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Sensing methodologies to determine automotive damper condition under vehicle normal operation

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ABSTRACT

Vehicles rely on the efficiency of dampers to dissipate energy from the motion of vehicle body and wheels, maintaining the vehicle more stable, and improving the contact between tires and the road surface. To achieve an effective monitoring of dampers (or shock absorbers), two different methodologies, capable of assessing, under vehicle normal operation, the condition of the automotive dampers are presented. The proposed methodologies are based in acceleration, temperature and pressure sensing to determine the shock absorber condition, and are therefore suitable for future implementation in low cost fabrication technologies. The results shown that it is possible to have an effective monitoring device, installed in the damper body, capable of continuously determining shock absorber status, and therefore enabling real time diagnosis. Such a diagnosis system can reduce the number of vehicles riding with defective suspension systems and increase the overall vehicle safety.

1. Introduction

The automotive suspension plays a crucial role in vehicle safety and driving comfort. Even if isolation from road vibration is definitely a very important aim, maintaining the contact between road surface and vehicle wheels is far more essential, as the control and stability of the vehicle relies totally upon it.

The suspension main components are dampers and springs, both positioned between the sprung mass (vehicle body) and the unsprung mass (wheel). Although the name shock absorber is commonly used for dampers, in reality vehicle shocks are absorbed by the deflection of the tyres and springs. The principal purpose of dampers is to dissipate energy. As the damper is compressed and extended, mechanical energy is converted into heat, damping the vehicle oscillations and improving road adhesion.

However, as any other mechanical system, suspension components are subjected to wear gradually. Shock absorbers in particular, due to its physical working concept, are prone to wear with time, distance driven and working conditions and suffer other damages like oil leakage. Dampers deterioration will result in excessive vehicle oscillations as a response to road disturbances, adversely affecting vehicle comfort, drivability and safety, with increased instability and longer breaking distances.

To assess the condition of shock absorbers, two methods are usually employed: dynamometer tests, which use an electric-hydraulic cylinder actuator or an electric motor, together with a crank mechanism, to run the shock absorber (that must be removed from the vehicle) through different input stimulus or sinusoidal frequencies. The measurement of the resultant force between the damper and the actuator as a function of velocity, gives accurate results and clear damper parameters identification [1–6].

On the other hand, ground suspension tests (for which shock absorber removal is not necessary) apply a shaking displacement to the tire at different frequencies and measure the tire contact force with the platform. The result, called adhesion, is a good measure of the suspension system status, but is not a sufficient indicator of the shock absorber condition [7,8].

Nozaki and Inagaki [9] presented a new concept test device capable of measuring the damping force; however, dedicated tools (vibration production device, computing unit, displacement sensor and the tire load cell) must be used, limiting the use and portability of such device.

Actually, no precise method to continuously evaluate dampers under normal operation exists; therefore, methodologies were developed for determining the damper condition during vehicle normal operation, to alert the driver whenever replacement is required.

The proposed methodologies can be implemented using an embedded autonomous device (which could be self-powered from the dampers scavenged energy), acquire relevant parameters and transmit them wirelessly to the automotive diagnostic electronic system.

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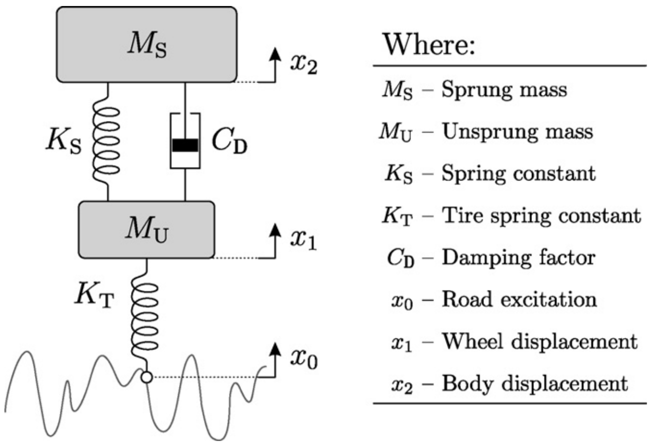

Fig. 1. One quarter of the vehicle suspension model.

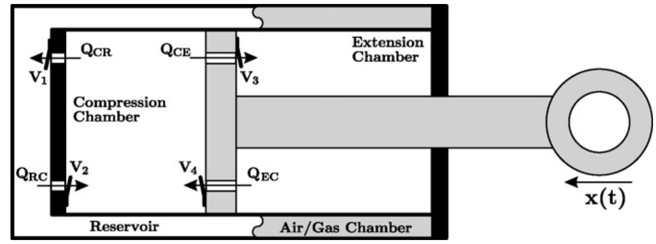

Fig. 2. Dual-tube internal shock absorber configuration.

The sensing part will be developed based in MEMS (Micro-Electro-Mechanical Systems) sensors, based on CMOS process, allowing batch fabrication of systems with small dimension, high quality, sensitivity and reproducibility. The small size, overall feasibility and reduced cost per unit are key factors for their selection and successful application, with proved results in many automotive applications [10–14].

2. Assessment methods

In order to study different ways of measuring vehicle damper's condition, two approaches were considered. The first consisted in the analysis of the suspension system as a whole, the second on the study of the damper internal working principles.

2.1. First method—suspension analysis

A theoretical analysis of an one quarter, two degrees of freedom vehicle suspension system, Fig. 1, was performed, considering that vibrations occur independently at each wheel.

The equations for the sprung and unsprung mass (M_U and M_S) force-balance are:

$$\begin{cases} M_S\,\ddot{x}_2 = K_S(x_1 - x_2) + C_D(\dot{x}_1 - \dot{x}_2) \\ M_U\,\ddot{x}_1 = K_T(x_0 - x_1) - K_S(x_1 - x_2) - C_D(\dot{x}_1 - \dot{x}_2) \end{cases} \quad (1)$$

By solving the above equations in the frequency domain, it is possible to determine the transmissibility T_R, output to input magnitude ratio between M_U and M_S accelerations ($\ddot{X}_1$ and $\ddot{X}_2$) in frequency domain:

$$T_R = \left|\frac{\ddot{X}_2(\omega)}{\ddot{X}_1(\omega)}\right| = \sqrt{\frac{1+(2\zeta r)^2}{(1-r^2)^2+(2\zeta r)^2}} \quad (2)$$

T_R is a function of the damping ratio, ζ, and the forcing frequency ratio, r. The damping ratio is a function of the damping factor, C_D, sprung mass, M_S, and the spring constant, K_S,

$$\zeta = \frac{1}{2}\frac{C_D}{\sqrt{M_S\,K_S}} \quad (3)$$

and r is the ratio between the excitation frequency f_r and the system natural frequency f_n:

$$r = \frac{f_r}{f_n} \quad (4)$$

In vehicle conventional passive suspension systems, the spring constant K_S has little variation during vehicle life. Vehicle mass M_S changes with vehicle load, this effect being more pronounced in the rear axle, as the front axle load is mainly determined by the powertrain mass (most common case: front engine mounting).

The damper condition is related with the damping factor C_D, so changes due to aging, wear or malfunction of the damper may be detectable by monitoring transmissibility values.

As the transmissibility is directly affected by the excitation frequency, a frequency domain analysis of the transmissibility is performed in order to determine its behavior for different excitation frequencies.

Therefore, as T_R is a function of the damping factor, a system that measures top ($\ddot{X}_2$) and bottom ($\ddot{X}_1$) acceleration in the shock absorber and computes the transmissibility in the frequency domain may be used as an indicator of shock absorbers condition.

2.2. Second method—damper analysis

As the above concept requires the use of two measuring points, an alternative method was developed that uses shock absorber internal pressure and wheel acceleration to determine shock absorber condition. Different models for the shock absorbers internal working principles were studied. Most of the models proposed in literature were developed to study particular features of the shock absorbers: models for using in vehicle dynamics studies [15,16], models of the dynamic characteristics for displacement-sensitive shock absorbers [17,18], thermal and heat transfer effects models [19,20], among others [21–24].

Most of the presented models require knowledge of detailed sets of the shock absorber parameters or experimentally determined parameters as a function of the results obtained in dynamometers. Therefore, a simple shock absorber model which correlates the damping properties with its main physical characteristics, internal pressures, and oil temperature was developed.

Using flow continuity Eq. (5), when a chamber is compressed the displaced volume, $A\dot{x}$, will be equal to the oil volumetric flow rate from that chamber, Q, corrected by the oil compressibility factor in the respective volume, where: V is the volume, K is the bulk modulus of oil and P is the pressure.

$$\frac{V}{K}\frac{dP}{dt} = A\dot{x} - Q \quad (5)$$

Considering the basic schematic of an internal dual-tube shock absorber as presented in Fig. 2, it can be reduced to four main valves, two in the piston and two separating the compression chamber from the outer reservoir. Therefore, in the compression stroke, for positive values of $\dot{x}(t)$, the oil flows from the compression chamber to the extension chamber, resulting in the volumetric flow rate Q_{CE}, and due to the difference of volumes between the two main chambers, the remaining oil is forced to flow to the reservoir, Q_{CR}. Note that the piston valve, V_3, offers small resistance to the oil flow so, in this stroke, the shock absorber force is controlled mainly by the compression chamber to reservoir valve V_1.

In the extension stroke (negative values of $\dot{x}(t)$), oil is compressed from the extension chamber to the compression chamber, resulting in the volumetric flow rate Q_{EC}, and as the oil coming from the extension hasn't enough volume to fill the bigger compression chamber volume, some oil will flow from the reservoir to the compression chamber, Q_{RC}. In this stroke the valve placed between the compression chamber and the reservoir, V_2, offers little flow resistance, being the damping force in the extension stroke controlled mainly by the valve placed in the piston, V_4.

Considering that oil is incompressible, the oil volumetric flow rate in the four main valves could be expressed as a function of the compression and extension chambers cross section areas, A_C and A_E, and the piston velocity, $\dot{x}(t)$:

$$\begin{array}{ll} \text{Compression} & \text{Extension} \\ \begin{cases} Q_{CR} = (A_C - A_E)\dot{x} \\ Q_{CE} = A_E\dot{x} \end{cases} & \begin{cases} Q_{RC} = (A_C - A_E)\dot{x} \\ Q_{EC} = A_E\dot{x} \end{cases} \end{array} \quad (6)$$

Based on the Bernoulli equation:

$$Q = C_d A_0 \sqrt{\frac{2}{\rho}\Delta_P} \quad (7)$$

was used to correlate the volumetric flow rate in the valves, Q, with the corresponding pressure drop, Δ_P. The discharge coefficient C_d depends mainly on the valves orifices geometry and Reynolds number (for automotive shock absorbers, constant value for C_d around 0.6 is often used [1,15]). A_0 is the valve orifice area and ρ is the oil density.

The reservoir is half filled with air/gas and sealed from the atmosphere, causing oil pressure in the reservoir, P_R, to be equal to the gas pressure, P_a. Therefore the reservoir oil pressure can be determined from the equation of the ideal gas, correcting the air/gas volume V_{a0} with in and out flow rates from the compression chamber, Q_{RC} and Q_{CR}, and the variation of oil volume with temperature, ΔV_{Oil}. The equation for the gas and reservoir oil pressure will come as:

$$P_a = \frac{m_a R T}{V_{a0} - \int Q_{RC}\,dt + \int Q_{CR}\,dt + \Delta V_{Oil}} \quad (8)$$

Knowing the oil pressure in the reservoir, P_R, and the flow rates through each valve is possible to obtain the corresponding pressure drops, Δ_{PXY}, so the pressure in the extension (P_E), and compression chamber (P_C), are obtained from

$$\begin{array}{ll} \text{Compression} & \text{Extension} \\ \begin{cases} P_C - P_R = \Delta_{PCR} \\ P_C - P_E = \Delta_{PCE} \end{cases} & \begin{cases} P_E - P_C = \Delta_{PEC} \\ P_R - P_C = \Delta_{PRC} \end{cases} \end{array} \quad (9)$$

As oil flow between shock absorber cameras is a function of oil density, equation

$$\rho_1 = \rho_0/(1 + \beta(t_1 - t_0)) \quad (10)$$

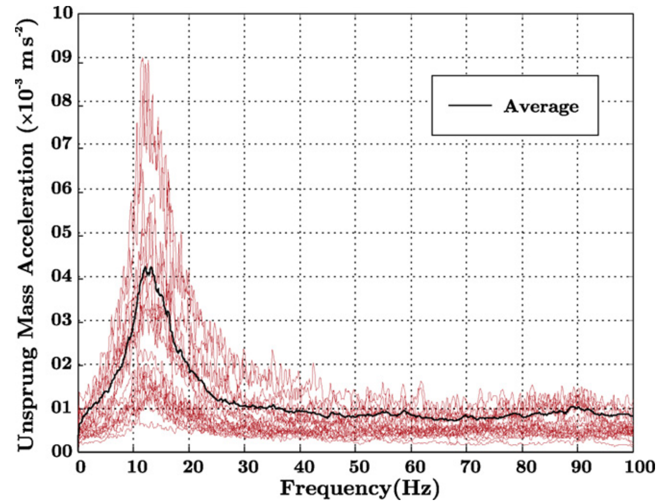

Fig. 3. Road input excitation power spectrum.

was used to find oil density change with temperature, where ρ_1 is the final density, ρ_0 is the initial density, β is the volumetric temperature expansion coefficient, t_1 is the final temperature and t_0 is the initial temperature.

The frictional forces between the piston and the inner chambers and between the piston rod and the shock absorber sealing were obtained from

$$F_{friction} = \mu A \frac{\dot{x}}{e} \quad (11)$$

where A is the contact area, μ is the dynamic viscosity of the oil and e is the oil thickness between the contact surfaces.

A look up table was used with the temperature/viscosity diagram for standard ISO hydraulic oils to determine changes in the oil dynamic viscosity μ with temperature.

As a result, knowing the damper internal pressures, cross section areas and frictional forces, the damping force, $F_{damping}$, can be obtained from:

$$F_{damping} = P_C A_C - P_E A_E + F_{friction} \quad (12)$$

As in a shock absorber the damping factor , C_D, can be obtained as a function of the damping force and excitation velocity,

$$C_D = \frac{F_{damping}}{\dot{x}} \quad (13)$$

considering that dampers are usually asymmetric in operation, as damping is higher in extension than in compression [1], $F_{damping}$ is mainly determined by the pressure in the extension

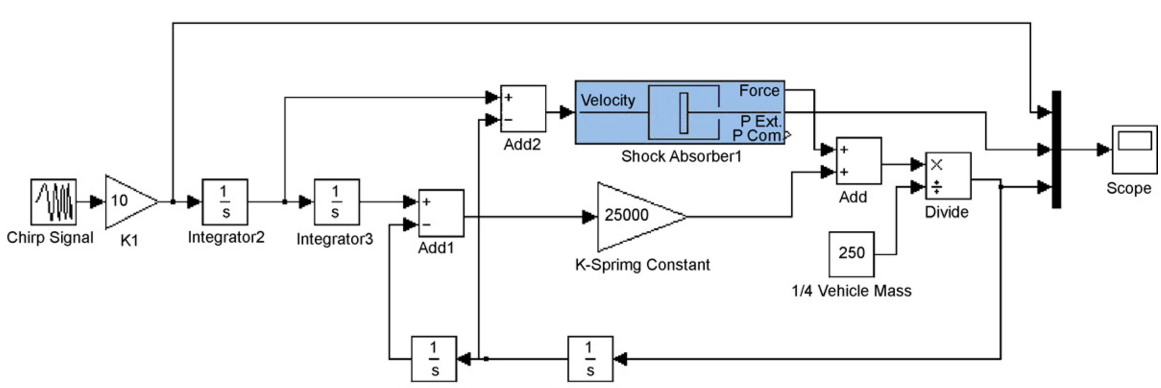

Fig. 4. Matlab Simulink suspension and damper models.

chamber, P_E, and as velocity can be obtained from input acceleration, it should be possible to determine de damping factor, a measure of the damper condition, as a function of these two parameters.

2.3. Validation

To know the excitation applied to shock absorbers under normal operation conditions, essays were made with different vehicles driven on multiple condition roads, measuring sprung mass acceleration (road excitation) and determining the power spectral densities. Test showed that (Fig. 3) there is a broad peak of excitation in the 10–15 Hz range, corresponding to the sprung mass resonance frequency.

A Matlab Simulink model was developed to describe suspension behavior (Fig. 4), and the previously presented equations were employed for the shock absorber model. The model was used to validate the possibility of determining shock absorber condition by knowing transmissibility between sprung and unsprung mass accelerations or determining damping force by knowing wheel acceleration and pressure in the extension chamber. Broad range acceleration (0–25 Hz) was used as input, and damping force, extension chamber internal pressure and sprung mass acceleration were obtained as output.

Fig. 5 shows the computed $\ddot{X}_2/\ddot{X}_1$ transmissibility between the two accelerations for different condition shock absorbers. Transmissibility dependance with load and spring constant was also verified, Fig. 6, and results showed that spring constant has little influence in the frequency range of interest, Fig. 7. Moreover, in conventional automotive suspension systems, spring constant has small variation with time.

A similar study was made for the $P_E/\ddot{X}_1$ transmissibility, and Fig. 8 shows simulated transmissibility between wheel acceleration and extension chamber pressure for different condition shock absorbers.

The model was also validated for the displacement-force and velocity–force diagrams (not shown) usually employed on shock absorber characterization.

From the above analysis, two methods were chosen and validated to assess shock absorber condition: computing the transmissibility between the sprung and unsprung masses accelerations and computing the transmissibility between the wheel acceleration and the shock absorber pressure in the extension chamber.

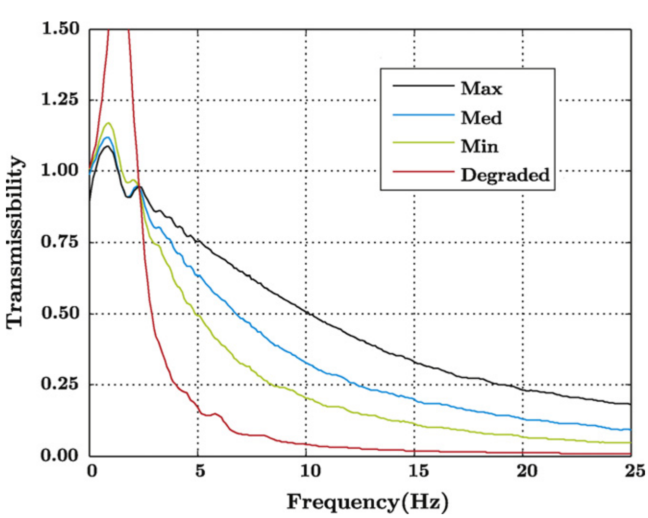

Fig. 5. Simulated $\ddot{X}_2/\ddot{X}_1$ transmissibility for different shock absorbers condition.

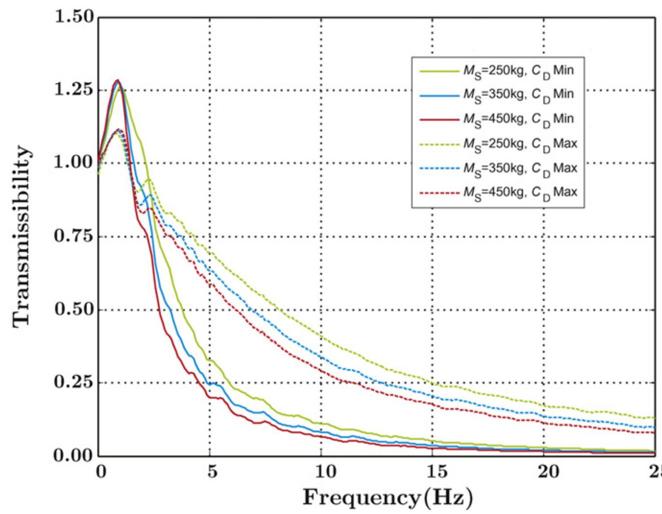

Fig. 6. Simulated load effect on $\ddot{X}_2/\ddot{X}_1$ transmissibility.

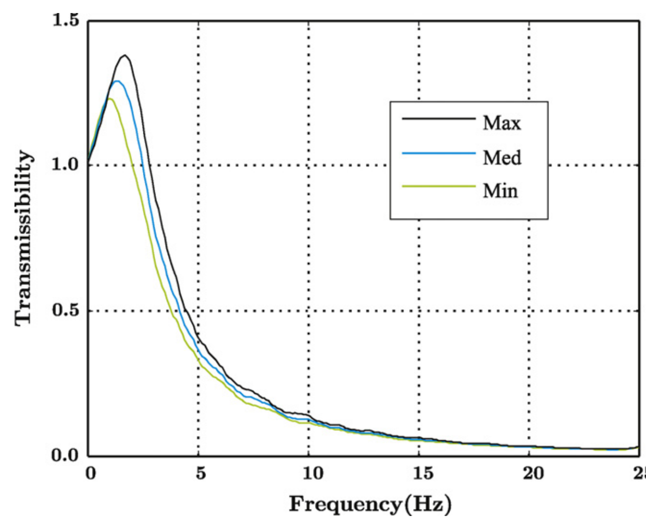

Fig. 7. Simulated spring constant effect on $\ddot{X}_2/\ddot{X}_1$ transmissibility.

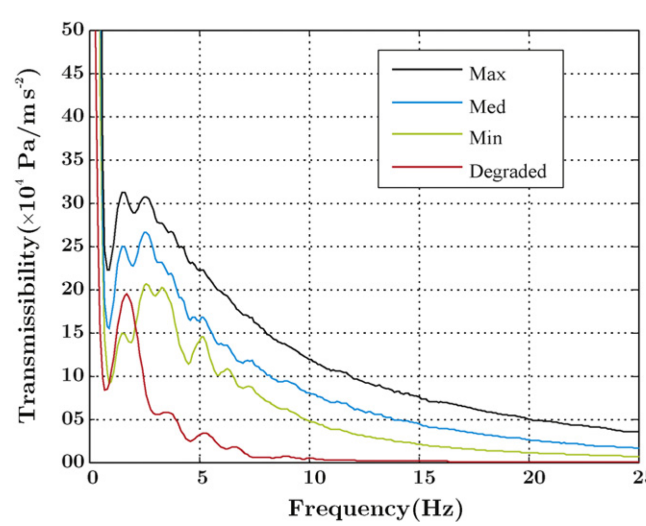

Fig. 8. Transmissibility $P_E/\ddot{X}_1$ for different shock absorber conditions.

Fig. 9. Instrumented shock absorber installed in a test vehicle.

3. Experimental results

To validate the above methods and to verify the feasibility of the in-vehicle shock absorber analysis in a multi-road excitation scenario, several experiments were conducted. Essays were conducted using several shock absorbers in different known conditions, with three different vehicles, a 1995 VW Polo, a 1996 FIAT Punto and a 1999 VW Sharan. The vehicles were driven on a 36 Km tour of differentiated road profiles, from very bad condition roads to city roads and highways.

A KONI dual-tube adjustable shock absorber was modified to measure both compression and extension chambers pressures and oil temperature. This shock absorber was subsequently tested in a dynamometer and in the VW Sharan rear axle. The installation of the instrumented shock absorber is depicted in Fig. 9.

Data was acquired using a NI USB6009 data acquisition board and signal conditioning electronics connected to a laptop running LabView. Two 0265 005 303 (Bosch, Germany) pressure sensors were used to measure internal chambers pressure, and a NTC M12 (Bosch, Germany) temperature sensor was used to measure internal oil temperature. The acceleration was measured with ADXL78 (Analog Devices, USA) accelerometers.

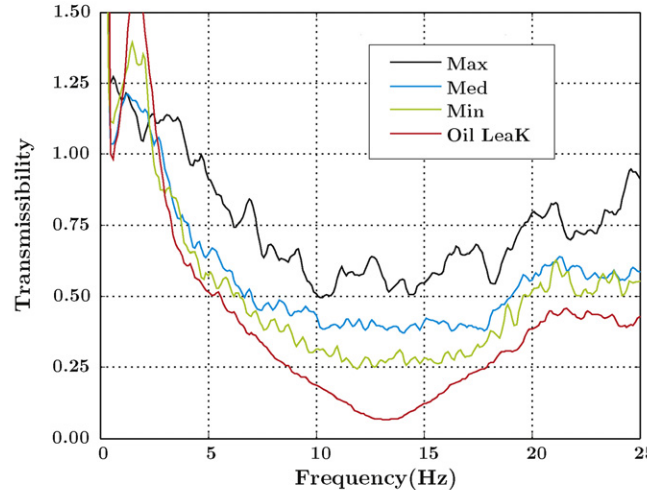

Fig. 11. Measured transmissibility $\ddot{X}_2/\ddot{X}_1$ for different dampers condition.

To verify the proposed shock absorber model, the modified shock absorber was run in a dynamometer. Fig. 10 shows measured and simulated extension and compression chambers internal pressures for a sinusoidal 1.8 Hz input, revealing a good correspondence between the model and the shock absorber behavior. The pressure peak in extension chamber is higher than the pressure peak in compression chamber, and, in the compression stroke, pressures in both chambers are almost equal due to the small restriction of the compression-to-extension valve.

3.1. Road test results

Fig. 11 shows the results of the average verified $\ddot{X}_2/\ddot{X}_1$ transmissibility for four different damper status, showing that this parameter is a good indicator of the damping condition, more evident in the 10–15 Hz range, corresponding to the unsprung mass resonance frequency of the test vehicle. Results showed that this method can be used to detect shock absorber condition. Load influence was verified, and proved to be notorious, Fig. 12, especially when dampers are in good condition, showing that load effect must be taken into account in order to avoid false results.

The calculation of the transmissibility between road acceleration and extension chamber pressure gave very fine results (Fig. 13), where, as expected, transmissibility diminishes with damping condition deterioration. It also showed good independence with sprung

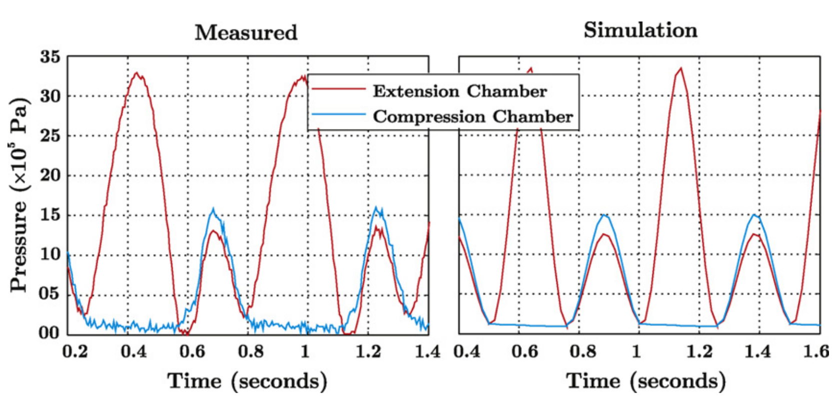

Fig. 10. Acquired vs simulated damper internal pressures.

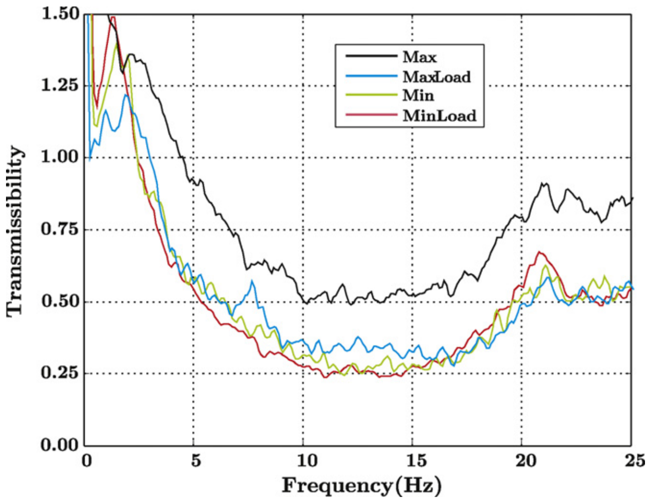

Fig. 12. Load influence on $\ddot{X}_2/\ddot{X}_1$ transmissibility.

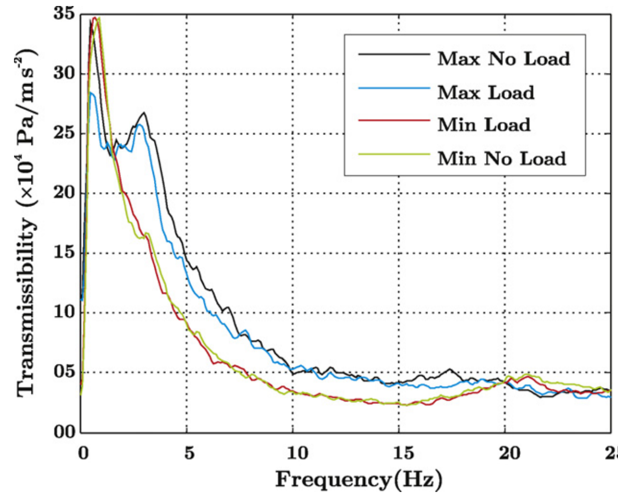

Fig. 14. Transmissibility $P_E/\ddot{X}_1$ for different vehicle loads.

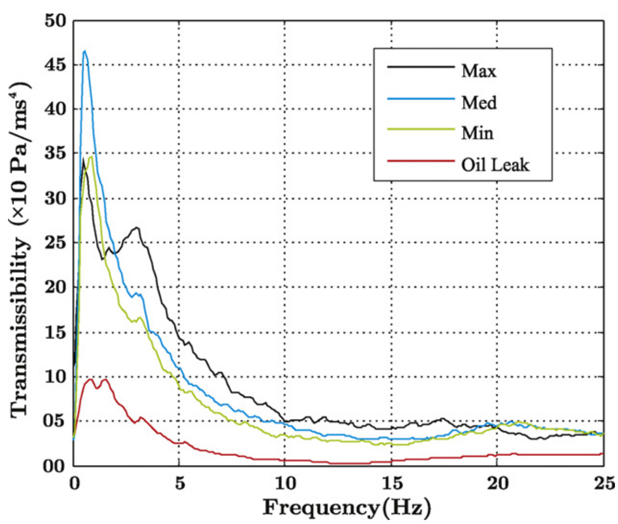

Fig. 13. Measured transmissibility $P_E/\ddot{X}_1$ for different dampers condition.

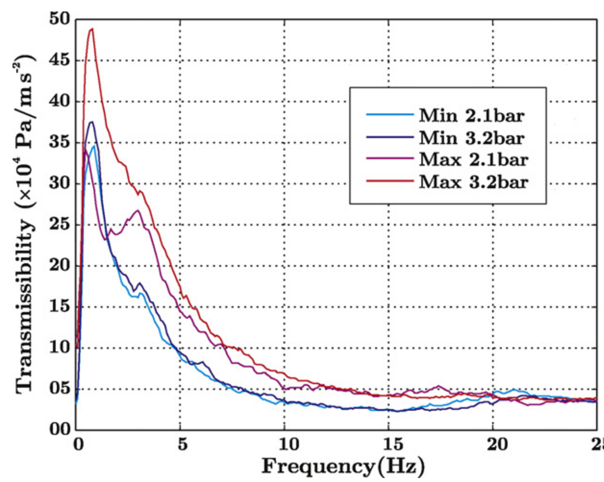

Fig. 15. Transmissibility $P_E/\ddot{X}_1$ for different tire pressures.

mass variation, Fig. 14, and independence with tire pressure, as shown in Fig. 15.

A key parameter that must be measured is temperature, as it determines oil volume and density, and subsequently the damping force. Fig. 16 shows the influence of temperature in shock absorber transmissibility between road acceleration and extension chamber pressure. As expected, results show shock absorber performance degradation with temperature, so an oil temperature sensor is mandatory to the accuracy of the methods.

4. Discussion

The first proposed method proved to be a reliable way of determining shock absorber status. It is easily implemented, as no modification in the shock absorber construction or installation is required. One of the sensing modules may simply be installed attached to the shock absorber body, in order to measure unsprung mass acceleration and shock absorber temperature, while the other may be installed in the top mounting support of the shock absorber. In the case where there is already an accelerometer installed in the vehicle body, that data may be used to determine sprung mass acceleration, simplifying the installation procedure even more. However, synchronization during data acquisition is mandatory, and that may pose a problem to a system that is intended to be wireless, and with low power availability. Also, as verified, load effects must be taken into account, implying the use of a vehicle load sensor. However, in most cases, the sensor is already installed

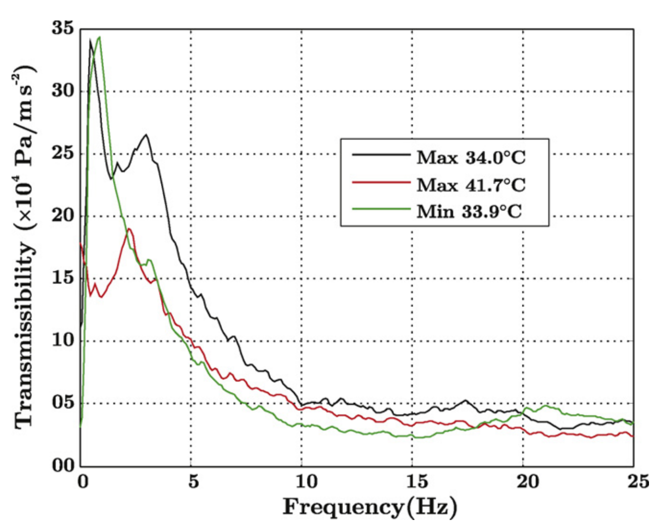

Fig. 16. Temperature effect on $P_E/\ddot{X}_1$ transmissibility.

in the vehicle, being a part of the braking distribution system and/or being used to correct the headlamps alignment.

The second proposed method is interesting as it gives accurate results, and enables just one measuring point (extension chamber pressure, wheel acceleration and temperature), making it more suitable to implementation through an embedded system. However, the installation of the measuring system requires modification in the shock absorber construction and assembly processes, as an access channel to the inner chamber pressure is required.

5. Conclusions and final remarks

This paper describes part of the work that is being carried out to achieve a complete embedded smart shock absorber condition sensor system.

Two methodologies were discussed and the results compared. Simulation and experimental results obtained with such measurement strategies were proven reliable and practicable to assess shock absorber condition under normal vehicle operation.

Measurement systems for the physical quantities necessary to assess shock absorber condition can be easily developed and full implemented with MEMS devices built over standard CMOS technologies, reducing fabrication costs.

A system to harvest available energy, e.g. in the form of mechanical vibration or temperature, thus making the device completely independent in terms of power supply, is currently being developed.

A wireless solution can provide a very significant reduction of wiring in vehicles, with the advantage of cost reduction and weight, allowing even an increase of reliability and reduction of assistance needed during maintenance.

An embedded sensor system capable of performing the presented methodologies will be a remarkable vehicle electronic feature, as it can clearly inform the driver when is necessary to replace shock absorbers. Such a self powered wireless sensing device will have the lifetime expectancy of the vehicle, becoming a maintenance free device.

To our best knowledge, this is the first time (patent pending) that such a system is proposed. This shock absorber monitoring device will be a breakthrough in vehicle comfort, drivability and overall safety.

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