24th ABCM International Congress of Mechanical Engineering
December 3-8, 2017, Curitiba, PR, Brazil

COBEM-2017-0286
SELF-POWERED ACTIVE SUSPENSIONS ON SMART VEHICLES

José Carlos de Carvalho Pereira
João Victor Pereira da Silva
Department of Mechanical Engineering, Federal University of Santa Catarina, Campus Reitor João David Ferreira Lima, 88040-900, Florianópolis, BR
carlosp@emc.ufsc.br / joao.victor.p.silva@grad.ufsc.br

Abstract. *Due to the limitations of electric energy suppliers in electric car systems, energy recovery from induced vibration in vehicle suspension has gained considerable attention by researchers in recent years. Among the new technologies investigated for this application in automotive fields, the piezoelectric vibration energy harvesting device has increased enormously. In this context, this paper presents an evaluation on the comfort improvement in a self-powered active double-wishbone type vehicle suspension with a piezoelectric element as the conversion mechanism and a simple actuator as the restored force. A mathematical model for the suspension is developed and modal analysis is used to compute the results. The performance is calculated based on an impact load from the road surface. The control system in calibrated as so the power requirement is the same as the generated power. The regenerated value reaches almost 80W RMS and, when it is fully reintroduced into the system, reductions of up to 20% in the RMS value of the vertical acceleration, main comfort parameter, along with 15% improvement in the displacement and over 50% reduction of the settling time are observed.*

Keywords: *Harvesting energy, Self-Powered, Active suspension, Double-Wishbone.*

1. INTRODUCTION

Smart structures have been widely studied in a vast range of applications in recent engineering systems. Piezoelectric materials have the property of coupling mechanical and electrical aspects and can be a powerful tool in such structures. An extensive variety of sensors and actuators in vibration control systems, including dynamic and acoustic applications, use this special constitutive relation to measure the vibrations of a structure by giving an electric potential response. Therefore, can control the vibration level by generating a closed-loop control system based on the voltage signal, by either applying a restored force or altering the dynamic properties of the system. Harvesting energy is a crescent engineering problem, as a result, there is a wealth of literature in this domain of the science such as Li et al. (2015), Davidson and Mo (2014), Zuo and Tang (2013), Nestorović et al. (2013), Reinhardt et al. (2013), Anton and Sodano (2007), Soldano and Inman (2004)-a and Soldano and Inman (2004)-b. Hence, the ability to restore this otherwise wasted mechanical energy through the piezoelectric crystals can be a good asset to power devices that request a low level of energy. Ottman et al. (2003), Roundy et al. (2003), Ching et al. (2002), Glynne–Jones et al. (2001) have studied the capabilities of the electromechanical coupling effect of these materials. Roundy and Wright (2004) stated the energy density based on piezoelectric harvesting is 43% higher than the one based on electromagnetic harvesting.

Regeneration in vehicle suspension systems is also an explored topic. Goldner et al. (2001) stated that approximately 200W could be generated in the four dampers of a passenger car traveling at low quality roads at a speed of 13.4m/s. Considering an impact scenario, the same evaluated in this study, (Hsu, 1996) reached a value of up to 100W in a highway at 16m/s. Kawamoto et al. (2007), Martins et al. (2006), Goldner et al. (2001), Li et al (2015) have presented possible regeneration values with a vast range, achieving a difference of 16700%. However, the common conclusion is that the energy generated is not neglectable and has potential to power devices inside vehicles.

Active suspension systems have for years being applied commercially. However, the necessity of various sensors, complex mechanisms and, specially, a high-energy demand transform them into very expensive systems. Thus, recent researches aim to develop control systems that will work with a low energy level. Considering regeneration as a strong alternative for these models, Yan et al (2017) and Zhang et al (2012) designed active-regenerative systems, creating criteria for a self-sustained suspension and showing that it is applicable.

Thus, the objective of this paper is to evaluate the feasibility of a regenerative active suspension, with a piezoelectric crystal acting as both the sensor for control input and the power source. A more comprehensive suspension model in taken into account, with elastic elements emulating a double-wishbone type suspension. The vertical accelerations of the lumped masses are the main comparison parameter, as they represent the overall comfort inside the vehicle. The control system is calibrated as to insure self-sustainability and the performance with and without the actuator is compared.

2. MATHEMATICAL FORMULATION

The suspension model is based on a Double-Wishbone construction. It is composed of two beams, representing both the arms; two spring elements; one strut component; two lumped mass elements and the piezoelectric crystal. The crystal is set between nodes 5 and 7 in the discretized system. It is represented by a uniaxial finite element in the form of a disk with diameter D and thickness h with aspect ratio (D/h) no smaller than 10. Both nodes containing the element have a mechanical load and an electric charge C_i associated. Either the electrical or the cinematic displacement are written as d_i. The properties that define the crystal's behavior are the Young's Modulus E, the piezoelectric constant C_p and the dielectric constant C_d. The whole system can be seen on Fig 1 and the piezoelectric element in detail in Fig 2.

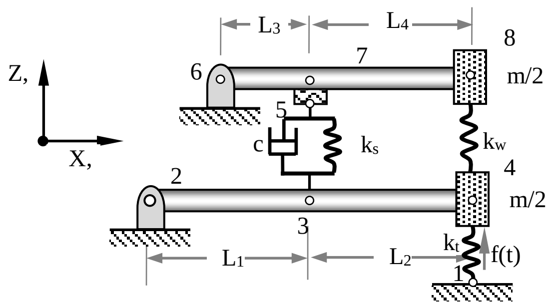

Figure 1. Mathematical model of a double wishbone suspension.

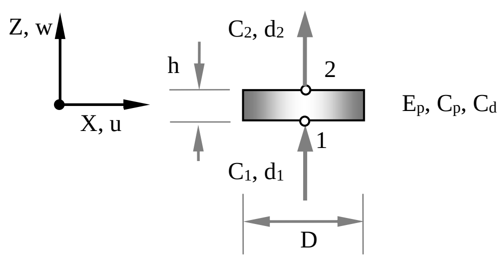

Figure 2. Piezoelectric crystal element

The IEEE Standard on Piezoelectricity, ANSY (1987) gives the electromechanical coupling effect governing equations, Eq (1) and Eq (2).

$$\sigma = E.\frac{\partial w}{\partial z} - C_p \frac{\partial V}{\partial z} \tag{1}$$

$$Q = C_p.\frac{\partial w}{\partial z} + C_d \frac{\partial V}{\partial z} \tag{2}$$

Where σ is the normal stress, Q the electric displacement, w the axial displacement and V the electric potential. The mechanical strain energy U_m and the electric energy U_e of the piezoelectric material element are written as:

$$U_m = \frac{1}{2}\int_V \sigma.\frac{\partial w}{\partial z}dV = \frac{1}{2}\int_V \left(E_p.\frac{\partial w}{\partial z} - C_p.\frac{\partial V}{\partial z} \right)\frac{\partial w}{\partial z}dV \quad (3)$$

$$U_e = \frac{1}{2}\int_V Q.\frac{\partial V}{\partial z}dV = \frac{1}{2}\int_V \left(C_p.\frac{\partial w}{\partial z} + C_d.\frac{\partial V}{\partial z} \right)\frac{\partial V}{\partial z}dV \quad (4)$$

A linear approximation of the displacement and the electric potential along the thickness of the piezoelectric element is assumed, hence the three necessary stiffness matrices of the system are obtained by the Lagrange equations and are shown in Eqs 5 through 7.

$$\left[k_m\right] = \frac{E_p.S_p}{h}\begin{bmatrix} 1 & -1 \\ -1 & 1 \end{bmatrix} \quad (5)$$

$$\left[k_{m\_el}\right] = -\frac{C_p.S_p}{2.h}\begin{bmatrix} 1 & -1 \\ -1 & 1 \end{bmatrix} \quad (6)$$

$$\left[k_{el}\right] = \frac{C_d.S_p}{h}\begin{bmatrix} 1 & -1 \\ -1 & 1 \end{bmatrix} \quad (7)$$

Where $[k_m]$, $[k_{m\_el}]$ and $[k_{el}]$ are mechanical stiffness, electromechanical coupling stiffness and electric stiffness elementary matrices, respectively, and Sp is the cross section area of the piezoelectric crystal.

The matrices allow the development of the full motion equation of the suspension and it is represented in Eq 8.

$$\left[M\right]\left\{\ddot{P}\right\} + \left[C\right]\left\{\dot{P}\right\} + \left[K\right]\left\{P\right\} = \left\{F(t)\right\} \quad (8)$$

[M], [C] and [K] are the global matrices of mass, damping and stiffness, respectively, <F(t)> is the applied force vector and <P(t)> represent the displacements, electric V and mechanical U accordingly with the degree of freedom.

$$\left\langle P \right\rangle = \left\langle \left\langle U \right\rangle, \left\langle V \right\rangle \right\rangle = \left\langle w_1, w_2, \theta_2, w_3, \theta_3, w_4, \theta_4, w_5, w_6, \theta_6, w_7, \theta_7, w_8, \theta_8, V_5, V_7 \right\rangle \quad (9)$$

Equation 8 is coupled by the electromechanical elementary matrix into the global stiffness matrix by displacements and electric potentials of the piezoelectric element. Considering there are only loads of the mechanical nature, the equations can be decoupled as followed in Eq 10 and Eq 11

$$[M_m]\{\ddot{U}\} + [C_m]\{\dot{U}\} + [K_m]\{U\} + [K_{m\_el}]\{V\} = \{F_m(t)\} \quad (10)$$

$$[K_{m\_el}]^t\{U\} + [K_{el}]\{V\} = \{0\} \quad (11)$$

Merging both the equations above it is possible to write the motion expression of the system and it takes the form of Eq 12.

$$[M_m]\{\ddot{U}\} + [C_m]\{\dot{U}\} + \left[[K_m] - [K_{m\_el}][K_{el}]^{-1}[K_{m\_el}]^t\right]\{U\} = \{F_m(t)\} \quad (12)$$

Previously, the modal analysis is performed and the two first vibration modes are used to obtain the response in the time domain of the decoupled equation of motion.

$$[m]\{\ddot{p}(t)\} + [c]\{\dot{p}(t)\} + [k]\{p(t)\} = \{f(t)\} \quad (13)$$

Where [m], [c] and [k] are modal mass, modal damping and modal stiffness matrices, respectively. {f(t)} is the modal force vector and {p(t)} is the generalized coordinate vector that is related to mechanical degrees of freedom by

$$\{u(t)\} = \sum_{j=1}^{2}\{U\}_j\, p_j(t) = \{U\}_1\, p_1(t) + \{U\}_2\, p_2(t) = [U]\{p_j(t)\} \quad (14)$$

Where $\{U\}_j$ is the j^{th} eigenvector of the system.

The time domain response can be calculates using integration methods, such as the Newmark Method, Bathe (1996) or Taylor expansion series, Lalanne et al (1986). Voth approaches yield to a numerical step-by-step solution for the <p> vector.

The mechanical load applied in the system f(t) takes the form of an impact force intensity F_o. It acts over the lumped mass closest to the ground, on node 4 during a short period, representing a sudden alteration on the road profile. These data are used to obtain the velocity as an initial condition in solving Eq 13 on a time response domain, Paultre (2010).

The response in the time domain in respect to mechanical degrees of freedom is calculated by using Eq 14 and after that, the electric potential is obtained as follows.

$$\{V(t)\} = -[K_{el}]^{-1}[K_{mel}]^{t}\left\{\sum_{j=1}^{2}\{U\}_j . p_j(t)\right\} \tag{15}$$

The control system used is based on the sky-hook type, Eq 16, it acts as closed loop feedback on one lumped mass and is based on the derivative of the control variable. The main difference being, the error is calculated through the electric displacement, or the electric potential generated by the piezoelectric material, as seen on Fig 3.

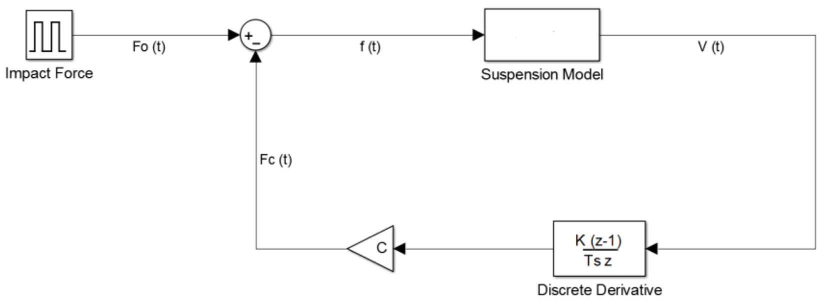

Figure 3. Control Loop

Therefore, the crystal acts both as the sensor for the loop and the energy supply for the control force, that is expressed as:

$$F_c(t) = C\dot{V}_5(t) \tag{16}$$

Where F_c is the force reintroduced in the system and C is the control constant. The final equation of movement for the complete system is, hence:

$$[M_m]\{\ddot{U}\} + [C_m]\{\dot{U}\} + \left[[K_m] - [K_{m\_el}][K_{el}]^{-1}[K_{m\_el}]^{t}\right]\{U\} = \{F_m(t)\} - C\{\dot{V}\} \tag{17}$$

The goal is to evaluate the control improvement without external energy inputs. Hence, the C constant is tuned in order that the root-mean-square (RMS) value of the power generated, P_e, is equal to the RMS value of the power required to active the control force, P_c.

The electrical power generated by the suspension can be calculated as the partial derivative of the electric energy, presented by Eq. 4, in respect to the time:

$$P_e = \frac{\partial U_e}{\partial t} = \frac{1}{2}\int_V \frac{\partial}{\partial t}\left(Q.\frac{\partial V}{\partial x}\right)dV = \frac{1}{2}\int_V \frac{\partial}{\partial t}\left(C_p.\frac{\partial u}{\partial x} + C_d.\frac{\partial V}{\partial x}\right)\frac{\partial V}{\partial x}dV \tag{18}$$

The development of Eq. 18 follows:

$$P_e = \frac{1}{2}\int_0^h\int_{S_p}\left[C_p.\left(\frac{\partial}{\partial t}\left(\frac{\partial u}{\partial x}\right).\frac{\partial V}{\partial x} + \frac{\partial u}{\partial x}.\frac{\partial}{\partial t}\left(\frac{\partial V}{\partial x}\right)\right) + C_d.\frac{\partial}{\partial t}\left(\frac{\partial V}{\partial x}\right)^2\right]dS.dx \tag{19}$$

Relating local coordinate system to global coordinate system and respective displacements, the expression of the electrical power is as:

$$P_e = \frac{1}{2}\frac{C_p S_p}{h}\left[(\dot{w}_7 - \dot{w}_5)(V_7 - V_5) + (w_7 - w_5)(\dot{V}_7 - \dot{V}_5)\right] + \frac{C_d S_p}{h}(V_7 - V_5)(\dot{V}_7 - \dot{V}_5) \tag{20}$$

Where $\dot{w}_5$, $\dot{w}_7$, $\dot{V}_5$ and $\dot{V}_7$ are time derivatives of displacement and electric potential of nodes of the piezoelectric element. In this example of conventional vehicle suspension, the electrical displacement in node 5 is null. Thus, the final expression of the electrical power is as:

$$P_e = \frac{1}{2}\frac{C_p S_p}{h}\left[(\dot{w}_7 - \dot{w}_5)V_7 + (w_7 - w_5)\dot{V}_7\right] + \frac{C_d S_p}{h}V_7\dot{V}_7 \tag{21}$$

The time derivatives are calculated using Eq 22 and Eq 23.

$$\begin{Bmatrix} \dot{w}_5(t) \\ \dot{w}_7(t) \end{Bmatrix} = \begin{Bmatrix} w_5|_1 \\ w_7|_1 \end{Bmatrix}\dot{p}_1(t) + \begin{Bmatrix} w_5|_2 \\ w_7|_2 \end{Bmatrix}\dot{p}_2(t) \tag{22}$$

$$\begin{Bmatrix} \dot{V}_5(t) \\ \dot{V}_7(t) \end{Bmatrix} = -[K_{el}]^{-1}[K_{m\_el}]^t \left\{ \begin{Bmatrix} w_5|_1 \\ w_7|_1 \end{Bmatrix}\dot{p}_1(t) + \begin{Bmatrix} w_5|_2 \\ w_7|_2 \end{Bmatrix}\dot{p}_2(t) \right\} \tag{23}$$

Where $w_5|_1$, $w_5|_2$, $w_7|_1$ and $w_7|_2$ are displacements of node 5 and 7 related to vibration modes 1 and 2, respectively.

The control system power demand is calculated using the classic mechanical definition and is evaluate at each time step according to:

$$P_c = \frac{\partial(F_c w_4)}{\partial t} = \frac{\partial F_c}{\partial t}w_4 + \frac{\partial w_4}{\partial t}F_c \tag{24}$$

Hence, the condition to calibrate the control force levels and to achieve a self-sustained system, where T is the total analyzed time, is:

$$P^2{}_{c\_rms} = P^2{}_{e\_rms} = \frac{1}{T}\int_0^T P_{c\_rms}{}^2 dt = \frac{1}{T}\int_0^T P_{e\_rms}{}^2 dt \tag{25}$$

ISO 2631 sets the main human comfort parameter when considering whole body vibrations to be the RMS value of the acceleration it suffers. For this case in particular, the vertical acceleration of the lumped masses will be considered as representative of the car interior. Hence, the goal is to reduce the RMS value of $\ddot{w}_4$, that is calculated similarly to Eq 25.

$$\ddot{w}^2{}_{4\_rms} = \frac{1}{T}\int_0^T \ddot{w}_4{}^2 dt \tag{26}$$

The analysis also extends to the settling time, which is the instant where the vertical movement does not reach 20% of the maximum displacement value anymore, and the maximum displacement itself. All calculates at node 4.

3. RESULTS

The beam elements Young's modulus is E = 200 GPa and have density ρ = 7800 kg/m3. The geometrical properties of the upper arm are L_1= 0.25 m and rectangular section of 0.18 x 0.05 m, the geometrical properties of the lower arm, are L_2 = 0.25 m and rectangular section of 0.18 x 0.05 m. The spring constant of the tire is k_t = 2000 N/m, of the suspension is k_s = 40000N/m and of the wheel is $k_w = 1x10^9$ N/m and the viscous damping constant is c = 4000 N/m/s. The lumped mass is 30 Kg. The dielectric constant $C_d = 1.3x10^{-8}$, and the piezoelectric constant is C_p = 23.3. The piezoelectric disc has 0.05 m of diameter and thickness of 0.005 m, and Young's Modulus of 126 GPa.

The impulsive force applied is Fo= 200 N and the period of time is Δt = 1s. A MATLAB® code was developed to obtain the results shown below. Figure 4 shows the time response domain of the system in respect to the displacement of node 4 without control force.

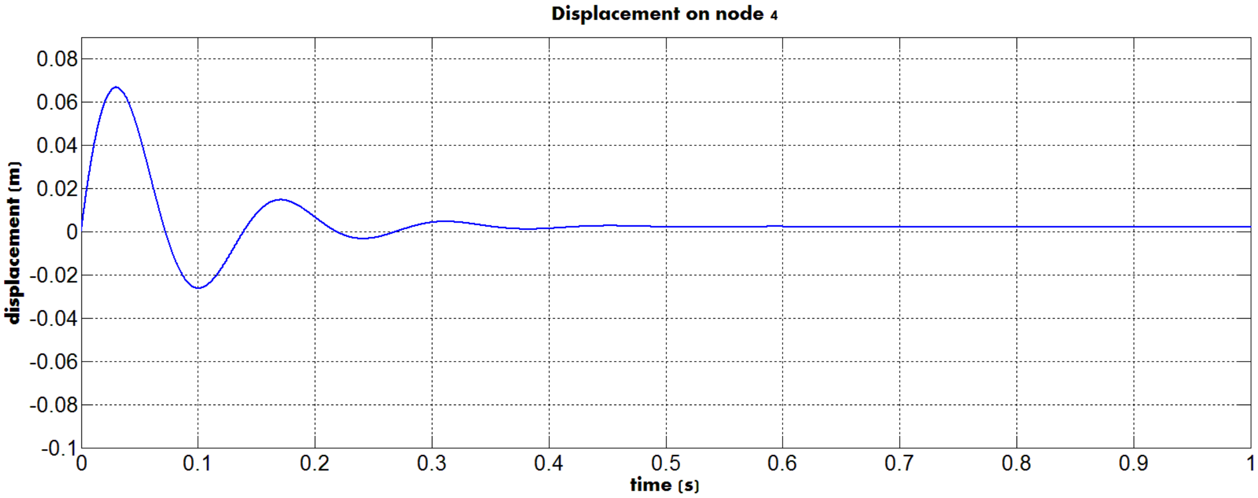

Figure 4. Displacement on node 4

The observed maximum displacement is of 0.066m, the settling time is 0.184s and the RMS value of the acceleration is 22 m/s^2.

Figure 5 displays the tuning of the control constant in order to achieve a null energy balance in the system. Both the power generated and consumed are on the plot. No energy loss in considered during the reintroduction of the power into the system. The value of C stands at 0.0223, at the point, which both systems act at a rate of 79.9W RMS.

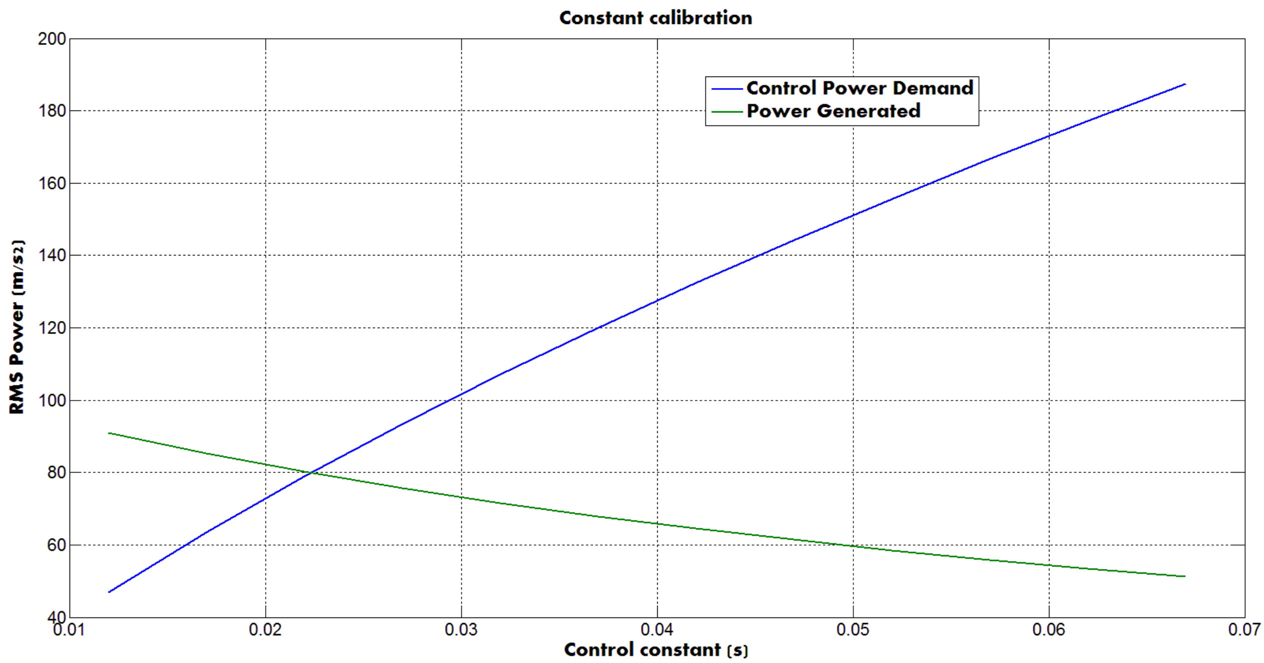

Figure 5. Constant calibration for self-sustained scenario

The new displacement, with the self-sustained control force applied is shown on Fig 6.

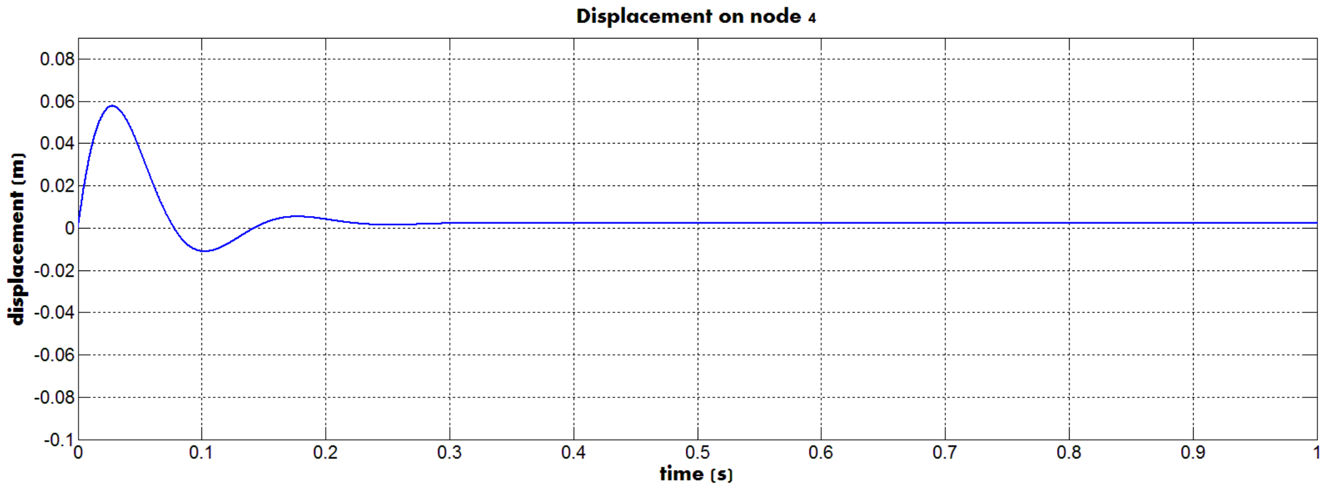

Figure 6. Displacement on node 4 with force feedback

The new values are: maximum displacement is of 0.059m, the settling time is 0.078s and the RMS value of the acceleration is 17,67 m/s^2.

4. DISCUSSION AND CONCLUSIONS

This paper has shown an evaluation on the comfort improvement in a self-powered active double-wishbone type vehicle suspension with a piezoelectric crystal for harvesting energy from environment vibration and a simple actuator as the restored force. The control system in calibrated as so the power requirement is the same as the generated power by the piezoelectric crystal.

It can be observed that the reduction of the maximum displacement reaches up to 15%. In addition, the RMS value of the acceleration, the main representative of the riding experience, faces an improvement of 20%, going from 22,0 m/s^2 without any control method to 17,67 m/s^2 in the feedback scenario. The settling time also gives a positive result of up to 58%.

This initial study represents a great opportunity to explore the possible uses of self-sustained control systems with regenerative suspensions. The piezoelectric material was not only able to generate enough power to activate smaller on-board appliances inside the vehicle, but also create a feasible improvement in the overall comfort and riding experience. It is possible to infer that these results can possibly achieve even better numbers when considering an optimized geometry both for the suspension and the disk for this particular application.

In addition, a hybrid system, with both the piezoelectric generation and an external source can be used. Providing an active suspension performance closer to consolidated and commercial systems, however with a great reduction of power required, and reduced quantity of sensors necessary.

5. REFERENCES

Anton SR and Sodano HÁ (2007) A review of power harvesting using piezoelectric materials. *Smart Materials and Structures* 16: R1–R21.

Ching NNH, Wong HY, Li WJ, Leong PHW, Wen, Z. A laser–micromachinedmulti-modal resonating power transducer for wireless sensing systems. *Sensors Actuators* 2002; A 97/98: 685–90.

Davidson J and Mo C (2014) Recent advances in energy harvesting technologies for structural health monitoring applications. *Smart Materials Research*; Article ID 410316, 14 pages.

Glynne–Jones P, Beeby SP, James EP et al. (2001) The modelling of a piezoelectric vibration powered generator for microsystems. *Transducers 01/Eurosensors XV*, June.

Goldner RB, Zerigian P, Hull JR (2001) A preliminary study of energy recovery in vehicles by using regenerative magnetic shock absorbers. SAE paper #-01-2071.

Hsu P (1996) Power recovery property of electrical active suspension systems. Proceedings of the Inter Society Energy *Conversion Engineering Conference* 1996; Washington D.C. (USA): IEEE, 1899–1904.

IEEE Standard on Piezoelectricity. ANSI Std. 176-1987.

Kawamoto Y, Suda Y, Inoue H (2007) Modeling of electromagnetic damper for automobile suspension. *Journal of System Design and Dynamics*" 1(3): 524–535.,

Li Z, Zuo L, Kuang J et al (2015) Energy–harvesting shock absorber with a mechanical motion rectifier. *Smart Materials and Structures*, accepted for publication.

Martins I, Esteves J, Marques GD. Permanent magnets linear actuators applicability in automobile active suspensions. *IEEE Transactions on Vehicular Technology* 2006; 55(1): 86–94.

Nestorović T, Shabadi S, Marinković D et al. (2013) Modeling of piezoelectric smart structures by implementation of a user defined shell finite element. *Facta Universitatis Series: Mechanical Engineering* 11(1): 1–12.

Ottman GK, Hofmann HF, Lesieutre GA (2003) Optimized piezoelectric energy harvesting circuit using step-down converter in discontinuous conduction mode. *IEEE Transactions Power Electronics* 18: 696–703.

Pacejka, HB (2006) Tire and vehicle dynamics. SAE International.

Paultre P (2010) Dynamics of structures. John Wiley & Sons.

Reinhardt L, Haynes, A, Cordes, J (2013) Finite element method mesh study for efficient modeling of piezoelectric material. Tech. Rep. ARMET-TR-12017, U.S. Army Armament Research, Development and Engineering Center.

Roundy S, Wright PK, Rabaey J (2003) A study of low level vibrations as a power source for wireless sensor nodes. *Computer Communications* 26: 1131–1144.

Roundy S and Wright PK (2004) A piezoelectric vibration based generator for wireless electronics. *Smart Materials and Structures* 13: 1131–1142.

Yan S, Sun W (2017). Self-powered suspension criterion and energy regeneration implementation scheme of motor-driven active suspension. *Mechanical Systems and Signal Processing*. Volume 94, Pages 297-311.

Zhang G, Cao J, Yu F. Design of active and energy-regenerative controllers for DC-motor-based suspension. *Mechatronics*, Volume 23, Issue 3, April 2013, Pages 395

Zuo L, Zhang P (2013) Energy harvesting, ride comfort, and road handling of regenerative vehicle suspensions. *Journal of Vibration and Acoustics* 135, February.

6. RESPONSIBILITY NOTICE

The authors are the only responsible for the printed material included in this paper.