicles have been contributed by such scholars as Vantsevich *et al*. (2020); George *et al*. (2017) and Vantsevich (2015). On the other hand, very limited studies have been reported on on-road conditions with internal factors such as engine efficiency, driveline systems and vehicle speed capability. Kim *et al*. (2017) developed algorithm to estimate the torque of each clutch separately. Their effect on the driveline during the gear shift translates to vehicle deceleration as a result of traction loss and driveline oscillation due to variation of torque transmitted. Recognizable mobility models to predict the extent of on the road mobility conditions exists for off-road and to a lesser extent for on road conditions Swamy *et al*. (2023); Taheri *et al*. (2015); Madi and Al-Qamzi, (2013). Their works neglected mobility analytical scenarios encompassing internal factors affecting vehicle mobility on on-road condition. The current model presented herein, now model vehicular mobility performance characteristics based on internal factors. It addresses the oversights by developing the model predicting the relationship between road surface roughness with vehicle speed as a function of applied pedal and attractive force. The approach is analytical and depends on the Laplace transfer function system.

METHODS AND MATERIALS

Model Development

The development of the model was based on on-road measured factors obtained from field work. In order to conduct data analysis mathematical modelling approach was used. This involved a determination of mathematical expressions for each individual system components of the model developed which affect mobility internal factors sub-system governor, engine, gear train, differential unit and Mobility performance unit which is speed.

The approach deploys both analytical and numerical methods using ordinary differential equations (ODE) to develop sub-system vehicle models. The ordinary differential equations for each of the system components have to be considered. There are numerical solutions of the equations requiring to be expressed as system of 1st and 2nd order ordinary differential equations and applying numerical integration schemes, such as Runge Kutta integration. In view of time constraint an alternative integration method was adopted to solve transfer functions using MATLAB.

The resulting conceptual model representing a general form for determining vehicular mobility performance characteristics is presented in form of block diagram given in Figure 1.

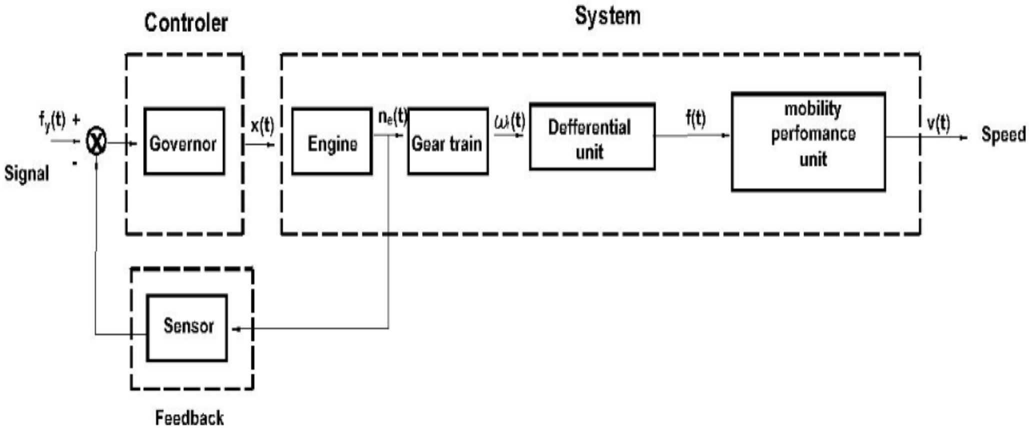

Figure 1: Conceptual block diagram of vehicular mobility characteristics model.

where, $f_y(t)$ is the force applied on to the vehicle accelerator pedal (N), $x(t)$ is the governor fuel rack displacement (mm), $n_e(t)$ is the engine speed (rpm), $\omega_1(t)$ is the angular speed of gear train (rad/s), $f(t)$ is the engine's tractive force (N) and $v(t)$ is the vehicle mobility speed (m/s).

Sub-models Development

In order to evaluate the resulting model, each of the subsystem was evaluated to be used for studying the model behavior.

Governor sub model, In a diesel engine, the governor controls the speed of the engine by measuring angular speed of the engine (ω_e) in (rad/s) and responding to speed variations due to operating conditions. To model the corresponding governor rack displacement x(t) responding to the accelerator pedal force $f_y(t)$, a spring – mass damper system was used to model the engine governor as shown in Eq. 1.0.

$$m\frac{d^2x(t)}{dt^2} = f_y(t) - f_s(t) - f_d(t) \quad (1)$$

where, m is the mass of governor flyweight, in (kg), $\frac{d^2x(t)}{dt^2}$ is the acceleration of the body in m/s^2, $f_y(t)$ is the force applied on to vehicle accelerator pedal (N), $f_s(t)$ is the force in (N) exerted by the spring, $f_d(t)$ is the force in (N) exerted by the damper. The resulting transfer function $G_1(s)$ is given in Eq. 2.0.

$$G_1(s) = \frac{X(s)}{F_y(s)} = \frac{d}{s^2 + as + b} \quad (2)$$

where, $G_1(s)$ is Transfer function of the governor, $X(s)$ is the fuel rack displacement in mm, and $F_y(s)$is the force applied on to vehicle accelerator pedal (N), a, b and d are constant parameter.

Transfer functions describing the behavior of vehicular mobility for the governor is represented by a block diagram depicted in Figure 2.

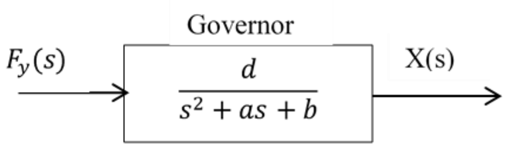

Figure 2: Block diagram of the Governor.

Engine performance sub model

The relationship between fuel rack displacement and engine speed is determined by making use of a mathematical relationship derived and presented in Eq. 3.0:

$$\tau_e \frac{dn_e(t)}{dt} + K_e n_e(t) = x(t) \tag{3}$$

where, τ_e is the time constant of the engine in seconds (s), n_e is the engine speed obtained from the sensor of revolution per minutes (rpm), K_e is the dimensionless coefficient of self –regulation of the engine. The resulting transfer Equation 3 is presented in Equation (4.0) as;

$$(\tau_e s + K_e)N_e(s) = X(s) \tag{4}$$

The resulting transfer function of the engine $G_2(s)$ is obtained in Eq. 5 as:

$$G_2(s) = \frac{N_e(s)}{X\ (s)} = \frac{1}{\tau_e\ s + K_e} \tag{5}$$

where, τ_e is the time constant of the engine, K_e is the dimensionless coefficient of self-regulation of the engine. The corresponding block diagram is shown in Figure 3

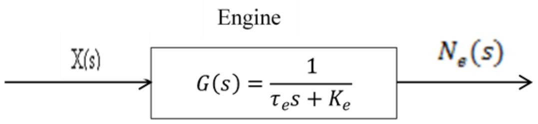

Figure 3: Block diagram of the engine performance.

Transmission system model

The transmission system composed of gear train and differential unit. The corresponding mathematical equations representing a gear train and differential unit are describe,

Gear train model

The relationship between engine speed $n_e(t)$, and the resulting angular speed $\omega_1(t)$ of the primary shaft can be expressed with relation to input engine speed as follows:

$$\omega_1(t) = \frac{2\pi n_e(t)}{60} = \frac{\pi n_e(t)}{30} \quad \text{in (rad/s)}$$

where, $\tau_n = \frac{30}{\pi}$ a constant,

The corresponding transfer function $G_3(s)$ of the gear train as shown in Eq (6.0).

$$G_3(s) = \frac{\omega_1(s)}{N_e(s)} = \frac{1}{\tau_n} \tag{6}$$

Hence, the resulting block diagram is given in Figure 4 below;

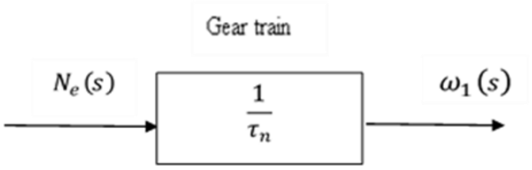

Figure 4: Block diagram of the gear train.

Differential unit system model

The driveline of a vehicle equipped with transmission is composed of the primary shaft which transmits the angular speed (ω_1) from the engine to achieve the corresponding angular speed of the secondary shaft (ω_2) transmitted to the differential is obtained by making use of a mathematical equation expressed in form of speed ratio;

$$\tau_r = \frac{\omega_2(t)}{\omega_1(t)} \tag{7}$$

$$\omega_2 = \tau_r\,\omega_1(t)$$

where, N_r represents speed ratio 4:1, hence, τ_r = 0.25. The corresponding transfer function $G_4(s)$ is the differential unit. From Laplace transform obtained in Eq. (8.0) as:

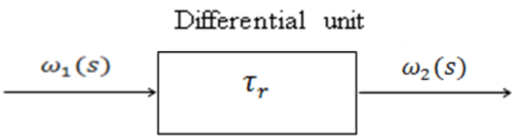

Figure 5: Block diagram of the differential unit.

Engine tractive force model

Taking into consideration of the block diagram of the vehicle wheel shown in Figure 6 the expected vehicular translational speed can be expressed by making use of the momentum equation put into the following form;

$$\beta F_t = M_v \frac{dv(t)}{dt} \quad (9)$$

where β is number of driving wheels. For the purpose of this study β = 2 wheels.
The corresponding Laplace transform is presented in Eq. (10.0) as,

$$F_t(s) = \frac{R_w}{\beta}\omega_2(s) \quad (10)$$

The block diagram representing the transfer relationship between the differential angular speed and the resulting tractive force is as given in Figure 6:

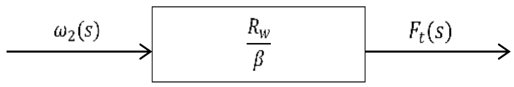

Figure 6: Block diagram of the Vehicle wheel.

Overall transfer function for test vehicle

The overall transfer function in block diagrams is obtained emulating speed governor settings and the resulting tractive effort realised is presented in Figure 7.

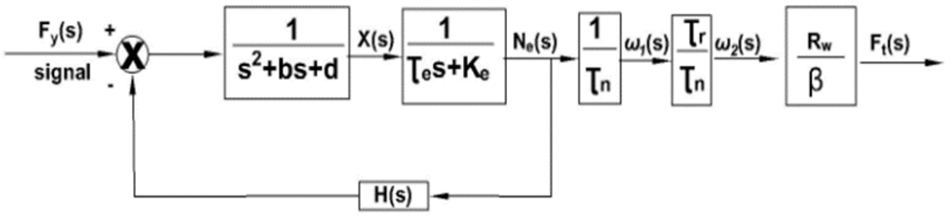

Figure 7: The overall transfer function of test vehicle engine.

where, $F_y(s)$ is the force applied on to vehicle accelerator pedal (N), $X(s)$ is the governor fuel rack displacement (mm), $N_e(s)$ is the engine speed (rpm), ω_1(s) is the angular speed of gear train (rad/s), ω_2(s) is the angular speed of Differential unit (rad/s), F_t (s) is the engine's traction force (N).

Vehicular Mobility Model Formulation

Study assumes surface is horizontal and hence the dynamic equation used to model the vehicle motion can be expressed in the following form:

$$M_v \frac{dv(t)}{dt} = \beta F_t(t) - F_{ae}(t) - F_{ro}(t) \quad (11)$$

where, β- is the number of driving wheels, M_v – is the vehicle mass (kg), and $v(t)$ is the vehicle mobility speed (m/s). F_t is the tractive force (N), F_{ro} is the tyre rolling resistance (N) and F_{ae} is the aerodynamic drag (N)..
In order to evaluate the vehicular mobility based on the internal factors based on surface conditions use is made of the derived model expressed in the form of a transfer function given in the form.
After some re-arrangement Equation 11.0 obtained resulting equation used to simulate vehicular speed under different road surface condition presented in form of Eq. 12.0

$$G_{mob}(s) = \frac{V(s)}{F_t(s)} = \frac{\frac{\propto}{M_v}}{s^2 + \boldsymbol{IRI}s + \frac{K_o}{M_v}A_v(R_w\omega_2)} \quad (12)$$

where, the constant (**IRI**), the International Roughness Index represents frictional resistance (m/km), is a function of road surface quality, A_v is the frontal area of the vehicle (m^2), v is the speed of vehicle (m/s), M_v – is the vehicle mass (kg), $K_o = \left(\frac{C_{dr}\rho_a}{2}\right)$ = constant, R_w is the radius of the vehicle wheel (m), ω_2 is the angular speed of the differential (rad/s), β is number of driving wheels $\propto$ - is the total number of vehicle wheels, C_{dr} drag coefficient-dimensionless, ρ_a air density (kg/m^3) and F_t is the tractive force (N).

$$V(s) = F_t * \left[\frac{\frac{\propto}{M_v}}{s^2 + \boldsymbol{IRI}s + \frac{K_o}{M_v}\left(A_v \frac{{R_w}^2}{\beta} * F_t\right)} \right] \tag{13}$$

RESULTS AND DISCUSSIONS

Road surface roughness analysis data is compared with the recommended minimum ISO values of less than 3.0 m/km. An IRI value of 3.5 m/km, is considered along with that of IRI value of 2.0 IRI. The larger IRI value greater than the ISO minimum value signifies a severe deterioration of road surface which may affect the average degree of mobility in terms of speed, (Lei *et al.* 2017) but extremely smooth roads can also lead to reduced tyre road friction, affecting braking performance and safety. Actual IRI less than the ISO minimum, infer the road surface is smoother than the standard. Smoother roads generally offer better speed, ride comfort, reduce vehicle wear and improved fuel efficiency Adeli *et al*. (2021). Poor mobility result into excess fuel consumption Botshekan *et al*. (2019) and Louhghalam *et al*. (2015).

Based on the statistical significance of the internal factors, i.e tractive force and applied force indicated that at the tractive force of 4500 N, when applying a pedal force of 50 N the time spent on road segment of IRI value of 3.5 m/km is 122.0 seconds. The same at the tractive force of 4500 N with pedal force of 50 N on road segment of 2.0 m/km value the time spent is 67.5 seconds these means that the longer transacted time the more time for vehicular mobility to cover the road segment.

In addressing the findings, it was realized that vehicle internal factors enhance positively vehicle mobility.

Practical implication on findings

The implications of longer transacted time and low fuel displacement at the tractive of 4500 N have lead to lower engine revolution speeds occurs. In this case, for urban traffic management can be significant in slower vehicle movement and contributing to vehicle immobility, especially during peak hours. The urban planners might prioritize the development and use of public transportation. Therefore needs of implementing traffic management strategies such as synchronized traffic lights, dedicated bus lanes, and congestion pricing can help manage the flow of traffic more effectively, reducing the negative impacts of vehicle immobility. However, this can worsen air quality in urban areas and contributing to environmental and health issues.

The limitation of this study is its reliance on simulation data, which may not fully captured real-world. Figures show the predicted values along with their 95% confidence intervals.

Model performance Analysis

Analysis is made on simulation data obtained by running a MATLAB program with roughness (IRI) values (m/km), given in Table 1.

Table 1: Input pedal force corresponding Tractive force with Roughness values

S/N	Pedal force $f_y(t)$ [N]	Tractive force [N]	Segments roughness IRI m/km		
1	50	4500	3.5	2.0	3.0*
2	100	8800			
3	150	13100			
4	200	17750			

* Means stipulated standard value of roughness < 3.0 m/km

Table 2 shows time taken for a vehicle to cover 1000 m at a particular pedal force and tractive force for different IRI. The analysis of vehicle tractive forces at applied forces with their IRIs is indicated that at the applied pedal force of 50 N, the tractive force response to 4500 N on road segment of 3.5 m/km IRI value, the transacted time is 122.2 seconds. Similarly, at the applied force of 100 N, the tractive force response to 4500 N on road segment of 3.0 m/km IRI value, the transacted time is 52.0 seconds.

Table 2: Time taken for a vehicle to cover 1000 m at a particular pedal force and tractive force for different IRI

F_t (N / Fy	50	100	150	200	IRI
4500	122.2	61.0	48.6.0	34.0	3.5
	105.0	52.0	34.0	27.0	3.0
	67.5	31.5	26.7	20.0	2.0
8800	120.0	57.0.0	39.5	33.0	3.5
	104.0	46.0	34.6	26.0	3.0
	66.0	33.0	22.0	19.0	2.0
13100	118.0	55.0	36.0	32.0	3.5
	109.5	38.0	36.5	24.0	3.0
	96.00	64.0	20.0	18.0	2.0
17750	124.0	50.0	35.0	31.0	3.5
	100.0	30.0	34.6	22.0	3.0
	69.3	34.6	18.0	17.5	2.0

Note: x = 1000 m (Distance), v = 47 m/s

(a) *IRI values of 3.5 m/km*

Figure 8 shows simulation responses on a surface roughness of IRI = 3.5 m/km along a segment length of 1000 m from rest to a speed of 47 m/s. The time spent on this segment is 122.0 seconds at a tractive force of 4500 N when applying a pedal force of 50 N on vehicle accelerator pedal, the time spent at a tractive force of 8800 N by applying force of 100 N on vehicle accelerator pedal is 61.0 seconds; 40.0 seconds at a tractive force of 13100 N on vehicle accelerator pedal of 150 N and 31.0 seconds at a tractive force of 17750 N on vehicle accelerator pedal force of 200 N.

(b) *IRI value of 2.0 m/km*

Figure 9 shows case for a road segment which has a length of 1000 m. In considering simulation response, vehicle was run from rest to a speed of 47 m/s on on road roughness with value 2.0 m/km.k.. The time taken to cover the distance at a tractive force of 4500 N by applying force of 50 N on accelerator pedal was 67.5 seconds. Similarly, the time taken to cover the tractive forces of 8800 N on accelerator pedal of 100 N, 13100 N on accelerator pedal of 150 N and 17750 N on accelerator pedal of 200 N was 33.0 seconds, 32.0 seconds and 17.5 seconds respectively.

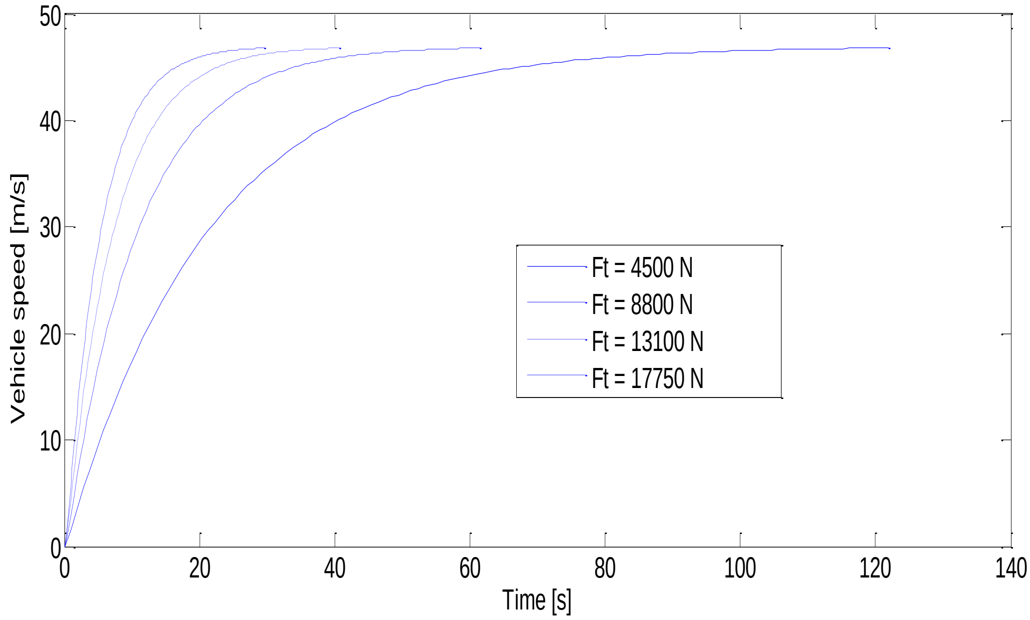

Figure 8: Mobility results of the measured road segment at IRI 3.5 m/km.

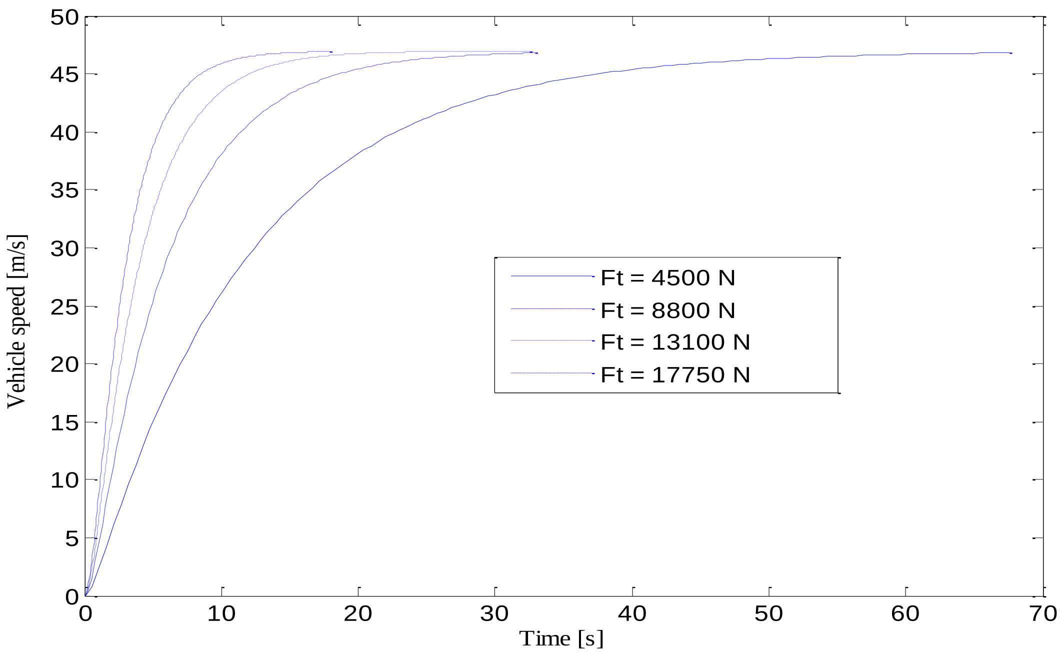

Figure 9: Mobility results of the measured segment at IRI 2.0 m/km.

(c) *IRI value of 3.0 m/km*

Figure 10 show results for a length of 1000 m also. The simulation response was made for a vehicle run from rest to a speed of 47 m/s on the road roughness. The time taken to cover the distance at a tractive force of 4500 N by applying force of 50 N on accelerator pedal was 105.0 seconds. The time taken to cover the tractive forces of 8800 N on accelerator pedal of 100 N, 13100 N on accelerator pedal of 150 N and 17750 N on accelerator pedal of 200 N was 52.0 seconds, 36.5 seconds and 26.0 seconds respectively.

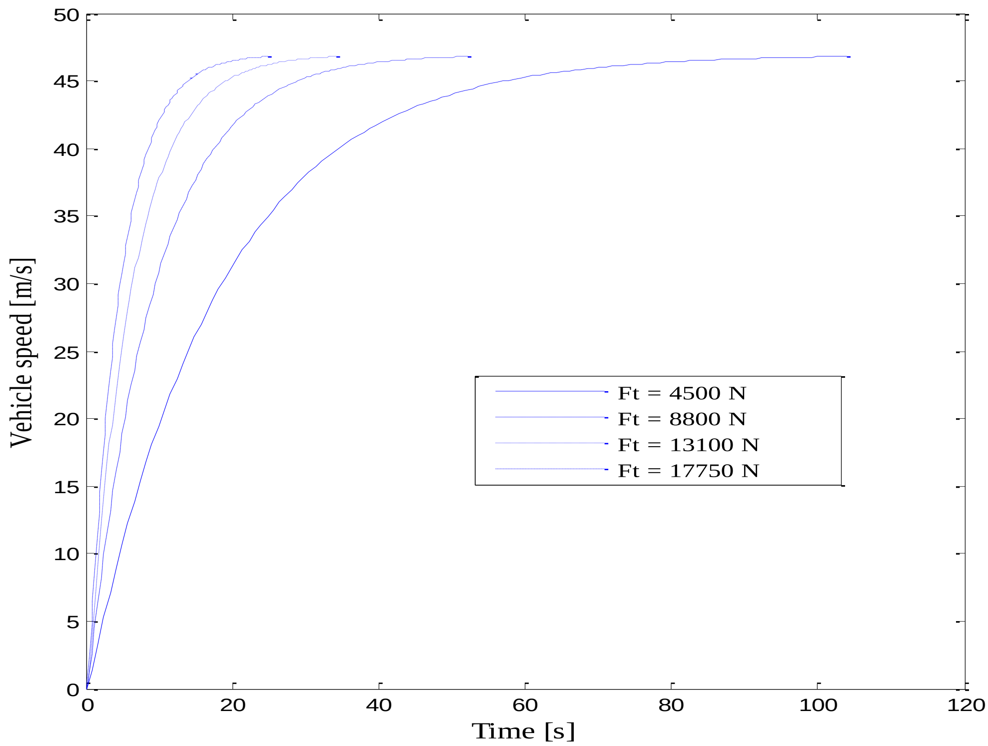

Figure 10: Mobility results of the measured segment at IRI 3.0 m/km.

CONCLUSION

This study successfully developed a mathematical model to analyze the dependence of road surface roughness on vehicle mobility, specifically considering the internal factors of tractive and pedal force The findings reveal a clear relationship between road roughness (International Roughness Index - IRI) and vehicle mobility, where increased road roughness correlates with longer travel times, reduced fuel efficiency, and lower engine revolution speed. This study also highlights the negative impacts of higher IRI values, such as discomfort, driver fatigue, increased risk of accidents, and overall reduced mobility due to the need to overcome surface irregularities. These outcomes emphasize the importance of maintaining road quality to enhance vehicle performance, reduce travel time, and improve fuel efficiency, ultimately contributing to safer and more comfortable driving conditions. The insights gained from this research can inform road maintenance practices and vehicle operation strategies, aiming to optimize mobility and minimize the adverse effects of road roughness on vehicle performance.

Recommendations

To ensure optimal vehicular mobility and road safety, it is strongly recommended that maintenance on the road surface be conducted regularly to maintain an International Roughness Index (IRI) value below 3.0 m/km. This proactive approach will help mitigate the negative impacts of road roughness on vehicular performance and overall driving experience. Furthermore, the model developed in this study provides a robust framework for analyzing the interactions between road surface conditions and vehicular mobility performance. It is suggested that this model be further extended and refined to explore additional factors influencing vehicular dynamics. By doing so, this work can serve as a foundation for the continued development of more advanced vehicular mobility models, potentially leading to significant improvements in road design, maintenance strategies, and vehicle performance.

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