Introducing the Modified Tire Power Loss and Resistant Force Regarding Longitudinal Slip

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Abstract

Investigation of vehicle resistant forces and power losses is of crucial importance owing to current state of energy consumption in transport sector. Meanwhile, considerable portion of resistant forces in a ground vehicle is traced back to tires. Pneumatic tires are known to be a source of energy dissipation as a consequence of their viscoelastic nature. The current study aims to provide a modification to tire resistance by considering the power loss in a tire due to longitudinal slip. The modified tire resistance is comprised of rolling resistance and a newly introduced resistance caused by tire slip, called slip resistance. The physical model is chosen for parameters sensitivity study since the tractive force is described in this model via tangible physical parameters, e.g. tire tangential stiffness, coefficient of friction, and contact patch length. Hence, the tire physical model is capable to investigate the influence of different parameters such as coefficient of friction, normal load, and inflation pressure on dependency of tractive force upon longitudinal slip. The results show that tractive force is a determining factor in the modified tire resistance. In addition, it is seen that behavior of the modified tire resistance undergoes qualitative changes so that for a specified tractive force, it might be a strictly decreasing function of the mentioned parameters. But as the tractive force gradually increases, it might be changed to a non-monotonic function and after that to a strictly increasing function of the above parameters. Furthermore, the results are validated by measuring the vehicle constant-speed fuel consumption experimentally.
Introduction

Current status of GHG emissions, its influence on climate change, and its increasing trend on one hand, and harvesting of non-renewable energy sources such as fossil fuels on the other hand have led to legislate several environmental laws, aiming to improve efficiency and/or reduce emissions. Transport, as one of the most energy consuming sectors, has a great potential in improving the efficiency and energy consumption as well. The generated power in combustion engine or other propulsion systems like electric motor must overcome resistive forces and relevant power loss. Hence, study of resistive forces is of great importance in improving the energy consumption in vehicles. In a well-known classification presented in [1-4], vehicle resistive forces are comprised of grade resistance, air drag, inertia, and tire rolling resistance. Accordingly, study of the resistive forces has been conducted by many contributors. For example, air drag resistance has been studied by Saab et al. [5]. Also, to reduce the inertia resistance which might be prompted by drivers’ inefficient behavior, eco-driving assistance system has been proposed [6, 7].

Tire rolling resistance includes considerable share of resistive forces, especially in low speed driving, and affects vehicle fuel efficiency [8]. About 18% and 21% of total resistive forces in gasoline passenger cars is comprised of rolling resistance in NEDC and WLTP driving cycles, respectively [9]. As one of the main sources of energy consumption, rolling resistance has been already conducted in detail. Experimental results show that rolling resistance is influenced by several parameters such as tire structure, normal load, inflation pressure, and road pavement [10]. The ongoing progress for achieving more efficient tires has led to introduce low rolling resistance tires that are claimed to deliver up to 30% reduction in rolling resistance [11]. Experiments have shown that using low rolling resistance tires can improve the constant-speed fuel consumption of a passenger car by about 0.5 lit/100 km [12]. Beside researchers’ attention to rolling resistance, governmental institutes are also involved due to determining role of road transport in energy consumption and CO2 emission which requires financial and strategic planning. For example, in NHTSA technical report [13] the effect of rolling resistance on vehicle fuel efficiency, tire traction, and tread-wear is amply investigated by testing 15 different types of tire in a standard laboratory.

Although rolling resistance is well-known as one of the influential parameters on fuel economy, recent studies have shown that tire longitudinal slip results in a power loss through driveline and also affects the vehicle fuel economy [14]. Considering this power loss is necessary to make an accurate analysis on vehicle longitudinal motion. Nevertheless, most of the studies have neglected this term by assuming that wheel purely rolls without any slip. However, this assumption may not be always true, because tire longitudinal slip inevitably occurs whenever a driving or braking torque applies to the wheel [4].

According to the literature, most of the contributors have omitted to investigate slip loss in tires. Therefore, this paper aims to study the tire slip loss to cover the research gap and to open up a topic that is hardly ever studied. To this end, after proposing a formulation for slip power loss on the basis of work and energy principle, the modified tire resistance is defined. Then, tire physical model is chosen to describe the dependency of tractive force to longitudinal slip. After that, in some specified conditions, the modified tire resistance and influence of determining parameters are discussed. Results are validated by performing some experiments on a B-class vehicle.

Slip Power Loss and Modified Tire Resistance

Tire longitudinal slip occurs whenever a torque is applied to the wheel. As a result, longitudinal speed of the wheel center differs from circumferential speed. In case of applying a driving torque as illustrated in Figure 1(b), sliding velocity at...
the contact patch is negative; it points backward. One of the most common definitions proposed for tire longitudinal slip ratio is given as following:

\[
\lambda = \frac{\nu_s - \nu_x}{r_{dyn} \omega_w} = \frac{\nu_s}{r_{dyn} \omega_w} - 1
\]

Eq. (1)

Where \( \lambda \) is longitudinal slip ratio, \( \nu_s \) is sliding velocity, \( \nu_x \) is longitudinal speed, \( \omega_w \) is wheel angular velocity, and \( r_{dyn} \) is dynamic radius of the wheel. The above equation is valid for the case of applying a drive torque to the wheel. According to the equation, when the wheel purely rolls, the longitudinal slip ratio is zero and is equal to 100 percent when the wheel spins while the vehicle is stationary. The other definition used in the literature [3] for tire slip ratio is the sliding velocity to the vehicle longitudinal speed. In this definition, the slip ratio can vary between zero and infinity which is a singularity that should be treated in numerical simulations. Another disadvantage is that it is less physically meaningful comparing to the former definition.

As a result of tire slip, the tire and ground interacting force known as tractive force is generated at the contact patch. The work done by tractive force due to tire slip is defined as the product of the tractive force and displacement of an arbitrary point at contact patch. Accordingly, the differential of the slip work is given as:

\[ \frac{dW_s}{d\lambda} = F_t \cdot dr = F_t dr \cos \pi \]

Eq. (2)

Owing to opposite directions of force and displacement vectors, the resultant work would be negative; and the slip power i.e., the derivative of the work with respect to time for a constant tractive force would result in:

\[ P_s = -F_t \nu_s \]

Eq. (3)

By substituting the slip velocity from Equation 1, the slip power would be defined as:

\[ P_s = -F_t \lambda r_{dyn} \omega_w \]

Eq. (4)

Both slip and rolling resistance powers of tire have negative signs and can be considered as dissipation or power loss. Rolling resistance power is negative because rolling resistance torque opposes driving torque, whereas negative sign in slip power loss is due to opposite directions of tractive force and sliding velocity.

Another approach to observe the need for the slip power loss to be considered in the vehicle dynamics is by looking at the power balance equation. The power flow through the vehicle powertrain is illustrated in Figure 2. The engine output power, \( P_e \), is transmitted to the wheel through the driveline. Therefore, transmitted power to the wheel, \( P_w \), can be written as:

\[ P_w = \eta P_e \]

Eq. (5)

Where \( \eta \) represents the driveline efficiency. The transmitted power to the wheel by the powertrain, after subtraction of tire loss, is used to overcome other resistive forces, i.e. air drag, grade resistance, and inertia. Therefore, the power balance equation of the wheel is as following:

\[ P_w = P_t + P_d + P_G + P_in \]

Eq. (6)

Where \( P_t \) is the tire power loss and \( P_d, P_G, \) and \( P_in \) are resistant powers due to aerodynamic drag, road gradient, and inertia, respectively and are given as following:

\[ P_d = \frac{1}{2} \rho_D c_D A_f v_{rel}^2 \nu_s \]

Eq. (7.a)

\[ P_G = mg \sin \theta_G v_s \]

Eq. (7.b)

\[ P_in = ma_x \nu_s \]

Eq. (7.c)

In the above equations, \( \rho_D \) is air density, \( c_D \) is drag coefficient, \( A_f \) is frontal area, \( v_{rel} \) is wind speed relative to vehicle body, \( m \) is vehicle mass, \( g \) is gravitational acceleration, \( \theta_G \) is grade angle, and \( a_x \) is longitudinal acceleration. The modified tire power loss comprises of rolling resistance and slip power, i.e.,

\[ P_t = P_R + P_s = (T_R + F_x \lambda r_{dyn}) \omega_w \]

Eq. (8)

Where \( P_R \) and \( T_R \) represent rolling resistance power and torque, respectively. One can also write the power balance equation through the powertrain as:

\[ P_e = (1 - \eta) P_t + P_t + P_d + P_G + P_in \]

Eq. (9)
Which means the engine power should be equal to sum of resistant powers and driveline loss. Substituting Equations 7a, 7b, and 7c into Equation 9 yields:

\[
\eta P_e = P_e + \left( \frac{1}{2} \rho_x c_D A_x v^2 + mg \sin \theta_{cg} + ma_x \right) v_x \quad \text{Eq. (10)}
\]

The left-hand side of the above equation according to Equation 5 is equal to the transmitted power to the wheel, which also can be written as following:

\[
P_e = T_w \omega_w \quad \text{Eq. (11)}
\]

Where \( T_w \) denotes the wheel torque. Using the equilibrium equation of the rotating wheel shown in Figure 1(b), the wheel torque can be expressed as:

\[
T_w = F_x r_{dy} + T_R \quad \text{Eq. (12)}
\]

Substituting Equations 1, 8, 11, and 12 into Equation 10 gives:

\[
\left( F_x r_{dy} + T_R \right) \omega_w = \left( T_n + F_x \lambda r_{dy} \right) \omega_w + \left( \frac{1}{2} \rho_x c_D A_x v^2 + mg \sin \theta_{cg} + ma_x \right) (1 - \lambda) r_{dy} \omega_w \quad \text{Eq. (13)}
\]

The tractive force namely \( F_x \) is equal to sum of resistive forces, i.e. air drag, road gradient, and inertia. Therefore, Equation 13 is identically satisfied which shows that our assumption for slip loss is correct. In addition, the modified tire resistance can be expressed as:

\[
F_x = F_x + F_{kx} = F_x + \lambda F_x \quad \text{Eq. (14)}
\]

Where \( F_x \) and \( F_{kx} \) are modified tire resistance and rolling resistance, respectively, and \( F_{kx} \) is the equivalent force of the slip loss that introduces as “slip resistance”.

**Tire Model**

In tires, the generated force at the contact patch depends on tire slip. Several theories have been proposed to interpret the dependency of tire and ground interacting force on tire slip. In the physical model, the interacting force is explained via tangible physical parameters like tire stiffness, coefficient of friction, contact patch length, and normal load. But extension of the semi-empirical or experimental tire models to study the changes in physical parameters requires time-consuming experiments as well as expensive test equipment. In the current study, the physical model is chosen to investigate the dependency of interacting force on tire slip due to its capability of considering the influence of parameters variations.

In the physical model, the contact patch is divided into adhesion and sliding regions. The interacting force is determined by tire elasticity in adhesion region and by tire/ground adhesive properties in sliding region. A characteristic length, \( x_c \), as illustrated in Figure 3, is defined such that for the adhesion region \( x \leq x_c \), and for the sliding region \( x > x_c \).

![FIGURE 3 Stress distribution along the contact patch.](image)

Therefore, the stress distribution on the contact patch can be given as following [15]:

\[
\sigma(x) = \begin{cases} 
  k_x \lambda (a - x) & a > x > x_c, \\
  \mu_z q_z & x_c > x > -a 
\end{cases} \quad \text{Eq. (15)}
\]

Where \( k_x \) is tire longitudinal stiffness, \( \mu_z \) is longitudinal coefficient of friction, and \( q_z \) represents the normal force distribution. In fact, the normal force distribution has a complicated shape and depends on various parameters, but can be approximated via a parabolic distribution given by Equation 16 which is being suggested in several studies, e.g. in [15, 16].

\[
q_z(x) = \frac{3F_z}{4a} \left( 1 - \left( \frac{x}{a} \right)^2 \right) \quad \text{Eq. (16)}
\]

It is further assumed that normal load is uniformly distributed along the width of the contact patch and the shape of the contact patch is rectangular. However, the contact patch shape of radial tires is very close to a rectangle and this assumption is almost true in reality [17].

According to Figure 3 and Equation 15, the stress in the adhesion region correlates linearly to \( x \) with a slope equal to \( k_x \lambda \). Thus, the stress increases along the contact patch until it approaches the frictional stress distribution. This is the point at which sliding occurs because the stress in adhesion region becomes larger than that of sliding region. Accordingly, the sliding condition can be given by Equation 17 which is used to find the characteristic length described in Equation 18.

\[
k_x \lambda (a - x) \geq \frac{3F_z}{4a} \left( 1 - \left( \frac{x}{a} \right)^2 \right) \quad \text{Eq. (17)}
\]

\[
x_c = \frac{4a^2 k_x \lambda}{3\mu_z F_z} - a \quad \text{Eq. (18)}
\]

By substituting Equation 18 into the stress distribution given by Equation 15 and integrating along the contact
patch, from \(-a\) to \(a\), the tire/ground interacting force would be obtained as:

\[ F_x = 2a^2k_x\lambda \left( 1 - \frac{2\lambda^*}{\lambda} \right) + \mu_p F_z \left( \frac{\lambda^*}{\lambda} \right)^2 \left( 3 - 2\frac{\lambda^*}{\lambda} \right) \]  
Eq. (19)

Where \(\lambda^*\) represents the characteristic slip ratio at which the maximum longitudinal force is generated and is given as following:

\[ \lambda^* = \frac{3\mu F_z}{2a^2k_x} \]  
Eq. (20)

As slip ratio goes further than the characteristic slip ratio \(\lambda^*\), the longitudinal coefficient of friction \(\mu_x\) gradually decreases and finally at \(\lambda = 100\%\) becomes equal to a so-called sliding coefficient of friction \(\mu_s\). This phenomenon could be roughly expressed via a linear approximation. Thus, the tire/ground interacting force would be defined as:

\[ F_x = \begin{cases} 2a^2k_x\lambda \left( 1 - \frac{2\lambda^*}{\lambda} \right) + \mu_p F_z \left( \frac{\lambda^*}{\lambda} \right)^2 \left( 3 - 2\frac{\lambda^*}{\lambda} \right), & \lambda \leq \lambda^* \\ F_z \left( \mu_p + \frac{\mu_s - \mu_p}{\lambda^* - 100\%} \right) (\lambda^* - \lambda), & \lambda > \lambda^* \end{cases} \]  
Eq. (21)

Figure 4 illustrates the longitudinal force given by Equation 21 as a function of slip ratio \(\lambda\). The peak and sliding coefficient of friction are assumed to be 1 and 0.75, respectively. Other parameters used in the tire model are defined in Table 1. For \(\lambda > \lambda^* = 16\%\) simple linearization does not accurately expose the physical phenomenon, though this simplification does not affect the results because in the current study it is assumed that the slip ratio does not go further than the characteristic slip ratio. In practice, slip ratio becomes larger than the characteristic ratio in the case of transferring a higher drive torque than the maximum frictional torque, i.e., \(T_w > \mu F_s F_{zw}\). However, this case does not occur in normal driving conditions and therefore is excluded.

To obtain the slip ratio \(\lambda\) for a given tractive force, \(F_s\), the method proposed by Oftadeh et al. [18] is used which converges faster than commonly used Newton–Raphson method. The proposed iterative equation to find the roots of an arbitrary function \(F(x)\) is given as following:

\[ x_{n+1} = x_n \left( 1 - k \frac{F(x_n)}{F'(x_n)} \right)^{1/k} \]  
Eq. (22.a)

\[ k = 1 + \frac{x_n F'(x_n)}{F''(x_n)} \]  
Eq. (22.b)

Where \(F'(x)\) and \(F''(x)\) denote the function’s first and second derivations, respectively and \(n\) is the number of iteration.

### Results and Discussion

Characteristics of the tire used in simulations are shown in Table 1. The modified tire resistance described in Equation 14 is influenced by several parameters such as tractive force, coefficient of friction, normal force, and contact patch length, which itself is affected by normal force, vertical stiffness, and tire inflation pressure. But in the present study only the influence of coefficient of friction, inflation pressure, and normal load on the modified tire resistance is considered for some specified tractive forces described as a coefficient of the maximum frictional force, namely \(\mu_p F_z\).

### Coefficient of Friction

The dependency of tire resistant forces to coefficient of friction is illustrated in Figure 5. Bearing in mind that the tractive force cannot be larger than the frictional force, the middle and bottom curves begin at \(\mu = 0.4\) and \(\mu = 0.7\), respectively. As coefficient of friction increases, the slip resistance reduces due to decrease of tire longitudinal slip, but rolling resistance remains constant since the coefficient of friction per se does not affect the tire deformation work. Also, it is evident that a larger tractive force leads to a considerable increase of slip resistance. The slip resistance is much lower than the rolling resistance when the tractive force is small (0.1\(\mu_p F_s\)), is almost equal to rolling resistance as tractive force increases to 0.4\(\mu_p F_s\), and finally becomes much larger than rolling resistance when the tractive force is equal to 0.7\(\mu_p F_s\).

### Table 1: Tire characteristics.

<table>
<thead>
<tr>
<th>Characteristic</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Tire marking</td>
<td>165/65 R 13</td>
</tr>
<tr>
<td>Static normal force</td>
<td>2500 N</td>
</tr>
<tr>
<td>Recommended inflation pressure</td>
<td>200 kPa</td>
</tr>
<tr>
<td>Longitudinal stiffness</td>
<td>2300 kN/m²</td>
</tr>
<tr>
<td>Rolling resistance coefficient</td>
<td>0.008</td>
</tr>
<tr>
<td>Vertical stiffness</td>
<td>120 kN/m</td>
</tr>
</tbody>
</table>
Inflation Pressure

Change of inflation pressure affects tire deformation work as well as contact patch length. As tire inflation pressure increases, rolling resistance reduces due to a smaller deformation work and simultaneously the tire slip increases as a result of contact patch shrinkage. Length of tire contact patch can be obtained using the Pythagorean Theorem and the tire radius. For a rolling wheel, contact patch length is determined by the dynamic radius, a speed-dependent radius that is larger than the static radius as a result of centrifugal force. Dynamic radius would be obtained by dividing the vehicle longitudinal slip by the angular velocity of a freely rolling tire, and for the tested tire, it is reported to be about 260 mm [8]. However, the effect of speed increase on the dynamic radius, which is less than 0.2% for speeds up to 110 km/h, is not considered here. In addition, it is assumed that the variation of the dynamic radius at different inflation pressures is the same as the variation of the static radius shown in Figure 6.

Previous studies have shown that a power function can predict the rolling resistance coefficient as a function of inflation pressure [19]. For the tested tires used in this study, the dependency of rolling resistance coefficient to inflation pressure is given as:

\[
\frac{f_R}{f_{R0}} = p_{norm}^{\alpha_2} \quad \text{Eq. (23)}
\]

In the above equation, \(f_{R0}\) is rolling resistance coefficient at the recommended inflation pressure and \(p_{norm}\) is normalized inflation pressure, i.e. ratio of the inflation pressure to the recommended inflation pressure.

Tire resistance as a function of normalized inflation pressure is illustrated in Figure 7. When the tractive force is low, rolling resistance is the dominant resistance and as a result, the modified resistance is a decreasing function of inflation pressure, while in the case of a large tractive force, the slip resistance is dominant and the modified resistance becomes an increasing function of inflation pressure.

**FIGURE 5** Influence of coefficient of friction on modified tire resistance.

**FIGURE 6** Tire static radius at different inflation pressures.
However, the modified tire resistance is not always a monotonically increasing/decreasing function of inflation pressure, as it is shown in Figure 8 for a tractive force equal to $0.1 \mu_p F_z$. If other parameters, i.e., normal load, coefficient of friction, tire characteristics, are maintained, when tractive force varies in the range of $(0.1 \mu_p F_z, 0.2 \mu_p F_z]$, the modified tire resistance is not monotonic. Also, the minimum point moves backward along the $x$-axis, normalized inflation pressure, as tractive force increases.

Normal Load

In previous studies, the rolling resistance has been proved to be a strictly increasing function of normal load. But normal load increment has a contradictory influence on slip resistance, since it increases the contact patch length, which together with normal load increase, leads to a reduced slip resistance. Although the rolling resistance coefficient does not remain constant as normal load changes, experiments have shown that normal load change has a minor influence on rolling resistance coefficient in radial tires [10]. Accordingly, the impact of normal load on rolling resistance coefficient has been neglected in the current study.

The modified tire resistance for three different tractive forces is shown in Figure 9. Increase of normal force reduces the slip resistance which consequently decreases modified tire resistance, provided that slip resistance is dominant. A small tractive force diminishes the slip loss in a way that in this case the minimum tire resistance is achieved at the point that rolling resistance meets its minimal. In a fairly large tractive force, shown in the middle image of Figure 9, the change of slip resistance is counterbalanced by the rolling resistance change in a wide range of normalized normal forces, between 0.8 and 1.5. Because it is not physically possible for the tractive force to be larger than the frictional force, when the tractive force is equal to $0.7 \mu_p F_z$, the tire resistance curves can only be drawn in an interval from 0.7 to 1.5.

Constant-Speed Fuel Consumption

Several approaches have been performed to experimentally evaluate the tire power loss, from a laboratory test to a real-world fuel consumption test. The latter one, which is conducted in this study, could be more challenging owing to numerous uncertainties comprise of real-world conditions, but is of major importance too. Furthermore, in most of test benches,
tires are placed on cylindrical surfaces and consequently the normal force distribution at the tire contact patch differs from that on a flat surface which is assumed prior to deriving the tire longitudinal force relation given by Equation 21.

Experiments are done at an urban highway with an average slope of 4%, on a B-Class sedan with specifications given in Table 2. To observe the effect of tractive force increment, experiments are carried out in both directions of the test route namely ascent and descent paths. Prior to data acquisition, tires have been warmed up in order to exclude the effect of tire temperature on the measurements. Then, as the inflation pressure of the driven wheels are held at the recommended pressure, by changing the inflation pressure of the drive wheels, the consumed fuel in the test route is measured by means of a flowmeter. Also, a GPS device is utilized to capture the travelled distance and the vehicle speed. All experiments are performed in one day with stable weather conditions and therefore the change of air density and wind speed, that are determining factors in aerodynamic force, could be neglected. The changing rate of accelerator pedal depression is controlled to be insignificant, so that tests are conducted under the quasi-steady state conditions. In addition, to obtain reliable results and eliminate the experimental error, each test is repeated twice and results are averaged.

Fuel consumption of the vehicle for different scenarios is shown in Figure 10. As vehicle speed and/or road slope increases, fuel consumption routinely increases due to a larger resistant force, i.e. aerodynamic drag and/or grade resistance. Since in each scenario, power losses due to air drag and road grade are almost constant and the tire power loss is the only variable, the influence of tire inflation pressure on constant-speed fuel consumption qualitatively represents the tire power loss trend. As illustrated in Figure 10, on the descent path, an increase in the inflation pressure would result in a slight decrease in the fuel consumption. This is rooted from the fact that when the tractive force is moderately low, rolling resistance is dominant at the drive wheels which would be reduced as tire inflation pressure increases. Increment of tractive force on the ascent path would increase the slip resistance, and the tire resistance as shown in Figure 8 changes to a non-monotonic trend. As a result, contrary to observations on the descent path, the fuel consumption is minimized at a lower pressure than the highest one, i.e. in cruising at 110 km/h and 90 km/h the minimum fuel consumption is measured at normalized inflation pressures of 0.75 and 1, respectively.

In the case of 70 km/h cruising on the ascent path, the measured fuel consumption remains almost constant and it does not undergo a considerable change at different inflation pressures.

**TABLE 2** Vehicle specifications.

<table>
<thead>
<tr>
<th>Specification</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Vehicle classification</td>
<td>B-Class</td>
</tr>
<tr>
<td>Curb weight</td>
<td>870 kg</td>
</tr>
<tr>
<td>Engine type</td>
<td>SI - 4Stroke</td>
</tr>
<tr>
<td>Engine capacity</td>
<td>1323 cc</td>
</tr>
<tr>
<td>No. of cylinders</td>
<td>4</td>
</tr>
<tr>
<td>Max. power</td>
<td>61 hp @ 5000 rpm</td>
</tr>
<tr>
<td>Max. torque</td>
<td>103 Nm @ 2800 rpm</td>
</tr>
<tr>
<td>Drive wheels</td>
<td>Front</td>
</tr>
</tbody>
</table>
Conclusions

In the present paper, a modified tire resistance is proposed by considering the longitudinal slip. A newly defined term due to tire slip, called slip resistance, is added to the well-known term namely rolling resistance. It is shown that normal force and inflation pressure have contradictory effects on slip and rolling resistances. In addition, the tractive force determines the trend of tire resistance change; such that when the tractive force is moderately low, the modified tire resistance is highly affected by the rolling resistance, whereas increasing the tractive force would gradually dominate slip resistance. Moreover, increasing of tractive force leads to a larger tire resistance. The conducted experiments on fuel consumption in constant-speed situation qualitatively validate the developed modification to the tire resistance as well as the trend of modified tire resistance change as a function of inflation pressure.

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Nomenclature

- \( a \) - longitudinal distance from wheel center to contact patch edge, m
- \( a_l \) - Longitudinal acceleration, m/sec\(^2\)
- \( A \) - area, m\(^2\)
- \( c_D \) - drag coefficient, –
- \( f \) - coefficient, –
- \( F \) - force, N
- \( F(.) \) - function, –
- \( g \) - gravitational acceleration, m/sec\(^2\)
- \( k \) - tire stiffness, N/m\(^2\)
- \( m \) - vehicle mass, kg
- \( p \) - tire inflation pressure, kPa
- \( P \) - power, W
- \( q \) - force distribution, N/m
- \( r \) - radius, m
- \( r \) - displacement vector, m
- \( T \) - torque, Nm
- \( v \) - velocity, m/sec
- \( W \) - work, J
- \( \eta \) - driveline efficiency, –
- \( \lambda \) - slip ratio, –
- \( \mu \) - coefficient of friction, –
- \( \theta \) - road angle, rad
- \( \rho_a \) - air density, kg/m\(^3\)
- \( \sigma \) - stress distribution, N/m
- \( \omega \) - angular velocity, rad/sec

Superscripts

* - characteristic
Subscripts
0 - initial
a - air drag
c - characteristic
dyn - dynamic
e - engine
f - frontal
G - road grade
in - inertia
n - number of iteration
norm - normalized
p - peak
rel - relative
R - rolling resistance
S - sliding or slip
t - tire
w - wheel
x - longitudinal
z - vertical

Abbreviations
GHG - greenhouse gases
GPS - global positioning system
NEDC - new European driving cycle
NHTSA - national highway traffic safety administration
WLTP - worldwide harmonized light vehicles test procedure

References

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