https://doi.org/10.15623/ijret.2018.0709011 Received: 25-06-2018, Accepted: 06-08-2018, Published: 27-08-2018

MEASUREMENT AND ANALYSIS OF VEHICLE COMFORT AND ROAD QUALITY

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Abstract

The suspension system of a vehicle is an essential component because it is related to the dynamic performance of the vehicle. It ensures the tire-grounding contact, absorbing the oscillations excited by the road, and maintains the stability of the vehicle by distributing the wheel forces, providing comfort and safety to vehicle occupants. Structurally it links the vehicle sprung mass to the unsprung mass, and it connects to other subsystems, such as the steering and the brake components. In order to analysis and design a suspension system, several models that simulate the suspension dynamics behavior are available, however each has a different level of complexity and the accuracy of the results varies. Thus, the aim of this study was to compare a quarter car model with the real dynamics of a vehicle using a Matlab simulation. A test vehicle was instrumented with three-axis accelerometers, through which data on the acceleration of the sprung mass and unsprung mass was obtained when the vehicle was excited by a speed bump. From this simulation, it was possible to perform a model update to obtain the approximate vehicle parameters. The road surface quality plays a fundamental role in the vehicle oscillation and therefore in the comfort of the vehicle occupants. Using the instrumented vehicle and the adjusted parameters of the model, some routes along different pavement types in the city of Joinville were chosen to collect the vibration data in a vehicular test. The results were used to determine the international roughness index (IRI), an index to evaluate road quality. It was verified that the city roads with high IRI values, for instance, on streets paved with cobble stones, are associated with greater discomfort to passengers inside the vehicle, reflecting a low quality of the pavement. In contrast, lower IRI values are obtained for roads with a higher quality surface, resulting in greater comfort for the passengers.

Keywords: - *Suspension, quarter vehicle, IRI, comfort and road quality*

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1. INTRODUCTION

Suspension systems have been used in the automotive industry since the first vehicles were manufactured, to absorb the vibration excitations due to vehicle maneuvers and road irregularities. According to Stone and Ball (2004) [1], the suspension system forms an interface between the road surface and the vehicle body and it plays a vital role in ensuring the comfort of the user, absorbing impacts and vibrations. The system also provides stability and vehicle drivability, maintaining a constant contact pressure between the tire and the ground.

These features are very important to maintaining high levels of vehicle safety and passenger satisfaction [2]. For consumers, comfort is one of the most significant factors in selecting a vehicle. Thus, the focus in the development of suspension design is to provide a certain amount of comfort, avoiding sickness of the occupants, as a client requirement[3],[4]. Walker et al. (2015)[5] describes that drivers generally demonstrate a remarkable sensitivity toward vehicle response, and automakers need to take into account subjective factors characteristic of consumer groups to optimize the vehicle design process.

According to Wong (2001)[6], although a vehicle represents a complex system of multiple degrees of freedom (dof), the suspension system can be simplified using only the most relevant degree of freedom to describe its functional behavior. Thus, in vehicle dynamics studies, simplified models are essential tools that are used to facilitate an understanding of the vehicle parameters ensitivity and to design the vehicle components. For example, Gao et al. (2015)[7] analyzed and compared the effects of the hydraulic bushing of control arms on the vehicle performance. In this case, the model was built in the ADAMS software program. In another approach, Kuznetsov et al. (2011)[8] and Sezgin and Arslan (2012)[9] illustrated the mathematical formulation of a model and its application in system optimization and a vertical vibration study, respectively, to improve the vehicle ride quality. The model was based on a reduced mass-sprung system.

However, since these models are approximations, it is necessary to verify how reliably they represent the performance of the suspension system. In general, experimental tests are performed to compare the suspension response of the simplified model with that measured using a real vehicle. Quarter vehicle models are the simplest representation of a vehicle and they are commonly used to study independent suspension systems, such as the

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MacPherson system. As described by Gillespie (1992)[10], for a two lumped-parameter dof model for independent suspensions, an excitation from the road reaching the wheel on one side would not interfere with the dynamics of the wheel on the opposite side.

The MacPherson suspension is widely employed in the automobile industry because it is a very simple assembly and has a compact size. It is mounted with a telescopic shock absorber and a concentric spring along the strut structure. A lower control arm supports the strut and it links the spindle to the vehicle chassis. Also, the independent suspension has a stabilizer bar connecting the two sides of the lower control arms. Reportedly, this is one of the biggest advantages of MacPherson suspension[10] and it does not take up much of the engine space.

As mentioned above, in the automotive area comfort has been the focus of many studies to improve the quality of the vehicle ride and reduce interior noise, which forms part of the vibro-acoustical system[11]. The ride comfort is defined by Genta and Morello (2009)[12] as the ability of the tires and suspension system to filter external vibration excitations caused by the road irregularities. Among the main factors that result in complaints are noise and vibration, grouped together in noise, vibration and harshness (NVH) studies. These oscillations of low frequency, if not damped, can lead to passenger discomfort and are often responsible for customer dissatisfaction[13].

The vehicle dynamic response to pavement roughness can be measured by placing accelerometers on the sprung and unsprung mass. The correlation of the measured data with user satisfaction is proposed in the standard ISO 2631 (1997)[14], which regulates the parameters of the vehicular comfort and safety. This standard provides information on the parameters correlated with the vibration frequency and the vertical acceleration, in addition to the sensitivity of the human body to vibration from different directions. The standard presents, for example, a scale of discomfort due to vertical acceleration (Table 1). Table 1 shows the relation between the root mean square (RMS) acceleration of the sprung mass and the subjective sensation of the driver and passengers with regard to the comfort of the vehicle ride.

Table 1: Comfort scale according to ISO 2631

RMS acceleration (m/s^2)	Comfort scale
≤ 0.315	Comfortable
0.315 – 0.630	Slightly comfortable
0.500 – 1.000	Slightly uncomfortable
0.800 – 1.600	Uncomfortable
1.250 – 2.500	Very uncomfortable
> 2.500	Extremely uncomfortable

Another important system characteristic is the road condition. The road undulation affects the suspension motion and, consequently, the vertical and roll dynamics of the vehicle [15]. Poor road quality generates higher excitation forces on the wheels, which are transmitted to the vehicle (sprung mass). This condition is analyzed by Liang et al. (2013)[16] for an excitation caused by speed-control humps.

Due to the associated costs and user requirements, the quality of roads has become an important concern. Road managers, for instance, aim to maintain the quality of paved road surfaces through periodic maintenance. Thus, methods have been developed to identify and monitor road features to ensure that the conditions are appropriate for use. The main tool employed is the international roughness index (IRI), which is widely applied to assess the quality of roads in the United States [12].

In this context, to obtain data on vehicle suspension systems and also information regarding road conditions, sensors such as inertial accelerometers are used by automotive engineers and researchers. These data can then be used to adjust a numerical model of the vehicle, and with this model the IRI value for the road can be determined and the road conditions can be classified. The objective of this paper was to obtain vibration acceleration data from a low-cost data acquisition system, using an Arduino Uno microcontroller board, and determine the condition of a road pavement in the city of Joinville (Brazil).

2. EXPERIMENTAL SETUP

In this study, an Arduino Uno microcontroller board and two 3-axis accelerometers (ADXL345) were used to measure the vertical vibration of sprung and unsprung masses of a test vehicle. In the measurements performed, the accelerometers were configured to have a range of ±8g to avoid overloading in terms of the signals and a sampling rate of 55Hz. The electrical circuit of the Arduino board and ADXL345 accelerometers is represented in Figure 1.

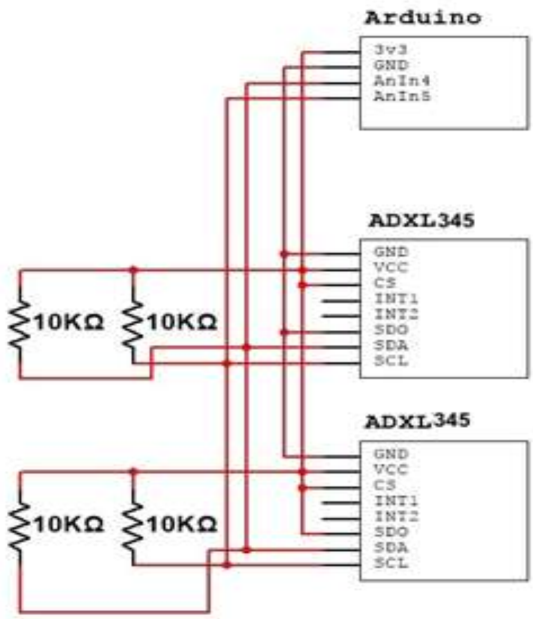

Fig 1: Circuit of the Arduino board and two ADXL345 accelerometers

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The system offers aneconomic advantage due to low-cost components, sensors and data acquisition hardware. For instance, the Arduino Uno board and two ADXL345 accelerometers cost around $30.00.

The tested vehicle is a passenger car with front-wheel drive and McPherson suspension in the front axle. The vehicle was instrumented on the right side of the car with two accelerometers. One was positioned on the lower control arm of the suspension (unsprung mass) and the other on the front wing support of the sprung mass. Figure 2 illustrates the sensors on the vehicle for sprung and unsprung masses (left and right figure, respectively).

For the characterization of the test vehicle, a weighting scale was used to determine the weight distribution on each wheel, as illustrated in Figure 3. During the measurement process, the mass was obtained considering the vehicle with a driver and one passenger present. The weight distribution is shown in Table 2. Also, an equivalent spring stiffness test was carried out by applying different sprung mass heights, using an automotive elevator. The measured suspension stiffness is reported in Table 2.

Fig 2: Accelerometer positioned on the sprung mass (left) and accelerometer positioned on the unsprung mass (right).

Fig 3: Test vehicle.

3. DYNAMIC MODEL

In this study, the analysis of the vehicle was carried out on a simplified dynamic model, that is, a quarter vehicle (car) model. The model has two degrees of freedom in the vertical direction, one representing the sprung mass (M) and the other the unsprung mass (m).

Table 2: Weight distribution on the vehicle and suspension stiffness.

Wheel	**Weight (kgf)**	**Stiffness (N/m)**
Front left	351.0	40413.3
Front right	350.0	44470.7
Rear left	224.8	19924.1
Rear right	245.2	23221.3
Total	1171.0	–

3.1 Equation of Motion

In this model, the car weight is divided into four parts and each part is associated with a wheel and the suspension system. Figure 4 shows a representation of the vehicle constituted by the sprung mass M, which represents the vehicle body, and the unsprung mass m, which represents the mass of the axle, brake disc/drum and the mass of the wheel. The suspension system has K_s stiffness and C_s damping parameters. The tire is represented by a stiffness of K_t. The excitation originating from the road irregularities is described by the tire displacement (Z_r).

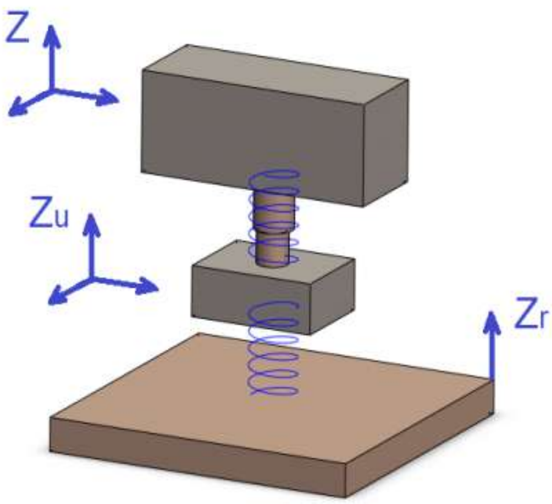

Fig 4: Quarter car (vehicle) model.

The resulting equations of motion are shown in Equations (1) and (2):

$$M.\ddot{Z} + K_s.(Z - Z_u) - C_s.(\dot{Z} - \dot{Z}_u) = 0 \quad (1)$$

$$\begin{aligned} m.\ddot{Z}_u + K_t.(Z_u - Z_r) - K_s.(Z - Z_u) \\ + \quad -C_s.(\dot{Z} - \dot{Z}_u) = 0 \end{aligned} \quad (2)$$

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3.2 Model Adjustment

The numerical adjustment of the model was performed based on a previous experiment on the test vehicle. Here, the acceleration of the sprung mass can be related to the model vibration amplitude. A known road profile (a bump) is applied to excite the front suspension on the right side at a vehicle speed of 9km/h. For the simulation, the same profile used in the experiment is measured and described using a parabolic function of the bump height. An ultrasonic tape measure supported on a horizontal structure was used for the discretization of the profile, as shown in Figure 5.

Fig 5: Speed bump used for the model adjustment.

Equation (3) presents an approximated formulation obtained from the measured points to describe the profile of this bump. The curve is shown in Figure 6, where y is the bump height and x is the traverse distance in meters.

$$y = -3.7143.x^2 + 0.9229.x - 0.0014 \qquad (3)$$

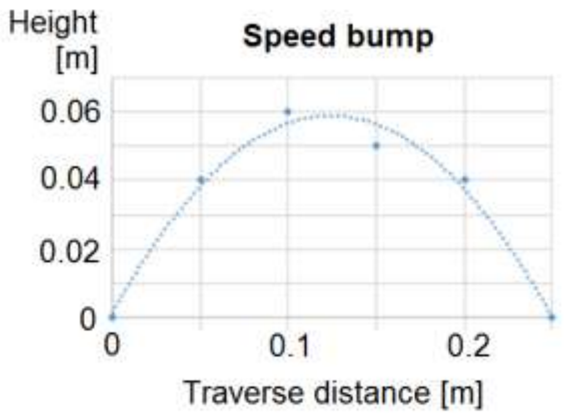

Fig 6: Speed bump profile obtained from parabolic curve.

The frequency of the mass oscillation was adjusted, for the right front suspension, by the weight distribution between the sprung and unsprung masses, and the decrease in the oscillation amplitude provided by the suspension damping coefficient. A comparison between the simulated response and the measured vibration is shown in Figure 7. The first two cycles of Figure 7 have similar decrements in the frequency and amplitude; however, after this period the effect of the rear suspension vibration adds a different frequency component to the signal. Since the quarter car model represents two dof dynamics, the simulated response would not identify the presence of the new frequency.

The parameters of the adjusted vehicle model are given in Table 3. This resulting numerical adjustment uses a slightly different stiffness value (less than 24%) in the function for modelling the response.

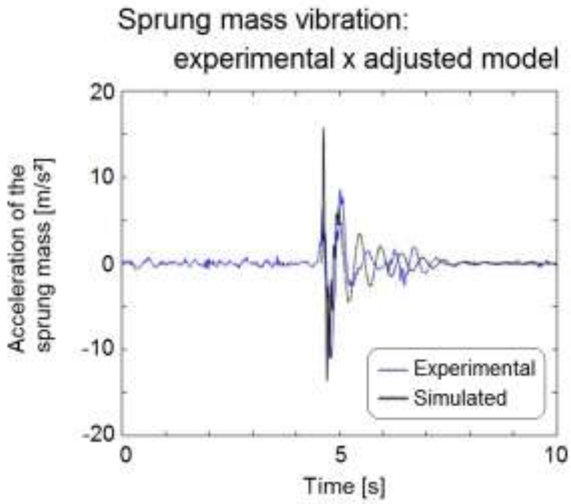

Fig 7: Comparison between simulationand measurement data.

Table 3: Parameters of the vehicle according to the model adjustment.

Variables	Symbol	Value
Suspension stiffness	K_S	55000.0N/m
Damping coefficient	C_S	800.0N.s/m
Sprung mass	M	294.0kg
Unsprung mass	m	56.0kg
Tire stiffness	K_t	550000.0N/m

4. INTERNATIONAL ROUGHNESS INDEX (IRI)

In general, the IRI has been used worldwide as a tool to quantify road smoothness. According to Arhin et al. (2015)[17], it is a standardized measure of vehicle reaction to the road profile and roughness developed by the World Bank in the 1980. This mathematical methodology is used to evaluate the road profile through measuring experimentally the acceleration of sprung and unsprung masses and using the adjusted quarter vehicle model [18].

The IRI calculation will return a value that is non-dimensional or with non-consistent units (m/km or in/mi). The latter is more commonly used. The relationship between the IRI and the road characteristics is shown in Figure 8. The higher the index value the poorer the quality of the road

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surface will be. In the figure, an IRI above 10.0 means unpaved road characteristics and an IRI below 4.0 indicates good pavement surface (i.e., interstate highway or airport pavement quality).

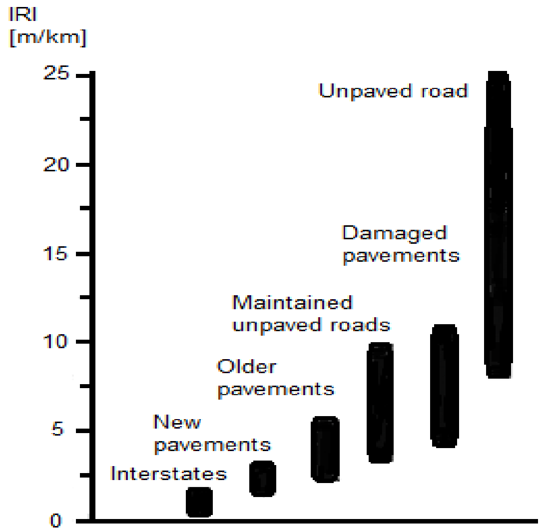

Fig 8: Relationship between IRI and road characteristics

4.1 Estimation of surfaceroughness

Based onthe vehicle data collected, a surface roughness value can be obtained. Isolating the excitation term, Z_r, of Equation (2) results in the road profile, as given in Equation (4).

$$Z_r = \frac{m.\ddot{Z}_u + K_t.(Z_u) - K_s.(Z - Z_u) - C_s.(\dot{Z} - \dot{Z}_u)}{K_t} \quad (4)$$

Since the terms on the right side of the equation are all known, being determined numerically or by measurements, it is possible to find the profile of the Z_r based on the parameters of the quarter car model. The dof of the velocities and displacements are obtained using numerical integration applying the trapezoidal rule.

4.2 Analysis of road quality

According to Du et al. (2014)[19], there is a strong relationship between the IRI and the vertical acceleration of the vehicle, since it is based on the quarter vehicle model. According to [12], using a new quarter vehicle model with two degrees of freedom, it is possible to calculate the IRI value. This is called the golden car model, and the input parameters are defined in Equations (5), (6), (7) and (8). In this study, the golden car model was defined for an adjusted sprung mass of $M = 294\ kg$.

$$\frac{K_s}{M} = 63.3\ s^{-2} \quad (5)$$

$$\frac{K_t}{M} = 653.0\ s^{-2} \quad (6)$$

$$\frac{m}{M} = 0.150 \quad (7)$$

$$\frac{C_s}{M} = 6.147\ s^{-1} \quad (8)$$

From the motion equations of the golden car model, the vibration response for road excitation is determined through a numerical solution. The integration of ordinary differential equations is solved using the Runge-Kutta method. The IRI value is calculated through Equation (9), defined by [19], for a reference speed of 80km/h.

$$IRI = \frac{1}{L}\int_0^L \left|\dot{Z} - \dot{Z}_u\right| dt \quad (9)$$

where L is the distance traveled by the vehicle (m),$\dot{Z}$ is sprung mass velocity (m/s) and $\dot{Z}_u$ is the unsprung mass velocity (m/s). The variables $\dot{Z}$ and $\dot{Z}_u$ are obtained as the numerical integration solution of the golden car equations with excitement by the road roughness obtained from Equation (4).

The flowchart in Figure 9 summarizes the steps followed to determine the IRI value.

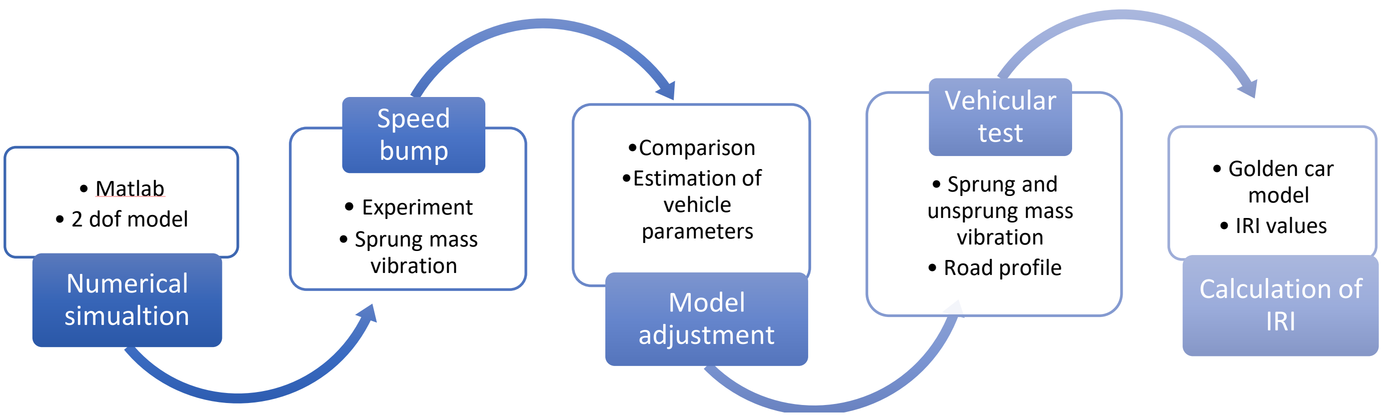

Fig 9: Flowchart showing calculation of IRI.

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5. EXPERIMENTALRESULTS

After the model updating procedure conducted with the test vehicle, on-road testing to verify the IRI calculation was performed in the city of Joinville in October 2017. After the data collection, signal processing was carried out using a digital filter. This procedure was configured for a second-order high pass Butterworth filter with a cutoff frequency of 1.0Hz.

Seeking to include different pavement qualities, the area chosen for the vehicular tests was comprised of junctions between main roads and small side streets. The driving route is shown on the map in Figure 10, where the stretches of road were numbered according to the streets included in the test. The test setting on the vehicle to evaluate the vibration measurement was a speed of 40km/h. Images of some of the pavements tested and their irregularities are shown in Figure 11.

Vibration measurements were taken on these routes and, as an example, Figure 12 shows the vibratory acceleration on route 1. Based on this figure, the measurement results can be compared with those obtained for the sprung and unsprung masses. The sprung mass has a more attenuated response than the unsprung mass, as expected from the suspension performance.

Numerical analysis with the spring-mass model was used to calculate the road roughness according to Equation (4). The result for route 1 is shown in Figure 13. The graph shows a typical profile for an old cobbled road. The vibration can be considered to be the oscillatory motion around the equilibrium position.

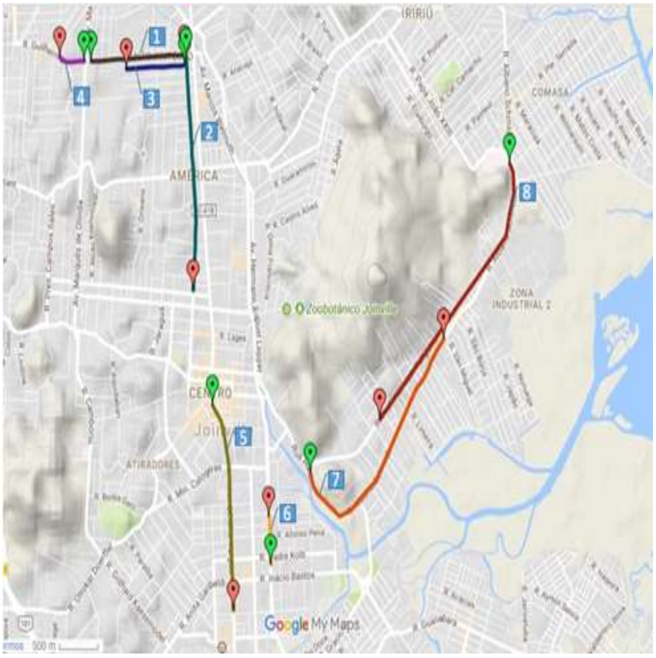

Fig 10: Map of Joinville showing the streets used to obtain the measurements.

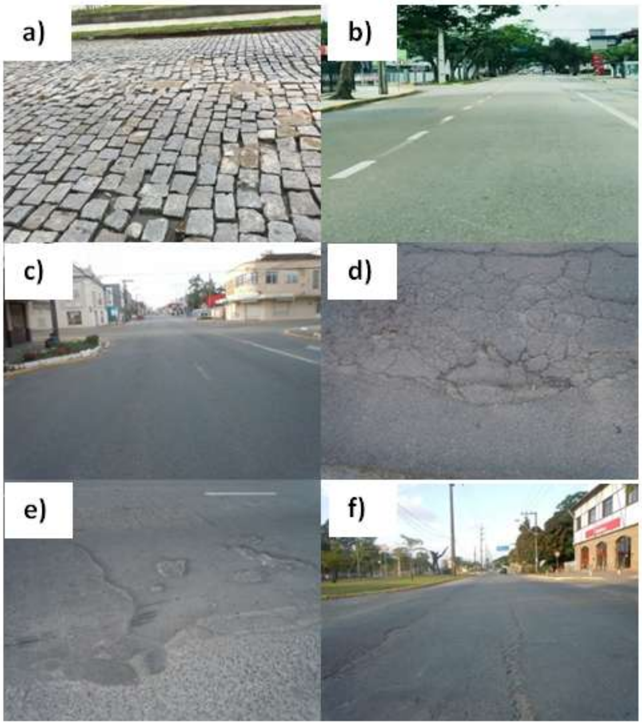

Fig 11: Illustration of road conditions. (a) route 1, (b) route 2, (c) route 5, (d) route 6, (e) route 7 and (f) route 8

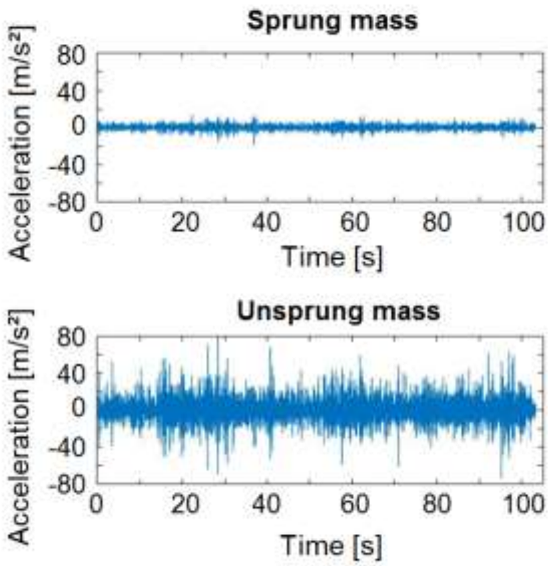

Fig 12: Sprung and unsprung mass vibration along route 1.

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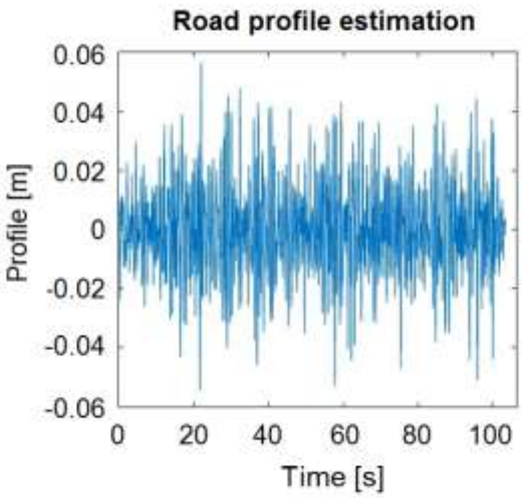

Fig 13: Estimation of road profile for route 1.

Through the estimated profiles and with the parameters of the golden car model, the IRI values for each of the routes in Table 4 were calculated. The values refer to the IRI for the whole stretch of road measured.

Table 4: IRI values for the stretches of road measured.

Route	Street	Distance (m)	IRI (m/km)
1	Prudente de Moraes	1149.0	14.9
2	Blumenau	1684.5	3.7
3	Marcílio Dias	744.1	5.3
4	Guilherme	109.0	16.0
5	Getúlio Vargas	1088.4	2.9
6	Coronel Procópio Gomes	356.9	4.8
7	Albano Schmidt	2499.6	6.9
8	Helmut Falgatter	2838.8	10.1

The graphs in Figure 12 show that for the more rugged roads paved with cobble stones, such as route 1 ($IRI = 14.9\ m/km$), the accelerations of the mass of the vehicle are greater than on paved roads and result in a higher IRI value. Also, as shown in Figure 8, based on [12] and ASTM E1926-08 (2015)[20], this higher IRI value indicates that the road is not paved or is damaged. When a stretch of road is not well maintained there will be a greater amount of unevenness and undulations, resulting in greater oscillation of the mass of the vehicle. Therefore, at speeds up to 40km/h the vehicle will perform with less comfort and less preservation of the mechanical components due to fatigue.

According to ISO 2631 (1997)[14], as shown in Table 1, accelerations above 1m/s² cause significant discomfort to passengers if sustained for a long period. The average RMS value for the sprung mass obtained from the data in Figure 12 reached 3.1m/s², indicating excessive discomfort to the occupants of the vehicle.

On the other hand, route 5, a road with better conservation characteristics, presented a lower IRI value ($2.9\ m/km$). A lower IRI value is consistent with low oscillation amplitude along the route and indicates new pavement conditions. Thus, vehicle speeds over 100km/h would be acceptable, based on the subjective assessment of the vibrations. For the vehicle occupants, this level of oscillation comes as close as possible to conditions of comfort. From an analysis of the RMS average, the sprung mass value calculated for route 5 is approximately 1.0m/s². According to Table 1, this characterizes a not very comfortable situation, due to the small irregularities present on the asphalt surface.

In relation to the other IRI values for routes 2 to 8, there is a variation in the values for routes 1 and 5. These show an intermediate road quality with different types of pavement roughness. For pavement roads with potholes the IRI values are also higher, as observed for route 8 ($IRI = 10.1\ m/km$).

Both the IRI methodology and objective evaluation, based on mass acceleration, for road quality classification indicate a strong correlation with subjective feelings. Therefore, the parameters obtained in this study are sufficiently accurate and effective for use as an auxiliary tool to express human assessments of road excitation.

6. CONCLUSION

During the design of a vehicle, the use of simulations has proved to bean important tool, as it is possible to obtain faster results, reducing the cost of the project and optimizing the product design. One of the factors that have contributed to increasing vehicle performance is the evaluation of the comfort experienced by the driver and passengers. A vehicle design needs to provide good ride quality and this requires a knowledge of road surface conditions. Therefore, the road condition has to be monitored and classified to ensure that international quality standards are met and this requires the use of an indicator to identify a relative quality of the road.

The international roughness index (IRI) standard is usually employed to measure road irregularities and this index has shown a strong relationship with the comfort of the vehicle user. In this regard, a high IRI value indicates low ride quality, while a low IRI value indicates acceptable or good ride quality. In this study, the methodology of indirect identification via the calculation of the IRI was performed by computing a two dof model (golden car model).A quarter car model based on the tested vehicle was adjusted from the mass vibration and sprung and unsprung masses to determine the road irregularity, which is employed as an input for the golden car model.

The measurements carried out in this study, in the city of Joinville, resulted in IRI values which represented the quality of the road conservation. A pavement surface with

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many irregularities, such as a cobbled street, provided IRI values around 15.0m/km (route 1), which is classified as a damaged pavement in Figure 8. In contrast, for a smoother road, such as an asphalt pavement, the IRI value was 2.9m/km (route 5), indicating anew pavement. Therefore, this methodology focuses on the application of analytical-based models to determine an index to classify road surface quality, using low-cost instrumentation.

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