STEERING INSPECTION BY MEANS OF TYRE FORCE MEASURE

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Summary

A few systems, such as steering, brakes or suspension, critically affect the vehicle's safety. In light of this, it is necessary to check these elements up to a certain age in order to maintain the vehicle in optimal safety conditions. This study is part of a project about effectiveness of Periodic Motor Vehicle Inspections (PMVI), in order to suggest metodologies, instruments and criterions to inspect the vehicle´s safety. It´s well known the relationship between steering geometries and the force in contact patch. The measurement of these geometries is important to evaluate the vehicle dynamic performance.

The main objective is to evaluate the steering inspection method at low speed, current rejection criteria in order to analyze the effect of these results in vehicle´s safety behaviour in real dynamic conditions. The angle that has more influence on the PMVI inspection is the toe angle, being this one the principal parameter of analysis. Experimental force in contact patch with dynamometer plate and simulation results with CarSimTM have been carried out in order to validate the methodology of steering inspection and to check the information obtained in these tests.

Keywords: Vehicle's safety, steering inspection, dynamometer plate, lateral tyre force, PMVI.

1. INTRODUCTION

The recent need to increase vehicle safety leads to study every one of the parameters that influence traffic accidents. Different authors such as Van Schoor, 2001 [17], or G. Rechnitzer, 2000 [13], have investigated PMVI influence on traffic safety, concluding that vehicle defects are a contributing factor over 6% of crashes.

Nowadays, PMVI analyses the vehicles steering system. Several studies show that steering disalignment could be an important factor in traffic accidents. However, very little statistics are available. It is very complicated to check that an accident has been produced by steering disalignment and generally other factors influence the accident.

International Motor Vehicle Inspection Committee (CITA) does not establish a reject value for the steering inspection by means of standard alignment plate. Moreover, spanish PMVI program establishes steering disalignment, measured by standard alignment plate, as a minor defect and no vehicle is rejected by this measure. In this article the need of measuring effectively the steering condition is presented, therefore, in order to solve this problem the dynamometer plate is proposed [1, 5].

A dynamometer plate has been used to carry out this investigation, because it allows measuring in the contact patch the force in the three spatial directions. Therefore, not only the lateral deviation will be obtained by means of the lateral force, as carried out in PMVI, but also the longitudinal and vertical force. In addition, to characterise the vehicle steering system in different dynamic situations, simulations have been carried out by means of CarSim^{TM}, a well known simulation tool [2].

The main objective of this study is to evaluate the steering inspection method at low speed, current rejection criteria in order to analyze the effect of these results in vehicle´s safety behaviour in real dynamic conditions. The angle which has more influence on the PMVI inspection is the *toe angle (toe* describes the angle between the tire's centerline and the vehicle's longitudinal plane*)*, being this one the principal parameter of analysis in this study. Therefore, not only the available adherence in the contact tyre-road for different toe angles will be investigated [3], but also the consequences over the final steering characteristics of the vehicle [18].

2. THEORETICAL ANALYSIS

Driving dynamics deal with the mechanical laws that govern a vehicle's motion with respect to the vehicle's properties and the ones of the road. The description of the vehicle performance is very complex and mathematical models are required for the design and construction of the vehicles themselves [2, 3, 4, 11, 12].

For instance, the forces acting in the contact area between the tyre and the road are the longitudinal or tangential force and the lateral or side force. The longitudinal force in a straight path causes a driving tractive or braking movement. During cornering, the front wheels are at an angle to the longitudinal axis of the vehicle, thereby causing the development of a lateral force F_R that is responsible of lateral or side friction f_R, defined as:

$$f_R = \frac{F_R}{Q} \tag{1}$$

where Q is the weight vertical force on the wheel.

When both the radial and longitudinal forces are present, the resultant force should not exceed an upper limit that could lead to the vehicle sliding off the road.

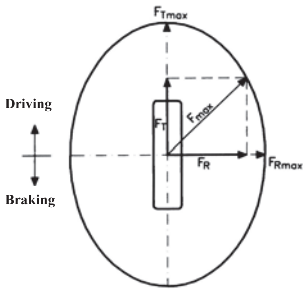

Fig. 1. Ellipse of adhesion limits, Kiencke U. and Nielsen L., 2000 [9]

In order to provide a safe and accurate directional control, the steering and suspension design on the front axle of the vehicle has lead to sophisticated steering geometries where the wheel alignment is governed by the following inclination angles which play an important role in the vehicle dynamic performance: camber, toe, caster and steering axis angle or kingpin [8, 14].

2.1. Cornering performance

When a vehicle supports a certain lateral load (wind, centrifugal force, etc.) a force is generated in the tyre contact to counteract that effect. The tyre ability to generate the needed forces so as to follow the correct path is very important, as it allows evaluating the safety vehicle condition. Therefore, the forces generated in the contact patch depend on the steering angles, and this relationship is the one analysed in this investigation. It has been considered as a reference case that, in which the vehicle follows a circular path and it is subjected to a centrifugal force [8,16].

For low values of the steering angles (the bend radius is much bigger than the vehicle wheelbase) and if the centrifugal force is applied in a perpendicular direction to the vehicle longitudinal plane the lateral forces on the front tyres F_{yd} and on the rear tyres F_{yt} are:

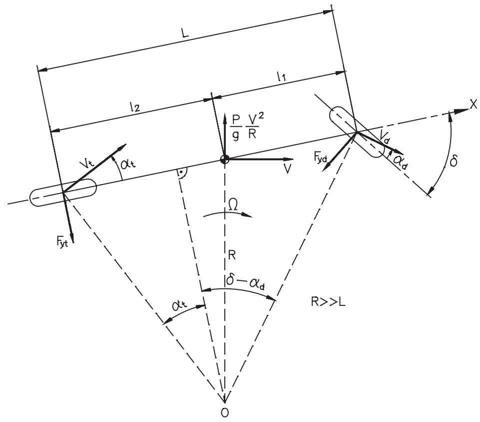

Fig. 2. Two axles vehicle model

$$F_{yd} = 2P_d \frac{V^2}{gR} \tag{2}$$

$$F_{yt} = 2P_t \frac{v^2}{gR} \tag{3}$$

Taking into account that $\alpha = F_y/K_\alpha$, where K_α is the cornering stiffness of one wheel, substituting in Eq. (2) and Eq. (3), and considering that both wheels of the same axle have two times the stiffness of one of them:

$$\alpha_d = \frac{F_{yd}}{2K_{\alpha d}} = \frac{V^2}{gR}\frac{P_d}{K_{\alpha d}} \tag{4}$$

$$\alpha_t = \frac{F_{yt}}{2K_{\alpha t}} = \frac{V^2}{gR}\frac{P_t}{K_{\alpha t}} \tag{5}$$

These equations for the slip angles for the front and rear axle can be introduced in Eq. (6), obtaining Eq. (7):

$$\delta - \alpha_d + \alpha_t = \frac{L}{R} \tag{6}$$

$$\delta = \frac{L}{R} + \left(\frac{P_d}{K_{\alpha d}} - \frac{P_t}{K_{\alpha t}}\right)\frac{V^2}{gR} \tag{7}$$

or:

$$\delta = \frac{L}{R} + K_V \frac{V^2}{gR} \tag{8}$$

$$K_V = \frac{P_d}{K_{\alpha d}} - \frac{P_t}{K_{\alpha t}} \tag{9}$$

K_V, is the **understeer increment** and its value is of great importance to analyse the vehicle steering performance. The vehicle cornering performance is measured by the understeer increment sign. From Eq. (8) it is obtained the the steering angle required for a constant radius turn varies with the speed, or it will be independent, depending on the sign of K_V. Thus, a vehicle can be **neutral steer**, **understeer** or **oversteer**. Considering R = cte:

Neutral steer:	**Understeer:**	**Oversteer:**
$K_V = 0$	$K_V > 0$	$K_V < 0$
(independant of V)	(δ decreases when V decreases)	(δ decreases when V increases)
$\delta = \dfrac{L}{R}$	$\delta = \dfrac{L}{R} + K_V \dfrac{V^2}{gR}$	$\delta = \dfrac{L}{R} - \lvert K_V \rvert \dfrac{V^2}{gR}$

Calculating K_v allows knowing the vehicle cornering performance. It is important to highlight that usually vehicles are designed neutral or understeer, due to the fact that in an oversteer vehicle the guidance angle will become negative for a certain value of V, called the **critical speed** (view Eq. (10)). From this point on, the steering wheel will have to be rotated in the opposite direction to the vehicle turning and the vehicle will become **unstable.**

$$V_{cri} = \sqrt{\frac{gL}{\lvert K_V \rvert}} \qquad (10)$$

3. EXPERIMENTAL ANALYSIS

In this study a dynamometer plate has been used to obtain the instantaneous force and momentums values in the contact patch.

The experimental data obtained by means of dynamometer plate has been registered in PMVI test conditions: a vehicle cross over the dynamometer plate at low speed (1-5 Km/h). It’s important to emphasize that the forces measured in this speed range have a very little variations (less than 2 %), therefore the dynamometer plate shows a great robustness for speed variations in the considered speed range.

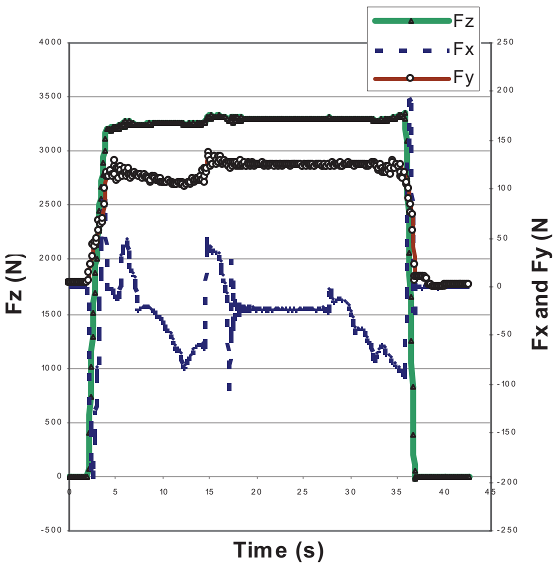

Fig. 3. Force measurement in contact patch (PMVI conditions).

A dynamometer plate consists of a metallic plate supported by eight load cells that measure the forces in the vertical direction and in the contact patch (longitudinal and lateral forces). The dynamometer plate allows a complete force and momentum measurement which is essential to obtain an objective reject value, whereas the standard plate, by itself, does not provide it. The standard plate does not register the vertical force which is a fundamental parameter to be taken into account.

The fig. 3 shows the force mesurement in this test, when the vehicle cross over the dynamometer plate. The results of the same test for different toe angles are depicted in fig. 4.

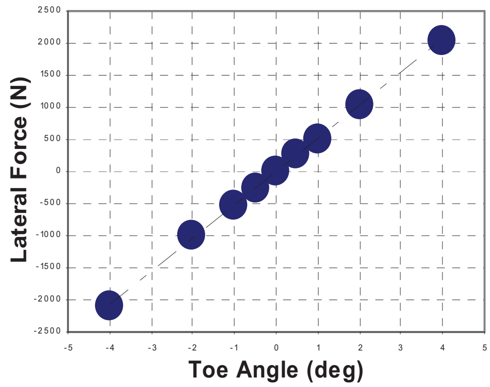

Fig. 4. Lateral forces variation obtained in diferents test for nine different toes angles (PMVI conditions).

The data obtained by means of dynamometer plate has been completed with other experimental results for diferent vertical loads and slips [18]. These measures have been obtained by means of dynamometer drum and trailer.

The input parameters of the simulation application are the measurements obtained by mean of the dinamometer plate, dynamometer drum and trailer.

Fig. 5 shows the experimental data of the lateral tire force in contact patch that was introduced in simulation tool *CarSim*^{TM}. Similar data about longitudinal tire force and aligning moment have been introduced.

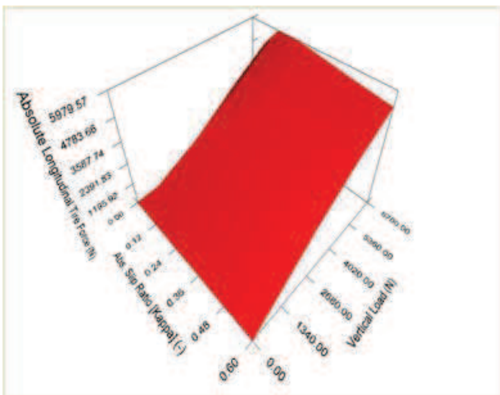

Fig. 5. Experimental three-dimensional graphics of the relationship between longitudinal tire force and slip ratio for diferent vertical loads.

4. SIMULATION PROGRAM

An existing simulation program *CarSim*^{TM} [10], [15] was used for this particular study. The program is a vehicle industry standard, specifically developed for simulating the dynamics of vehicles with tires. It shows how vehicles respond dynamically to inputs from the driver and the immediate environment (road and wind).

It produces the same kind of outputs that might be measured with physical tests involving instrumented vehicles. The program is based in part on technologies developed by the University of Michigan Transportation Research Institute.

The model of a vehicle is built up by using different components that are mathematically described by mass and inertias. The movements of the different components are restricted relative to each other by connection elements (ball joints, links and swing axels) which are also described mathematically. Force elements as coil springs, anti-roll bar, and dampers are placed between the components. The force characteristics of these elements may be nonlinear. A tire model based on Pacejka 5.2 [12] version of the Magic Formula is usually used by the program to simulate the behaviour of the tire. In this study the experimental measurements have been used in order to achieve the desired accuracy, so that safety conditions due to steering geometry are correctly evaluated.

The input to the system consists of a path that the vehicle has to cross at a certain speed. Other inputs are external forces acting on the vehicle (wind forces), the steering angle of the wheel, torque on the driver wheel and road excitation. For the simulation of the transition closed loop handling test (double lane change), it is necessary to use a driver model that steers the vehicle along the prescribed path.

Outputs of the simulation program, which can be extracted against time or other variable, include over 500 parameters as:

- Displacement, velocity and acceleration in any of the six degrees of freedom of the sprung mass.
- Tire force and moments.
- Spring and damping forces and displacements.

5. VEHICLE CHARACTERISTICS

A typical model for a small class European vehicle has been used for the purpose of this study. The main vehicle parameters which have been used are listed in Table 1. The rest of parameters that are not listed in Table 1 have less influence on side behaviour and medium values obtained from CarSim^{TM} database have been used.

The steer due to the steering system is obtained by combining the steering wheel control with nominal gear ratio, while nonlinear tables combines the geared-down steering wheel angle to the road steer angle, with Ackerman and others effects.

An important item to take into account is the performance of the tire. Parameters from a 175/65 R14 tire have also been used.

Table 1. Vehicle characteristics summary.

Item	Unit	Value
Wheelbase	m	2.49
Front Track	m	1.47
Rear Track	m	1.445
Total Weight	N	11368
Front Weight	N	6595.4
Rear Weight	N	4772.6
Height of centre gravity	m	0.54
Front Sprung mass	kg	603
Rear Sprung mass	kg	387
Front Unsprung mass	kg	70
Rear Unsprung mass	kg	100
Roll inertia (Ixx)	kg·m^2	288.0
Pitch inertia (Iyy)	kg·m^2	1152.0
Yaw inertia (Izz)	kg·m^2	1152.0
Total length	m	3.925
Width	m	1.68
Height	m	1.545
Tires		195/60 R15

6. TEST CONDITIONS

In order to evaluate the influence of wheel alignement (front axel) on handling characteristics of the vehicle, a double lane change manoeuvre has been tested. Three vehicle conditions have been compared:

Vehicle Condition A → (Toe angle = 0º) Usual design specification.
Vehicle Condition B → (Toe angle = 2º) Bad condition.
Vehicle Condition C → (Toe angle = 4º) Very bad condition.

The double lane change (DLC) is a well-known and commonly used test that has been prescribed in a concept standard ISO TR-3888-1 [7]. This test allows for the evaluation and comparison of the handling characteristics of vehicles through some objective parameters such as roll angle, roll rate, yaw rate, lateral acceleration (a_y) and the Dynamic Stability Index (DSI) [6].

Double lane change tests are implemented with a transition length of 35 m and a width of 3.5 m, following the path illustrated in fig. 5. Tests have been done following the suggestion of the standard ISO TR-3888-1 [7], which involves beginning at 40, 60, 80 km/h and increasing the speed until the vehicle fails the test.

The article simulations have been carried out at 60 km/h.

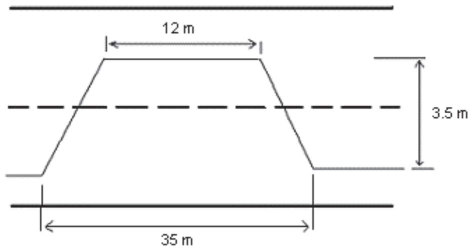

Fig. 6. Asymptotic driving course for a DLC test.

Stationary tests have also been considered to study the vehicle steering characteristics [16]. These tests allow obtaining the cornering stiffness, K_V, and how it changes when dynamic conditions are also altered, as well as its relationship with the toe angle. The proposed tests are detailed below:

- **Constant radius tests**: During this test the vehicle travels along a constant radius turn, at different speeds, measuring either the speed or the lateral acceleration $a_y = V^2/R$, and controlling the steering angle by means of the steering wheel angle δ_V, which allows knowing the wheel steering, allowing us to characterise the cornering stiffness as a function of the centrifugal force.
- **Constant speed tests**: Now the vehicle travels at a constant speed for different steering angles, the path will be of different radius and the vehicle will also be subjected to different lateral accelerations (V^2/gR), obtaining the cornering stiffness as a function of the centrifugal force.
- **Constant steering angle test:** In this case, the steering wheel angle δ_V is constant, and when the vehicle travels at different speed different paths and lateral accelerations will be obtained. Thus, the relationship between the cornering stiffness and the centrifugal force is obtained.

7. TEST RESULTS

Preliminary simulations show the lateral forces in a test for PMVI conditions (straight movement at 3 km/h) as fig. 7, 8 and 9 show. This information is important to compare the simulations with experimental test in the same conditions (fig. 4).

The results indicate the enourmous resemblance between both kinds of data. It is a normal conclusion because in these conditions, at low speed, there is no load transference neither other dynamic effects, and the results are a direct consequence of tire data introduced.

It can be seen that the introduced toe angle in each of the vehicles results in different lateral forces linear proportional for the studied range (0º - 4º). This implies, that the lateral force for vehicle A that travels in a straight path is lower than vehicle B and this one lower than vehicle C. It can be checked that the relationship between the toe angle and the lateral force is approximately equal when comparing the simulation results (Fig. 7, 8 and 9) and the experimental tests (Fig. 4).

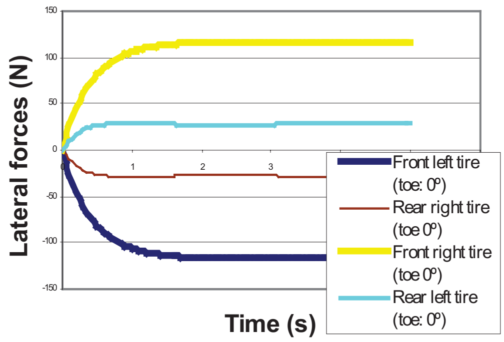

Fig. 7. Vehicle” A” lateral tire forces evolution in a simulation (PMVI conditions).

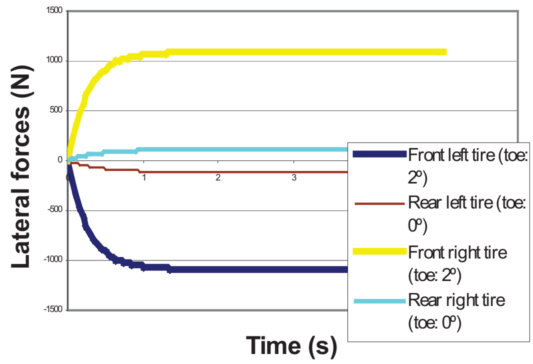

Fig. 8. Vehicle “B” lateral tire forces evolution.

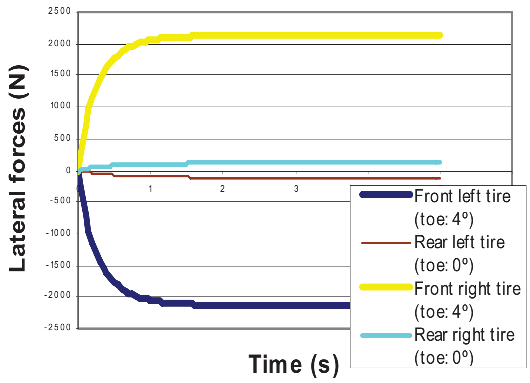

Fig. 9. Vehicle “C” lateral tire forces evolution.

If it is taken into account that the maximum forces that can be transmitted to the ground are those limited by adherence in the longitudinal and lateral direction, as shown by the adherence ellipse shown in Figure 1, it can be observed that the margin for case A is bigger than for case B, and for this one bigger than case C. Thus, a vehicle travelling along a straight line uses more lateral adherence when the toe angle is bigger, reducing its lateral adherence margin for dynamic conditions.

Without taking into account the load transfer and the slip angle, and considering a good pavement condition with lateral adherence value of $\mu_y = 0.64$

the 3500 N that each wheel of the front axle support leads to a maximum transmitted lateral force of 2240 N. This value is very near to that achieved for a toe angle of 4 degrees (2100 N), that is, vehicle C will therefore have less adherence capacity than vehicles A or B before beginning to slip in the lateral direction and less capacity to transmit longitudinal forces. Therefore, the proposed assumptions, although severs, allow carrying out a first order approximation to how the system is conditioned and to justify the need to have a good steering system inspection.

To complete this argument a simulation was done in a demanding dynamic situation, double lane change ISO TR-3888-1 [7], to compare the differences introduced for three differents toe angles.

Initially there were obtained qualitative outputs to compare the different behaviours for these vehicle configurations.

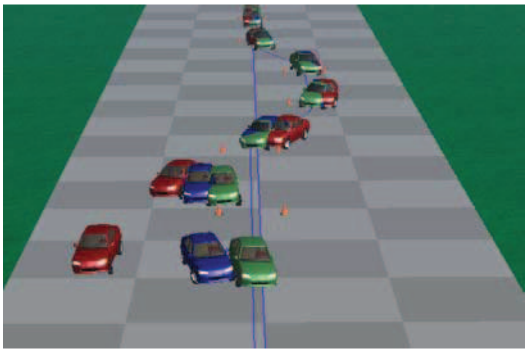

Fig. 10. Three different trajectory described for the vehicles: A (green), B (blue) and C (red).

The fixed trajectory is correctly described by vehicle A, however, vehicles B and C could not describe it correctly, as depicted in fig. 10, 11.

Vehicles that are not able to follow the trajectory describe the most nearer trajectory to the reference one under limit adherence conditions. It is defined a performance measure to evaluate the ability of the vehicle to follow a predetermined path as the maximum lateral deviation with respect to the reference one. For vehicle condition A, the maximum lateral deviation is 1.5 m, for vehicle B, 2.5 m and 7 m for vehicle C.

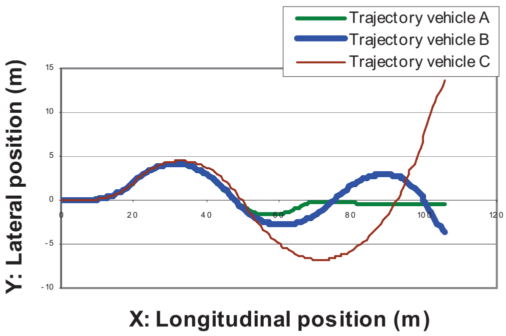

Fig. 11. Trajectory for the three different toe angles. Simulation time: 5s.

The double lane change ISO TR-3888-1 [7] is a very hard dynamic situation, and is difficult even for the vehicle A. The vehicle B keeps the trajectory until the last corner, and the vehicle C loses the trajectory on the third corner. As it has been described previously, the vehicle with a toe angle included between ±0.25º has more adherence available than vehicles with upper toe angles.

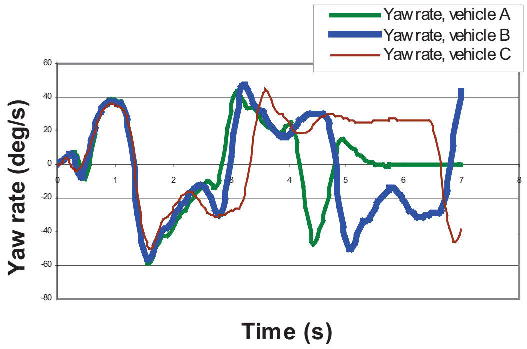

Fig. 12. Yaw rate (sprung masses). Simulation time: 7s

From Figure 12 it can be observed the progressive vehicle B held up concern the vehicle A, and the vehicle C concern vehicle B. The maximum values for yaw rate are similar for three vehicles. This maximum values for the vehicle A fit in with the corner positions for the double lane change test fixed.

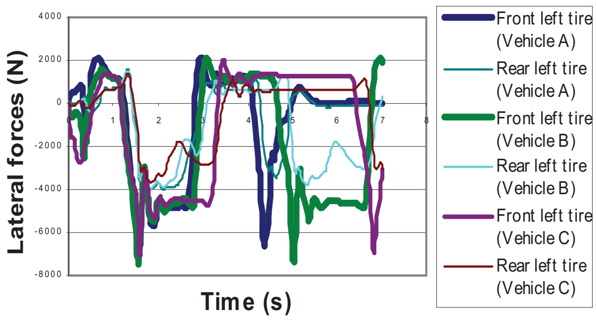

Fig. 13. Tire lateral forces (Fy), left side. Simulation time: 7s

It is very interesting to observe how at the beginning of the test the lateral forces are very different, before the first corner, as shows fig. 13. Before the first corner, vehicle C already consumes almost all force necessary to describe the corner and therefore, it does not trace the correct trajectory. From that moment on the vehicles with the three configurations cannot describe the same trajectory; their initial conditions do it impossible.

From above, it arise the need to characterise in an objective and precise procedure the steering characteristics of a vehicle. Therefore, simulations of the stationary tests were carried out in order to compare the values of the cornering stiffness for different vehicle configurations.

Next, different figures show the proposed tests (Fig. 14, 15, 16 and 17), and the influence of the cornering stiffness is analysed for each of them.

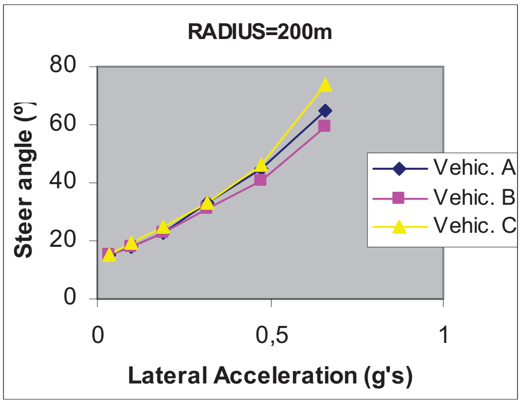

Fig. 14. Stationary test: constant radius turns of 200m.

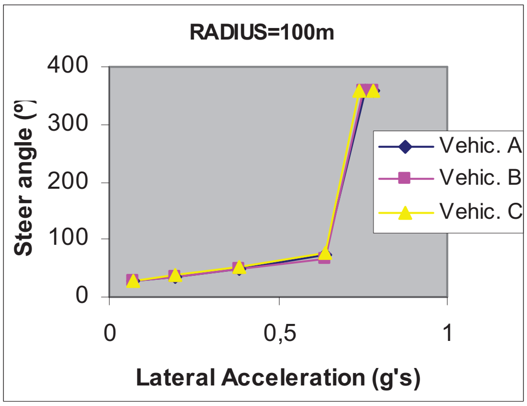

Fig. 15. Constant radius turns of 100m.

In Fig. 14, 15, 16 and 17 the most representative results for the stationary tests are depicted. Therefore, although other simulations with constant steering angle and many others for turns with different radius, different speeds, etc. because the four figures show the most important information about steering behaviour.

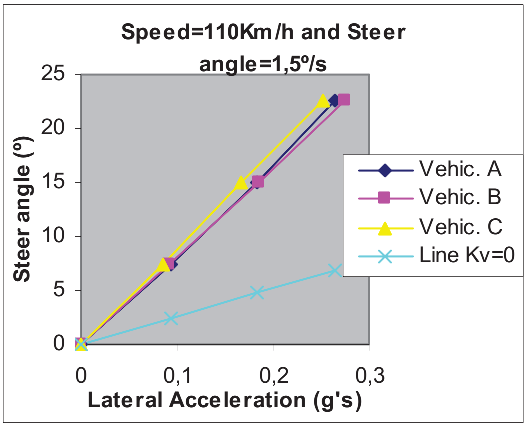

Fig. 16. Stationary test: constant speed of 110km/h constant steer angle variation 1.5º/s.

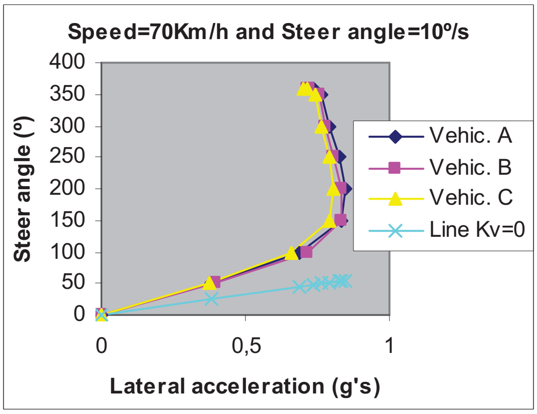

Fig. 17. Constant speed of 70km/h constant steer angle variation 10º/s.

In Fig. 14, 15, 16 and 17 it can be seen that the vehicle with bigger understeering behaviour is the vehicle with bigger toe angle. Specifically, in Fig. 14 and 15 it can be seen that for different radius turns the graphic slope is more positive for bigger values of the toe angle. If it is taken into account that the slope of the graphic has a direct proportion with K_V, which will be negative when the slope is negative (oversteer), equal to zero when the slope is zero (neutral) and positive when the slope is positive (understeer), it is clearly observed that there is an increase of the understeer behaviour for bigger values of the toe angle. On the other hand, Figures 16 and 17, also show that there a relationship between the graphic slope and K_v, but in this case the slope of 45º corresponds to K_v =0, slopes bigger than 45º will be for understeer vehicles and for lower slopes the vehicle will be oversteer. For the case of constant speed the vehicle will have an unstable steering behaviour for negative slopes, which is first arrived for bigger values of the toe angle, as shown in Fig. 17. All of these results indicate the due to the fact that the available adherence capacity is decreased due to the increase of the toe angle, the vehicle suffers a slip of the front axle leading to an increase of the understeer behaviour. However, this analysis has been carried out for vehicles with different types of suspension configurations and it has been seen that understeer behaviour strongly depends on the type of suspension employed.

Nowadays, and taking into account the tests and simulations shown in this investigation, a possible reject limit for the dynamometer plate of μ_y = Fy/Fz = 0.15 is proposed. However, no experimental tests have been carried out to validate that value. The possible range to establish a reject value of μ_y is between 0.12 and 0.2. For the case of the analysed vehicles: A, B and C, μ_y would be very near to 0.065, 0.57 to 1.14 respectively. However, it has to be taken into account that extreme cases have been considered; a value of 0.15 for this vehicle implies a toe angle of 0.5º in each test, which is bigger than the conventional values fixed during steering design.

8. CONCLUSION

- Measurements with dynamometer plate in periodic inspection conditions, double lane change simulations and stationary conditions simulations have been carried out. The results obtained show the need to carry out some kind of steering inspection to guarantee the safety vehicle conditions.
- Simulations show the toe angle influence, and the important behaviour dynamic variations. Also it has been analyzed the camber angle, but its influence is notably fewer than the toe angle influence, and for this reason it has not been included in this paper.
- Other steering angles have not been studied, because they should not change in the vehicle life, except if the vehicle suffers an accident or an important reform. These situations are out of the objectives of a periodic inspection and out of this study.
- For the cornering vehicle behaviour, it has been shown that the vehicle becomes understeer for bigger values of the toe angle. This is due to the increase of the slip of the front axle because the available adherence is diminished for bigger values of the toe angle.
- Dynamometer plate measurements have shown to be representative of vehicle safety.
- In future studies other kind of dynamic tests will be carried out, and dump conditions, toe angle influence in emergency brake, etc. will be included. Furtheremore the dynamometer plate system will be analyzed as an instrument used to carry out periodic motor vehicle inspections.

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