INFLUENCE OF THE TIRES PRESSURE IN THE VEHICLE FUEL CONSUMPTION

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Abstract: One of the segments into the vehicle dynamics is the longitudinal dynamics that works in the calculus of vehicle power consumption to attend a specific route. It estimates, by the equations, the forces acting on the system such as aerodynamic drag and rolling resistance as well as factors related to the road grade and driver behavior. This paper aims to study the influence of the tires pressure. The rolling resistance force is essentially caused by the tires deformation and the adherence phenomenon in the contact, it can be calculated in function of some factors such as: tires structure, tires geometry, tires material, temperature and filling pressure. At low speeds and on hard pavement, rolling resistance is the primary resistance force of the movement. The floor irregularities also cause influence in the rolling resistance, but the tires deformation is the most influential factor. There is a variety of tabulated values to estimate the rolling resistance, however they do not change with the vehicle speed. Based on experimental results, empirical equations were developed to calculate the rolling resistance. This paper aims to study the influence of the tires pressure in the calculation of the vehicle power required, according to the equation proposed by the literature and its effect on fuel consumption. The analysis were performed through co-simulation between the multibody dynamics program Adams$^{TM}$ and Simulink/Matlab$^{TM}$, where the power demand was defined based on the Brazilian urban standard driving cycle NBR6601, together with the equations of motion resistance forces.

keywords: Longitudinal Vehicular Dynamics, Fuel Consumption, Tires pressure, Co-simulation

1. INTRODUCTION

The vehicular dynamic studies and analyzes the interactions between the vehicle, the driver and the environment as well as load reactions involved. The literature proposes to divide the vehicular dynamic into three areas: longitudinal, lateral and vertical.

The longitudinal dynamics is responsible for calculating the vehicle power consumption required so that it can fulfill a path, estimating by means of equations: the forces acting on the system, the aerodynamic drag and the tire-ground interaction, factors related to the inclination angle and driver behavior.

The incorrect inflation pressure in tire affects vehicle handling, passenger comfort and braking conditions, as well it reduces fuel efficiency and tire life (Hamed et al., 2013a,b). According to Szabó et al. (2010), by keeping the recommended tire pressure, the vehicle maintains an optimal output of tires as well as optimal fuel economy.

The loss of hysteresis energy could be reduced by increasing the tire air pressure, but that would also decrease driving comfort and might reduce the grip to the road and thus driving safety Holmberg et al. (2012). The longitudinal traction or braking properties of tires are also dependent on the inflation pressure (Al-Solihat et al., 2010). The rolling resistance hardly influences the handling properties on the vehicle, and it represents a major part in fuel consumption (Rill, 2011).

Castillo et al. (2006) in the study of the contact patch provided by the bench makes it possible to characterize tire behavior under different loading states, inflation pressure, tire defects and toe and camber angles. Taghavifar and Mardani (2013) evaluate the effects of velocity, tire inflation pressure, and vertical load of tractor’s wheel on rolling resistance in a controlled condition using a single-wheel tester and a soil bin. Hernandez et al. (2013) studied the effect of applied load and tire-inflation pressure on the variation of longitudinal, transverse, and vertical contact stresses along the contact length for two types of tires used by the truck industry.

This paper aims to study the influence of the tires pressure in the calculation of the vehicle power required, according to the equation proposed by the literature and its effect on fuel consumption. The simulations are performed by a multibody dynamic analysis software Adams$^{TM}$ (Automated Dynamic Analysis of Mechanical Systems), with Matlab/Simulink$^{TM}$ where are implemented the equations proposed in the literature.
2. VEHICLE LONGITUDINAL DYNAMICS

In this paper it will be used the longitudinal vehicle dynamics methodology proposed by Gillespie (1992) where the model is based on the acting forces on the vehicle travel direction as shown in Fig. 1.

![Arbitrary forces acting on a vehicle adapted from Gillespie (1992)](image)

2.1 Aerodynamic drag

The aerodynamic load ($D_A$) is the resistance imposed by the air during the vehicle passage, this effect is proportional to the square of the vehicle speed. According to Ehsani et al. (2009), a vehicle traveling at a particular speed in air, generates a resistance force of its motion. This force is known as aerodynamic drag and it is resultant from two components: shape drag and skin friction.

Due to the complexity of the airflow outside the vehicle, this load is based on empirical constant and a term known as drag coefficient, as shown in Eq. (1).

$$D_A = \frac{1}{2} \rho V^2 C_D A \quad (1)$$

Where $\rho$ is the air density [kg/m$^3$], $V$ is the vehicle speed [m/s], the term $A$ refers to the frontal area of the vehicle and $C_D$ is the drag coefficient obtained empirically in function of the vehicle geometry as show at Fig. 2.

<table>
<thead>
<tr>
<th>Vehicle type</th>
<th>Coefficient of aerodynamic resistance</th>
</tr>
</thead>
<tbody>
<tr>
<td>Open convertible</td>
<td>0.5...0.7</td>
</tr>
<tr>
<td>Van body</td>
<td>0.5...0.7</td>
</tr>
<tr>
<td>Ponton body</td>
<td>0.4...0.55</td>
</tr>
<tr>
<td>Wedged-shaped body; headlamps and bumpers are integrated into the body, covered underbody, optimized cooling air flow</td>
<td>0.3...0.4</td>
</tr>
<tr>
<td>Headlamp and all wheels in body, covered underbody</td>
<td>0.2...0.25</td>
</tr>
<tr>
<td>K-shaped (small breakaway section)</td>
<td>0.23</td>
</tr>
<tr>
<td>Optimum streamlined design</td>
<td>0.15...0.20</td>
</tr>
</tbody>
</table>

Due to the complexity of the airflow outside the vehicle, this load is based on empirical constant and a term known as drag coefficient, as shown in Eq. (1).

$$D_A = \frac{1}{2} \rho V^2 C_D A \quad (1)$$

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![Drag coefficients $C_D$ for different vehicles (Ehsani et al., 2009)](image)
2.2 Rolling resistance

Rolling resistance is a result of energy loss in the tire, which is associated to the deformation of the area of tire contact and the damping properties of the rubber. These lead to the transformation of mechanical into thermal energy, contributing to warming of the tire (Reimpell and Stoll, 1996).

At low speeds on hard pavement, rolling resistance ($R_x$) is the primary resistance load caused essentially by: the tire deformation, the pavement and the tire adhesion on the ground. This paper will consider hard surfaces, such as asphalt and concrete. In these cases, the ground stiffness is higher than the tires, therefore the road can be considered undeformable. The rolling resistance is shown by the Eq. (2).

$$R_x = f_r M g$$  \hspace{1cm} (2)

Where $M$ is the vehicle mass [kg], $g$ is the gravitational acceleration $m/s^2$ and $f_r$ represents the rolling resistance coefficient. Usually the rolling resistance coefficient is given by constants depending on the tire type and pavement, or by equations based on vehicle speed $V$ like the Eq. (3).

$$f_r = 0.01 \left( 1 + \frac{0.62 V}{100} \right)$$  \hspace{1cm} (3)

At the same time, many other aspects also affect the rolling resistance coefficient, like vehicle’s weight, type of tire and its pressure of inflation, soil stiffness, temperature and residual braking (Gillespie, 1992; Heißing and Ersoy, 2010; Genta, 1997).

A general equation for is proposed by Genta (1997) taking into account the vehicle’s weight, tire type and pressure, described by Eq. (4).

$$f_r = \frac{K}{1000} \left( 5.1 + \frac{5.5 \times 10^5 + 90 M g \cos \Theta}{p} + \frac{1100 + 0.0388 M g \cos \Theta}{p} + V^2 \right)$$  \hspace{1cm} (4)

Where $K$ is a constant in function of the tire type (0.8 for radial and 1 for non-radial), and $p$ is the tire inflation pressure.

2.3 Road grade influence

This term refers to the weight force decomposition resulting from the road grade. In uphill, the weight force component acts retarding the vehicle movement, and in downhill, the weight force aids the movement.

The grade angle also results in a component of weight parallel to the ground, which as in the case of accelerating or braking on flat ground, results in a longitudinal weight transfer. The effects of grade and longitudinal acceleration can be combined in finding the changes in front and rear loads due to both (Milliken et al., 1995).

2.4 Acceleration performance

The vehicle acceleration generates resistance forces as the vehicle longitudinal displacement as the powertrain rotational inertia. The available traction force ($F_x$) in function of the engine torque and the transmission ratio is given by Eq. (5) proposed by Gillespie (1992).

$$F_x = \frac{T_e N_{fj}}{r} - ((I_e + I_t) N_{fj}^2 + I_d N_{fj}^2 + I_w) \frac{a_x}{r^2}$$  \hspace{1cm} (5)

- $T_e = $ Available engine torque [Nm];
- $N_{fj} = $ Total gear ratio;
- $\eta_t = $ Transmission overall efficiency;
- $r = $ Tire external radius [m];
- $I_e = $ Gearbox inertia [kgm$^2$];
- $I_t = $ Gearbox transmission ratio;
- $I_d = $ Differential inertia [kgm$^2$];
- $I_w = $ Wheels and tires inertia [kgm$^2$];
- $a_x = $ Vehicle longitudinal acceleration [$m/s^2$].

The vehicle acceleration performance is given by the Eq. (6):

$$M a_x = \frac{W}{g} a_x = F_x - R_x - D_A - W \sin(\Theta)$$  \hspace{1cm} (6)

Where $M$ is the vehicle mass [kg] and $\Theta$ the road grade [rad].
Joining the Eq. (5) with Eq. (6) and isolating the engine torque \( T_e \) is possible to estimate the vehicle drive required torque in a predetermined situation.

\[
T_e = \frac{M_{a_0} + \left( (I_e + I_t) N_f^2 + I_d N_f^2 + I_w \right) a_r + R_x + D_A + W \sin(\theta)}{N_f \eta_f} r 
\] (7)

The maximum performance in longitudinal acceleration of an engine vehicle is determined by one of two limits: engine power or traction limits on the drive wheels. Which limit prevails may depend on vehicle speed. At low speeds, tire traction may be the limiting factor, whereas at high speeds engine power may account for the limits (Gillespie, 1992).

The maximum contact force transmitted by the tire \( F_{\text{max}} \) is given by the Eq. (8), assuming a locked differential simplified model.

\[
F_{\text{max}} = \frac{\mu W_f}{1 + \frac{h}{L} \mu} 
\] (8)

- \( \mu \) = Peak coefficient of friction;
- \( W_f \) = Weight force acting on the front axle \([N]\);
- \( L \) = Wheelbase \([m]\);
- \( h \) = Vehicle gravity center height \([m]\).

3. DRIVING CYCLE

With the intention to establish a benchmark, standard cycles are utilized to determine the vehicle speed behavior, in a way that the mathematic model calculates the vehicle required power to follow the velocity profile predetermined by the cycle. A driving cycle represents the way that the vehicle is driven during a trip and also the road characteristics. In the simplest case, it is defined as a sequence of vehicle speed (and therefore acceleration) and road grade (Corrêa et al., 2011).

These driving cycles are designed to be representative of urban and extra-urban driving conditions, and they reproduce measures of vehicle speed in real roads. Some of them and the test procedures have been recently updated to better suit modern vehicles, following criticism towards the previous regulation (Serrão et al., 2005).

Even with the current improvements, the regulatory cycles should be considered a comparison tool rather than a prediction tool. In fact, it is not possible to predict how a vehicle will be driven, since each vehicle has a different usage pattern and each driver his or her own driving style. In order to obtain more realistic estimations of real-world fuel consumption for a specific vehicle, vehicle manufacturers may develop their own testing cycles (Corrêa et al., 2013).

The vehicle motion is restricted mainly by two forces, aerodynamic loads and rolling resistance. At low speeds and rigid pavement, rolling resistance is the primary resistance movement force (Gillespie, 1992). Because the aim of this paper is to evaluate the influence of the tires inflation pressure in the vehicle longitudinal dynamics, will use a standard urban driving cycle, where the vehicle remains at low speeds in most of the route in order to show the rolling resistance influence, maximizing its effect compared to the aerodynamic drag that has more influence at high speeds.

In the simulations was used the NBR6601 velocity profile proposed by ABNT (2005) representing the Brazilian urban driving cycle Fig. 3 with 12 km, average speed of 32 km/h and 91.2 km/h maximum speed. The vehicle remains stationary for 17.2% of the time, and the cycle does not include road grade information.
In real driving conditions, the vehicle stopped time can be greater than that shown in the standard velocity profile. It does not represent a real use conditions, anyway, the standard velocity profile can be used as a means of comparison between available technological solutions (Souza, 2010).

4. ENGINE TORQUE CURVES AND FUEL CONSUMPTION MAP

The simulated model considers the engine torque curves Fig. 4, that represents the real engine torque in function of the acceleration percentage and the engine speed, based on experimental results for a vehicle similar to the simulated.

The required torque calculated by the Eq. (7) is compared with the available engine torque from the curves of Fig. 4, if the required torque exceeds the maximum torque available, the simulation will use the maximum torque of the curves and there will be loss in the vehicle acceleration performance.

\[
C_l = C_e \cdot \frac{\text{Pot} \cdot dt}{\rho_c}
\]  

- \(C_e\) = Fuel specific consumption obtained from the consumption map
- \(\text{Pot}\) = Current engine power
- \(\rho_c\) = Fuel density (in this paper its used gasoline \(\rho_c = 754.2 \text{ kg/m}^3\))
5. DYNAMIC CO-SIMULATION

According to Oliveira (2005) the longitudinal dynamics simulation is used to compare the importance of vehicles energy balance characteristics, analyzing different propulsion concepts, but without the need to build prototypes that require high cost and time. The co-simulation technique is used in a development stage where the physical or mathematical mechatronic and control system are designed Brezina et al. (2011). Al-Hammouri et al. (2007), state that the co-simulation platform must support the communication between the softwares, being that the main technical difficulty is synchronizing the used programs in both directions. Hines and Borriello (1997), assert that the co-simulation allows a high detailing degree, keeping a good performance, however, models too much detailed tends to incur a high computational cost, so the simulation should provide only the needed information. In the automotive area, Kim et al. (2008) used co-simulation to optimize a vehicle stability control algorithm for a four wheel drive hybrid electric vehicle.

5.1 Adams™ model

The simulations implemented in this paper were made via the multibody dynamic analysis program Adams™ (Automatic Dynamic Analysis of Mechanical Systems), where is implemented the vehicular model analyzed. The control of variables related to longitudinal dynamics, as described earlier, is performed through the interface between Adams™ and Matlab/Simulink™.

The simulated vehicle was based on a compact hatchback equipped with 1.0L engine (Tab. 1).

<table>
<thead>
<tr>
<th>Components</th>
<th>Units</th>
<th>Speed 1st</th>
<th>Speed 2nd</th>
<th>Speed 3rd</th>
<th>Speed 4th</th>
<th>Speed 5th</th>
</tr>
</thead>
<tbody>
<tr>
<td>Engine inertia</td>
<td>kgm²</td>
<td>0.1367</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Transmission inertia</td>
<td>kgm²</td>
<td>0.0017</td>
<td>0.0022</td>
<td>0.0029</td>
<td>0.0039</td>
<td>0.0054</td>
</tr>
<tr>
<td>Transmission ratio</td>
<td>-</td>
<td>4.27</td>
<td>2.35</td>
<td>1.48</td>
<td>1.05</td>
<td>0.8</td>
</tr>
<tr>
<td>Differential inertia</td>
<td>kgm²</td>
<td>9.22E-04</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Differential ratio</td>
<td>-</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>4.87</td>
</tr>
<tr>
<td>Wheels + tires inertia</td>
<td>kgm²</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>2</td>
</tr>
<tr>
<td>Vehicle mass</td>
<td>kg</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>980</td>
</tr>
<tr>
<td>Tires</td>
<td>-</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>175/70 R13</td>
</tr>
</tbody>
</table>

The implemented model was designed based on a dynamometer bench, to enable future experimental validations. The effects of vehicle suspension system were neglected to simplify the model and also because these factors are disregarded by the current literature. The model consists of two rolls set to simulate the longitudinal displacement inertia, in which four cylinders representing the vehicle’s wheels are supported. The CAD model was exported to Adams™, where an appropriate revolution joints were created to allow the wheels movement and rotating masses, as shown in Fig. 6.

On the wheels were applied torques related to the power supplied by the powertrain and the brake system. In the rotating masses, were applied a movement resistance torque. In the model, the vehicle chassis was connected to the base to prevent longitudinal movement so that the wheels remain aligned with the rollers. The rotational movement between the rollers and the wheels are done by means of a joint, transmitting torques and acting speeds.
5.2 Matlab/Simulink\textsuperscript{TM} model

To facilitate the implementation of the vehicle dynamics equations, it was used a Simulink\textsuperscript{TM}/Adams\textsuperscript{TM} interface, generating a block of data from the dynamic model as shown in Fig. 7.

The Simulink\textsuperscript{TM} programmed algorithm works together with the Adams\textsuperscript{TM} solver. The Simulink\textsuperscript{TM} provides for Adams\textsuperscript{TM} torque values applied in the wheels and in the rotating masses. The Adams\textsuperscript{TM} generates a response from an angular velocity of the wheels, which supplies the Simulink\textsuperscript{TM} algorithm to recalculate the required torque according to the new demand.

The Fig. 8 shows the Simulink\textsuperscript{TM} model. The Orange Block corresponds to the Adams\textsuperscript{TM} dynamic model blocks shown in Fig. 7 which provide the simulated wheels angular velocity of the vehicle that allows the simulation block 1 calculates the vehicle longitudinal speed in function of the tire external diameter and the wheels angular velocity provided by the Adams\textsuperscript{TM} block.

The block 2 represents the vehicle powertrain where is set the gear ratio in function of the vehicle speed. This parameter is very important due to the transmission ratio changes the system equivalent inertia, therefore changing the engine power demand. In this paper was used the gear shifting speeds proposed by GM (2013) as a parameter to determine when the gear shifting will occurs.
In block 3 is determined the vehicle required acceleration, comparing the current vehicle speed with the standard velocity profile (Fig. 3). It used the cycle speed in the next simulation step to create a power demand to be enough to reach the cycle velocity requested speed when the simulation fulfill simulation step. The required acceleration \( a_r \) is given by the Eq. (10) where \( v_r \) is the cycle required speed, \( v_c \) is the vehicle current speed and \( \Delta t \) is the simulation step.

\[
a_r = \frac{v_r - v_c}{\Delta t}
\]  

In the block 4 is calculated the movement resistance forces using the Eq. (1) to determine the aerodynamic drag and the Eq. (2) that determines the rolling resistance applying the Eq. (4) to determine the rolling resistance coefficient \( f_r \) in function of the tire type and inflation pressure. Because the NBR6601 standard does not provide information about the road altimetry the inclination angle \( \Theta \) is considered null.

After defining the motion resistance forces, the required acceleration and the gear ratio is possible to apply the Eq. (7) to determine the engine required torque for the vehicle reaches the desired speed at the simulation step end.

In the block 5 are located the engine torque curves (Fig. 4) and the fuel consumption map (Fig. 9). Depending of the required torque is determined the engine acceleration percentage, and this torque is sent to the Adams\(^{TM}\) model. If the required torque overcomes the maximum engine available torque, it indicates that the power demand is greater than the supplied by the engine in the 100% acceleration regime, therefore will be sent the maximum available torque, which can cause a vehicle performance decrease. The specific fuel consumption \( C_e \) can be determined by the consumption map FIG as a function of the engine speed and torque by Eq. (9).

In the block 6 is determined if the vehicle is on an accelerating or braking process. If the required torque from block 4 is positive, the vehicle is accelerating and the torque from the engine curves from the block 5 multiplied by the transition ratio from the block 2 is apply to the Adams\(^{TM}\) model.

The block 6 determines if the vehicle is on accelerating or braking process. If the required torque from block 4 is positive, the vehicle is accelerating and the Adams\(^{TM}\) model receives the engine/powertrain torque from the block 5.

If the required torque is negative it indicates that the vehicle is on in braking mode. In this paper is used an ideal braking model, that apply exactly the required braking torque to vehicle wheels at Adams\(^{TM}\) model. This not influence the fuel consumption due to the fact that when the engine throttle is not requested this operates in a cutoff regime that interrupt the fuel injection, making the fuel consumption null until a new engine throttle.

Still at the block 6 is applied the Eq. (8) that calculates the tire traction limit. If the vehicle wheels applied torque overcomes the traction limit, the torque limit is used, changing the vehicle acceleration, therefore is necessary recalculate the power demand in the block 4 to adjust the torque required and the resistance forces to current acceleration condition. This procedure also corrects the vehicle acceleration when the required torque overcome the engine available torque as described previously.

After the convergence between the required and the available acceleration considering the tire traction limit and the available engine power, is sent to the Adams model block the resistance force torque which is applied to the rotating masses that emulated the vehicle longitudinal inertia and the torques applied to the vehicle wheels depending on the acceleration or braking situation, finished the simulation step.

6. RESULTS

Keeping the objective of this paper to evaluate the influence of the tires inflation pressure in the vehicle fuel consumption to fulfill the NBR6601 proposed route, were performed the co-simulations varying the tire pressures.

Initially we adopted the inflation pressure of 30 psi usually indicated for tires used in the simulated vehicle category. To evaluate the influence of this parameter was chosen a range of 5 psi Over Inflation and Under Inflation conditions. Figure 9 illustrates the effects of tire inflation pressure in the contact area between the tire and the ground.
In all simulations the vehicle performance was satisfactory compared to the standard velocity profile, because of the vehicle fulfills the 12 km route in the standard required time. The results are shown in Tab. 2.

As can be seen in Tab. 2, the inflation pressures below the recommended increases fuel consumption. According to Jazar (2008) in this condition, the tire will support less of the vehicle weight with the internal tire pressure therefore more of the vehicle weight will be supported by the tire structure. This increased tire load causes a larger tireprint, which creates more friction and heat, which can reduce by 25% the performance and tire life. Rievaj et al. (1926) concluded that, the vehicle's handling and stability worsens when the tires are under-inflated. In addition, the tire pressure has impact on the vehicle driving characteristic.

<table>
<thead>
<tr>
<th>Pressure Variation</th>
<th>Tires Pressure</th>
<th>Fuel Consumption</th>
<th>Average Consumption</th>
</tr>
</thead>
<tbody>
<tr>
<td>Psi</td>
<td>Psi</td>
<td>ml</td>
<td>km/l</td>
</tr>
<tr>
<td>-5</td>
<td>25</td>
<td>729.9</td>
<td>16.44</td>
</tr>
<tr>
<td>-4</td>
<td>26</td>
<td>720.4</td>
<td>16.65</td>
</tr>
<tr>
<td>-3</td>
<td>27</td>
<td>714.8</td>
<td>16.79</td>
</tr>
<tr>
<td>-2</td>
<td>28</td>
<td>707.8</td>
<td>16.95</td>
</tr>
<tr>
<td>-1</td>
<td>29</td>
<td>708.2</td>
<td>16.94</td>
</tr>
<tr>
<td>Standard Pressure</td>
<td>30</td>
<td>702.2</td>
<td>17.09</td>
</tr>
<tr>
<td>+1</td>
<td>31</td>
<td>695.0</td>
<td>17.26</td>
</tr>
<tr>
<td>+2</td>
<td>32</td>
<td>689.5</td>
<td>17.40</td>
</tr>
<tr>
<td>+3</td>
<td>33</td>
<td>685.7</td>
<td>17.50</td>
</tr>
<tr>
<td>+4</td>
<td>34</td>
<td>682.5</td>
<td>17.56</td>
</tr>
<tr>
<td>+5</td>
<td>35</td>
<td>678.1</td>
<td>17.69</td>
</tr>
</tbody>
</table>

The tire pressures above the recommended resulted in a saving fuel condition, because in hard surfaces, the rolling resistance generally decreases with the increase of inflation pressure. With higher inflation pressure, the deflection of the tire decreases, with consequent lower hysteresis losses (Wong, 2001). In this situation, most of the vehicle weight is supported by the internal tire pressure, reducing the tireprint, which makes the vehicle difficult to steer because only the central of the tireprint is in contact with the road surface (Jazar, 2008). An increase of the inflation pressure will also result in a higher carcass stiffness, leading to a relatively smaller side-slip angle experienced by the contact patch (Al-Solihat et al., 2010).

In the normal operating condition, using the recommended tire inflation pressure, approximately 95% of the vehicle weight is supported by the internal air pressure and 5% is supported by the tire walls Jazar (2008).

In this condition the tireprint is adequate in the contact with the road (Fig. 9) generating a condition where the tire rolling resistance is not excessive, preventing overheating and fuel consumption increase. This condition is also more suitable than the tire over inflated because despite reducing the rolling resistance coefficient degrades significantly the vehicle handling capacity.

7. CONCLUSION

This study evaluated the influence of the tires inflation pressure on vehicular fuel consumption by co-simulations based on the calculation methodology described in the book Fundamentals of Vehicle Dynamics (Gillespie (1992)) taking into account the transmission system inertia and geometric factors of a Compact Hatchback, equipped with 1.0L Otto cycle gasoline engine.

The equationing for the calculation of the vehicle rolling resistance coefficient was made based on the model proposed by Genta (1997) due to this take into consideration factors like tire type and inflation pressure.

The co-simulation has been performed by the interface between Adams\textsuperscript{TM} where is located the multibody simulated vehicle dynamic model and Matlab/Simulink\textsuperscript{TM} where the longitudinal vehicle dynamics equations were implemented.

The standard velocity profile used was proposed by the Brazilian urban cycle (NBR6601), to maximize the effect of rolling resistance, which is the most relevant factor in the vehicle power demand at low speeds.

Using the tires in a under inflation condition, it caused an fuel consumption increase, because a higher percentage of vehicle weight is supported by the tire walls, causing more structure deformation, which increases the temperature and the tireprint, consequently increasing the tire rolling resistance.

In the condition where the tire is over inflated, the fuel consumption reduces due the increase of tire stiffness that causes a reduction on the contact area between the tire and the ground. This effect can be observed in Fig. 9, however this should be avoided because of the vehicle handling decrease.

Finally this paper conclude that the tire inflation pressure influence significantly in the vehicle longitudinal dynamics changing the rolling resistance coefficient, therefore modifying the engine requested power and consequently the fuel consumption.
8. ACKNOWLEDGEMENTS

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