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ACTIVE DAMPER SYSTEM DESING AND CONTROL – PART A

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Abstract. *This paper presents a semi-active damper design. A flexible metallic bellows that pumps the working fluid through a controlled valve and produces a reactive damping force constitutes the damper. A piezoelectric actuator controls the valve orifice. From the bellows and the actuator specification, the model of the valve is developed by the finite element method, considering the interaction between the fluid and the structure of the damper. Numeric simulations are carried out to determine the pressure and velocity fields as function of different values of the valve closure and different values of the fluid velocity. These results are used to calculate the net forces acting on the model surfaces as function of the valve closure and the fluid velocity. An adaptive neural-fuzzy inference system is applied to the obtained numerical data to generate the non-linear fuzzy models of the valve fluid dynamics. The obtained results show that this fuzzy valve model can produce a wide range of the damping force and can be used as a simulator of the proposed semi-active vibration damper.*

Keywords*. Active damper, finite elemen, neural-fuzzy.*

1. Introduction

In several industrial applications it is necessary to absorb structural vibration energy, mainly when impulsive forces without any repetition pattern produce them. In these cases, the passive absorbers present low efficiency, because they are only projected for a specific condition of operation. Frequently, the hydraulic piston actuators are used where large damping forces are required. However, the implementation, operation and maintenance of this type of solution are very expensive. These active suspensions are used by high performance vehicles and represent a significant amount of the car price. The current passive shock absorbers used on regular vehicles are designed for a nominal behavior and don't allow damping factor adjustments in function of the pavement type or even for comfort or stability operating conditions.

New conceptions of vibration absorbers were proposed, investigated and tested in the last years. Some designs of active shock absorbers and dampers are presented here. Kipling (1999), Hagopian et al. (1999), Giliomee and Els (1998) developed shock absorbers systems based on hydraulic piston whose variable orifice is controlled by an electrodynamic valve. Oh and Onoda (2002) used metallic flexible bellows and electro and magneto rheologic fluids. Parker [1988] and Feigel [1996] investigated piezoelectric actuators applied to control valves.

The main proposal of this work is the development of semi-active damper that uses a metallic flexible bellows, replacing the hydraulic piston used in active suspension of vehicles. This choice is attractive because metallic bellows naturally prevents fluid leakage and presents low Coulomb friction and it is cheaper than hydraulic cylinder assembly. The bellows geometrical configuration can be designed to provide the required axial displacement and stiffness ranges, under a nominal internal pressure, to improve produced fluid flow. The damping force is generated when the fluid flows through an orifice that can be adjusted by a servo-valve. Piezoelectric actuators, allowing a larger dynamic range for the damping control position the valve, when compared to an inductive electrodynamic actuator.

This paper presents in session 2 the physical model characteristics of the proposed damper. In session 3 the finite element model of the fluid flow inside the valve is developed and the numerical simulations of the velocity and pressure fields are carried out, as functions of the applied axial velocity and of the size of the valve orifice. The net axial damping forces acting on the bellows and on the valve are calculated. A neural fuzzy inference model is developed to represent the damping forces numerical. Alternatively, the same data set is fitted to a deterministic model for the forces, using a nonlinear optimization procedure. The efficiency of these two models is compared for the full domain of input velocities at the bellows and the available range of the valve orifice size.

2. Characteristics of the Semi-active Damper

The proposed semi-active damper consists of two main parts: a piezoelectric actuator system controls the valve; and a hydraulic-mechanical system generates the damping force. The hydraulic circuit operates above the atmospheric pressure and contains a volumetric accumulator, a variable orifice, and a metallic cylindrical bellows.

The bellows is very flexible in the axial direction and its extremities are connected at two points of the vibrating structure that experience relative motion. The vibrating motion, applied in the bellows axial direction, pumps the internal fluid through the valve orifice. The displaced fluid volume travels between the bellows and the accumulator.

The flow restriction caused by the valve yield pressure gradients in hydraulic circuit, the depending of the orifice size. This internal pressure applied on the bellows effective area transversal generates the damping force that acts on the structure. The signal of the fluid velocity at the orifice changes opposed to de relative velocity applied to the bellows by the vibrating structure.

The system becomes active because the valve closure can be adjusted by a positioning control system. A linear displacement piezoelectric actuator is used to improve the frequency response and the dynamic range of the valve.

The main objective of the project is to incorporate variable damping to the mechanical system, associated to each specific position of the valve; that is, a new shock absorber can be set at each sampling time of the state variables. This rate depends on the time response of the actuator, on the external excitations and on the desired dynamic performance of the vibratory system.

In the diagram shown in Figure 01, it is assumed that the flexible metallic bellow is attached to a dynamic vibratory system and that all state variables are measurable by appropriate sensors. Then, considering one reference damping coefficient and the associated system state, a digital controller will command the piezoelectric actuator that will position the flow regulator valve. Due to a control action, the resulted orifice size will set the fluid velocity, producing the damping force. The bellows expansion and contraction movements produce pressure variations inside the hydraulic circuit. The product of the internal pressure in the bellows by its cross section area results in a value of the damping coefficient that depends of the control action applied to the valve. To avoid structural collapse of the bellows, the closure of the valve must be limited so that the internal pressure is always inferior to the maximum allowable pressure for the bellows.

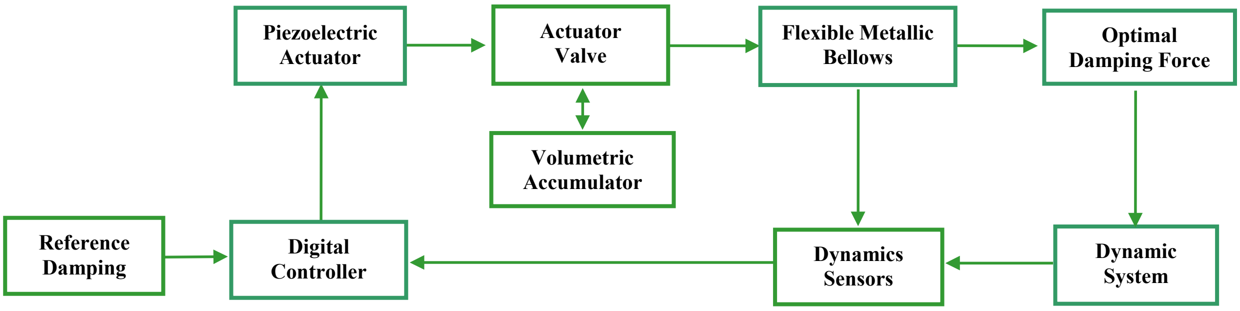

Figure 01: Blocks Diagram of the proposed active damper components

The volumetric accumulator can be a pressure vase endowed with an internal pneumatic lung. Adjusting the pressure in the lung can set the nominal static pressure in the hydraulic circuit. As an alternative solution, a second bellows can be used as the volumetric accumulator.

The efficiency of this device is strongly dependent on the fluid viscosity and on the absence of mixed bubbles of gas dissolved in the oil. To solve this problem a parallel circuit with a vacuum bomb can provide a preliminary remove the gas bubbles that are present in the working fluid. The choice of suitable value of the static nominal pressure in the hydraulic circuit can prevent the dissolved gas bubble generation in the low-pressure region of the circuit.

A commercial metallic bellows was specified. It supports internal pressures up to 3.3 GPa, has an effective cross section area of 383 mm^2, can generate axial loads up to 1300 N with a margin of safety of 25 %. The stroke of the bellows is in range of –8 mm up to 6 mm. In this project the valve was projected for damping forces varying from - 800 N to 800 N.

The piezoelectric actuator has the displacements range equal to 0.5 mm. It supports forces up to 570 N, as a function of the applied voltage in the frequency band from zero to 480 Hz.

3. The valve fluid dynamic model.

The hydraulic circuit model is developed by the finite element technique. As shown in Figure 2, the vibrations applied at the left end of the bellows imposes a known velocity to the fluid that is in contact with its the internal end surface. At the right end of the hydraulic circuit a constant pressure, equal to the nominal static pressure is imposed. The volumetric accumulator is not shown in this figure.

The valve orifice has conic shape. The orifice size can be adjusted by moving the conical core in the axial direction. In this region the fluid will reach higher velocities. When the bellows contracts the pressure will increase at the region located at the left side of the core and will be reduced at its right side. If the bellows expands, the pressure distribution will be reversed.

This model is used to determine the velocity and pressure fields developed in the fluid. By integrating the pressure on the core surface (excluding the actuator rod) the net force acting on the valve core is determined. Applying the same procedure on the left end surface of the bellows, the effective damping force acting on the vibrating structure can be calculated. The computational model of the hydraulic circuit was developed using the module FLOTRAN CFD, Computational Fluid Dynamics, of the computational program ANSYS 6.2, considering the K - ε model of turbulence, developed by Launder and Spalding (1974). According to the geometric characteristics of the problem, an axi-symmetric finite element model can be assembled. Figure 2 shows the configuration of the system components and the mesh geometry constructed with the FLUID141-2D quadrilateral finite elements The mesh is refined at the central region around the valve core, where higher velocity gradients are expected to occur. The mesh was composed by 2450 elements, since this configuration provides the numerical convergence for the velocity and pressure fields, with reasonable number of iterations

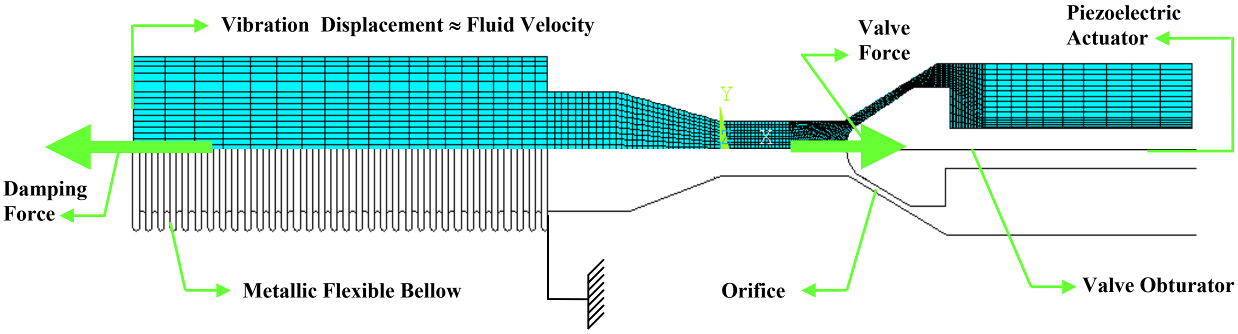

Figure 2. Finite element axi-symmetric model of the hydraulic circuit.

The working fluid is a newtonian lubricant oil with density equal to 8,87x102 kg/m3 and viscosity of 0.188 kg/ms. For all numeric simulations the viscosity is considered to be temperature independent and the density is kept constant.

The axial position of the conical core relative to the valve body is determined by the GAP variable measured at the X-axis of the inertial reference frame. As shown in Figure 3, GAP is the distance between the vertices of the triangles Δabc and ΔABC. The spherical tip of the core is generated by the circumference inscribed in the triangle Δabc. The right end of the core has a cylindrical shape, where the actuator rod is fixed.

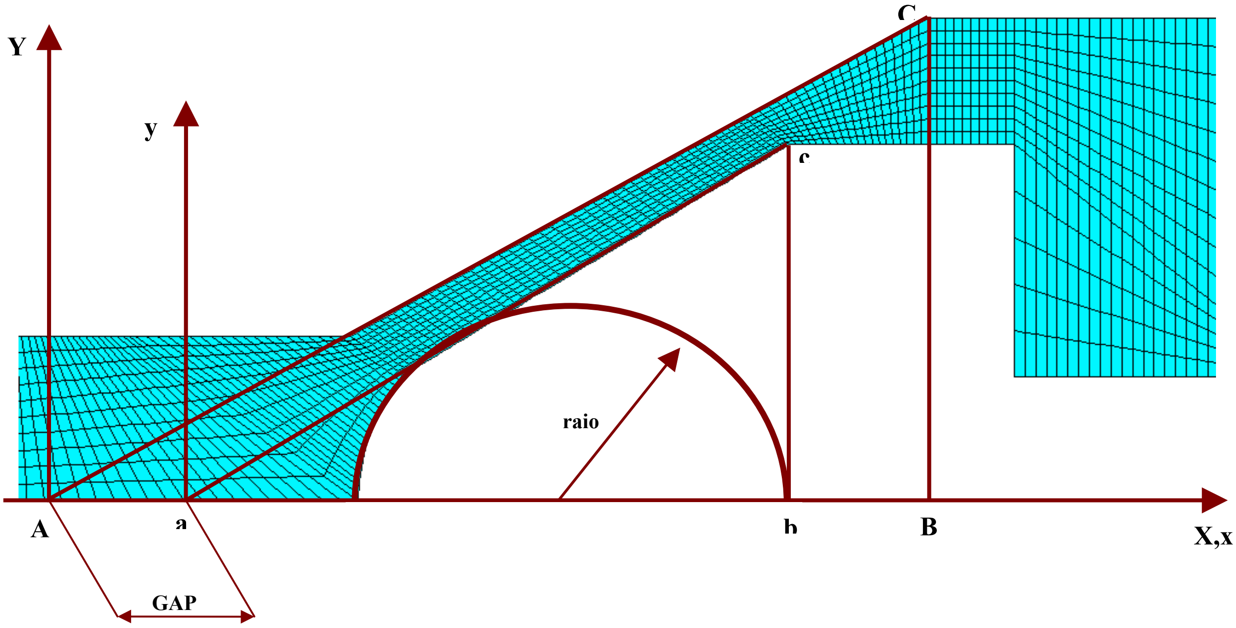

Figure 3. Valve geometry and mesh refinement

If the reference frames XY and xy are coincident GAP results equal to zero and the valve is totally closed. The refinement of the mesh near the conical wall and the mesh geometrical transition applied this region of the model is also shown.

Preliminary numerical simulations shown that the obtained Reynolds Number were low, so that the fluid flow can be considered laminar.

For steady state flow ten thousand simulations were accomplished varying combinations of the two following conditions:

- Valve closure: 50 values are set to GAP in the range of 0.8 to 1.3 mm, respectively, the minimum and the maximum size of the valve orifice;
- Vibration relative velocity: 50 values of axial speed Vx in the range of -0.4 to 0.4 m/s, which physically corresponds to the velocity resulting from the contraction or expansion of the bellows.

Other fixed conditions were adopted and imposed for problem resolution. The static or reference pressure inside the hydraulic circuit is set equal to the atmospheric pressure. The applied boundary conditions are Vx= 0 and Vy= 0 at all internal walls, except at the left surface of the bellows. Due to the symmetry Vy = 0 at the points where Y=0.

Figure 4 show the fluid velocity and the pressure distribution at the valve region for the left face of the bellows moving at Vx=0.4 m/s. The valve core is positioned with GAP=1.3 mm.

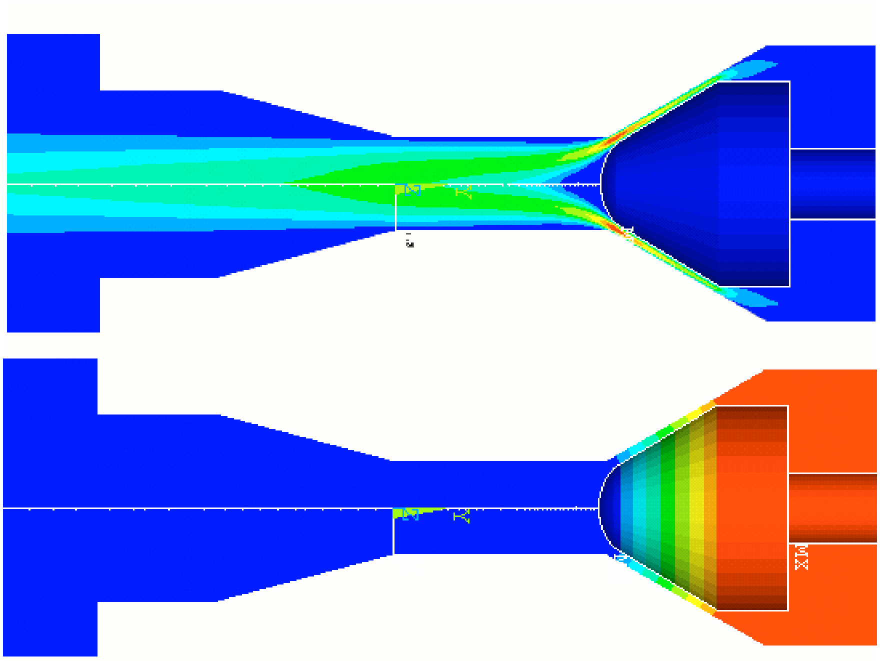

Figure 4. Velocity field and pressure distribution, for GAP=1.3 mm and Vx= 0.4 m/s.

For the input velocity Vx=0.4 m/s, the maximum velocity amplitude resulted 13.86 m/s, occurring in the region locate between the core and the valve body. At all other regions of the model the fluid velocity presents much lower values. This indicates that size of the orifice, defined by the variable GAP, is dominant to determine the damping propriety of the device. The maximum pressure resulted equal to 0.34 MPa and occurs at spherical face of the core. The pressure gradient increases in the orifice region and beyond. The minimum negative value is –0.01Mpa, meaning that it is lower than the constant static pressure imposed in the hydraulic circuit. The pressure inside the bellows is positive and practically constant.

Figure 5 show the fluid velocity and the pressure distribution at the central region of the model, for input velocity, Vx=-0.4 m/s, and the same GAP conditions of the previous simulation.

For the case of a negative input velocity the fluid flow is reversed, and the maximum velocity magnitude resulted 14.12 m/s, occurring in the same region. The maximum positive pressure is 50 Pa, occuring at the right side of the valve core. At the bellows the pressure has negative constant value equal to 0.37 Mpa. This indicates the inverse behaviour observed for positive input velocity.

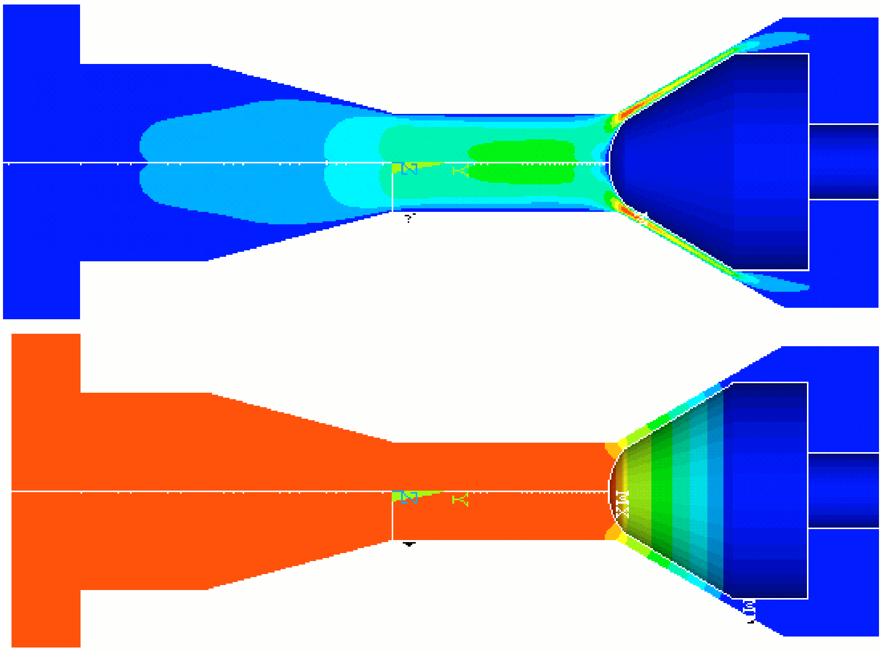

Figure 5. Velocity field and pressure distribution for GAP=1.3 mm and Vx=-0.4 m/s.

The minimum value of the static pressure necessary to prevent the bubble formation in the fluid flow, can be determined using this simulation, considering that its value must be at least equal to the pressure drop across the valve.

Integrating the pressure over the core surface results the net force at the valve core. This force has to be supported by the piezoelectric actuator. The damping force applied to the vibrating structure is calculated by integrating the pressure over the area of the bellows left end.

Figures 6 and 7 show the nonlinear behavior of these forces, as functions of the input velocity, Vx, and of the valve closure, GAP.

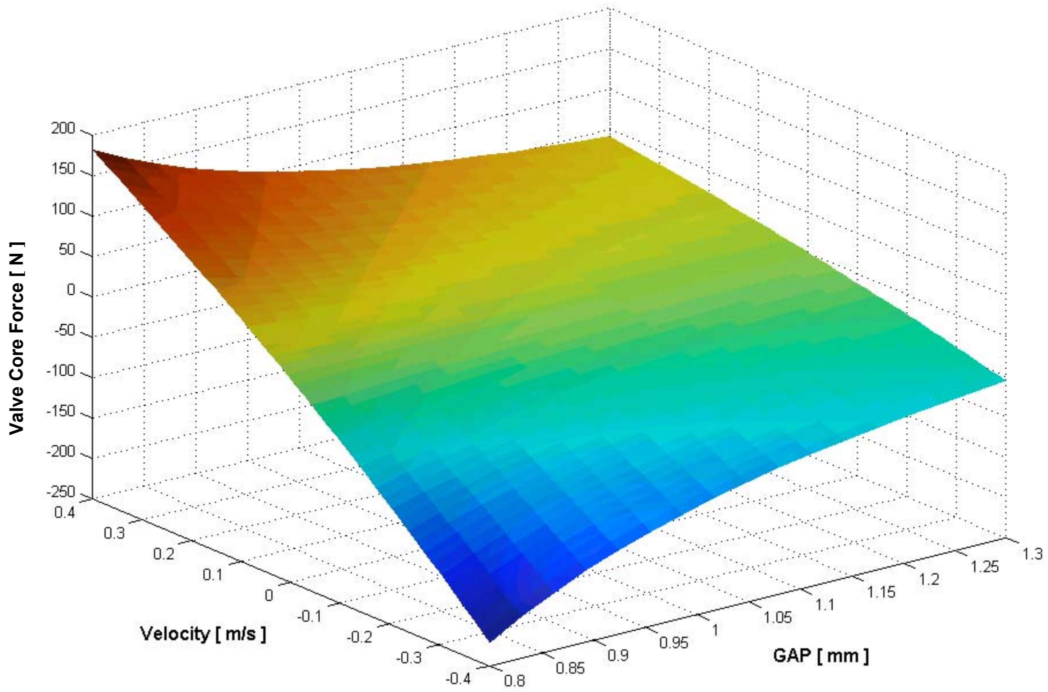

Figure 6. Finite elment simulations of the forces at the valve core.

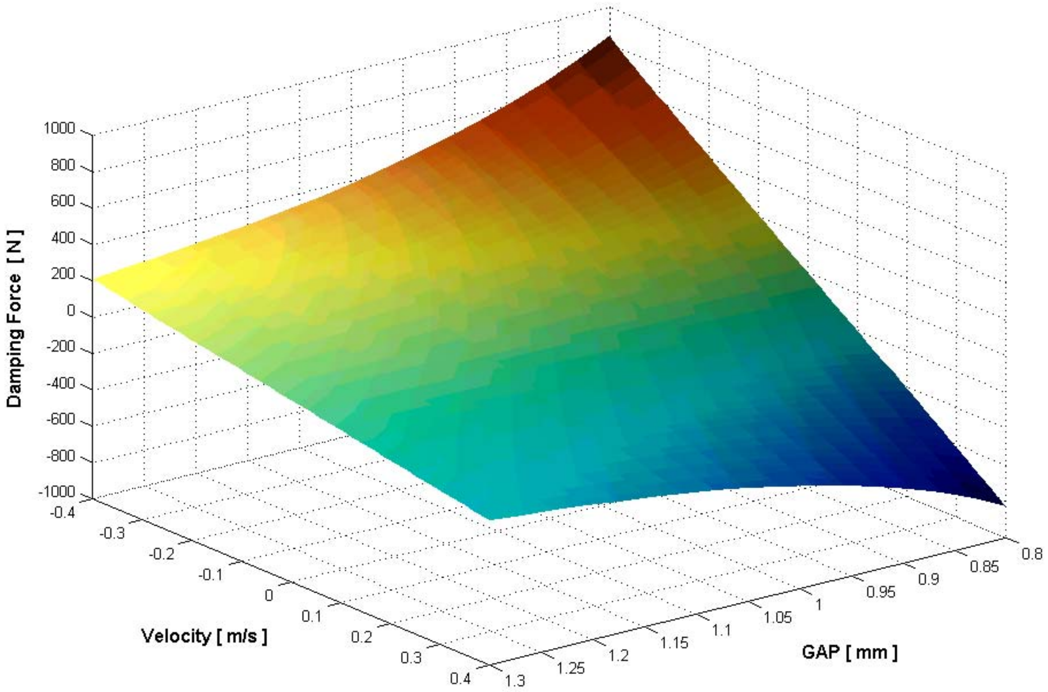

Figure 7. Finite element simulations of the damping force

It can be observed that the forces present behavior approximately linear with the velocity and exponential with GAP. This characteristic is very interesting since a small change in the valve position implicates an expressive variation of the damping force. In other words, that exponential effect reflects the high sensibility of the valve. This is necessary since the piezoelectric actuator, despite its high force capacity, has limited displacement range; witch is equal to 0.5 mm, for the selected one. On the GAP scale the value of 0.8 mm corresponds to the condition of smaller orifice, generating larger damping forces, and the value of 1.3 mm indicates the position of larger valve opening. For practical operation of the device, an offset can be superposed to the available 0.5 mm range of the piezoelectric actuator. This way, the damping force range can be set to accomplish the actual application. Two physical limits must be taken into account: the maximum pressure inside the bellows must be lower than fatigue limit of the material; and the maximum force at the valve core must be lower the blocked force capacity of the piezoelectric actuator.

4. Reduced models for the damping and valve core forces

In order to apply the controlled damping device to a mechanical system a reduced model for the damping force and for the force at the actuator must be developed. The direct use of the finite element model isn't practical since the computational time involved in the numerical simulation is to high for control applications.

A first data reduction method is to fit an analytical model for the force functions F= $f\,(Vx,\ GAP)$. Considering the behavior observed in Figures 6 and 7, a reasonable model that uses an exponential behavior with the GAP and a linear one with Vx, is defined in Equation 1.

$$f\left(Vx, GAP\right) = a\;Vx\;e^{bGAP} \tag{1}$$

The values of the parameters *a* and *b* are determined by a multidimensional unconstrained nonlinear minimization procedure that uses Nelder-Mead simplex method. For all GAP values, this approach fitting errors are less than 0.1% for |Vx| >0.002 m/s, but about 8 % for small values of the input velocity. The model for force at the valve core has a_c=-226.6 and b_c= -3003, and the damping force parameters are a_d=951.4 and b_d=2998. The analytical inverse model is easily obtained, and the GAP or Vx can be estimated from the force value. This ability can be helpfull on the operation of the controlled system. Figures 8 and 9 present the forces estimated by the analytical models. The GAP scale vary from – 0.25 mm to 0.25 mm correspond respectively to 0.008 mm and 0.0013 mm valve positions.

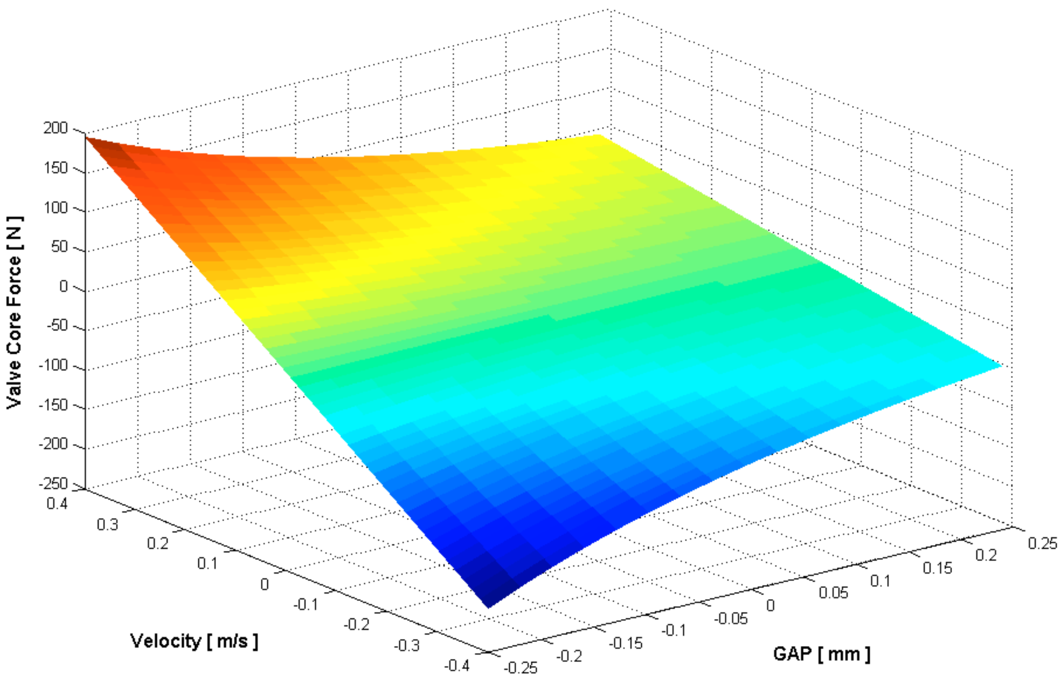

Figure 8. Analitical model for the Force at the valve core..

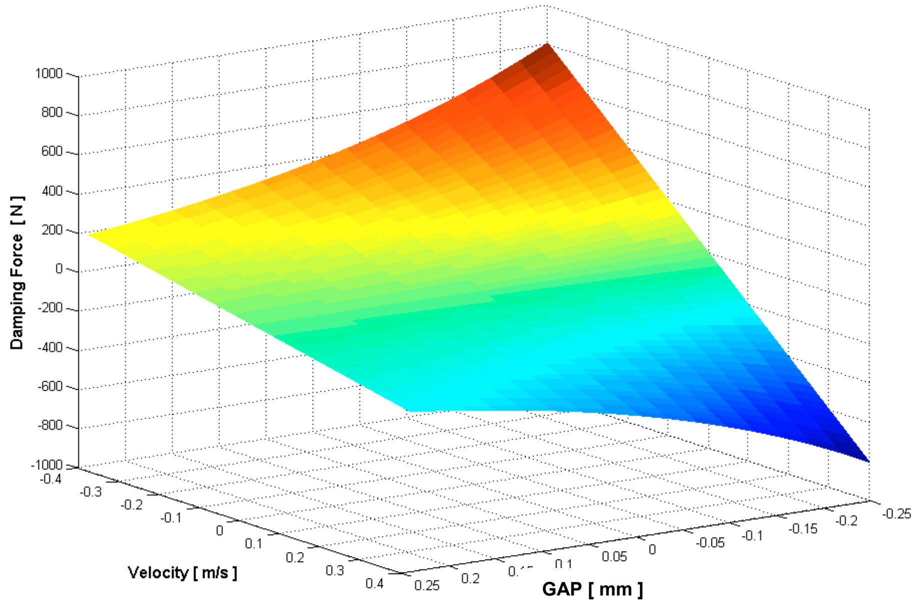

Figure 9. Analitical model for Damping Forces.

A second data reduction method is achieved by using a neural-fuzzy mapping of the force surfaces generated by the simulations of the finite element model. This approach leads to a non deterministic model that is determined by the method ANFIS – Adaptive neural fuzzy inference system, proposed by Jang et al. (1997). The unrestrained role Sugeno fuzzy model is a universal estimator to any nonlinear function; witch can be proved mathematically by the Stone-Weierstrass theorem.

A two inputs *(x, y)* and one output *(z)* Sugeno first order fuzzy system will be used to understand the ANFIS architecture, shown in Figure 10. For sake of simplicity, lets assume the following rules for the Sugeno system:

Rule 1: if x is A then $f_1 = p_1\, x + q_1\, y + r_1$

Rule 2: if y is A then $f_1 = p_2\, x + q_2\, y + r_2$

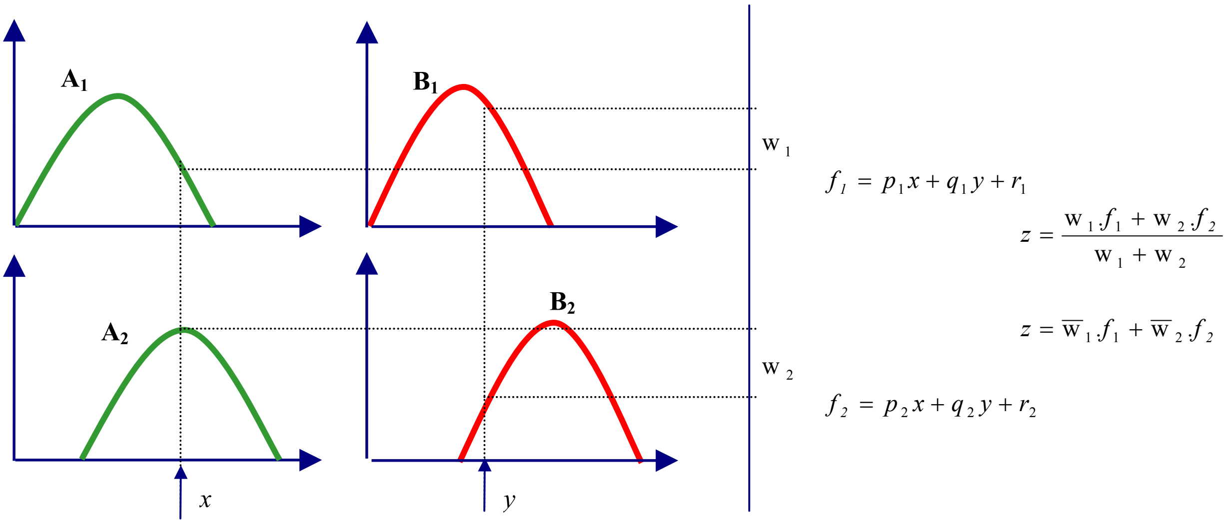

Figure 10. First order Sugeno fuzzy system with two inputs and one output.

The Sugeno fuzzy system can be represented by the ANFYS equivalent system shown in Figure 11. Each one of the layer (j) performs similar operations for each node (i). The algorithm to calculate the output, S_{ji}, for each layer is as follows:

Layer 1: Each node on this layer is adaptive whose output is the value of the pertinence functions of all fuzzy sets.

$$S_{1,i} = \mu_{Ai}(x) \quad , \; for \;\; i = 1, 2, or$$
$$S_{1,i} = \mu_{Bi-2}(x), \;\; for \;\; i = 3, 4.$$

The following bell shape function can be parameterized to represent the fuzzy set, using adaptive values *a*, *b* and *c* (premises):

$$\mu_A(x) = \frac{1}{1 + \left|\frac{x - c_i}{a_i}\right|^{2b_i}}$$

Layer 2: The output of each node is the product of the input signals at this layer, resulting the weighs w_i:

$$S_{2,i} = w_i = \mu_{Ai}(x).\mu_{Bi}(y) \;\; , \; for \;\; i = 1, 2.$$

Layer 3: The weights, obtained at the previous layer, are than normalized:

$$S_{3,i} = \overline{w}_i = \frac{w_i}{w_1 + w_2} \;\; , \; for \;\; i = 1, 2.$$

Layer 4: The nodes in this layer are adaptive, using the consequent parameters *p*, *q* and *r*. The output is calculated by the following function:

$$S_{4,i} = \overline{w}_i.f_i = \overline{w}_i.(p_i x + q_i y + r_i) \; , \; for \;\; i = 1, 2.$$

Layer 5: There is only one node in this layer. Its output is the sum of all input signals at this layer:

$$S_{1,j} = z = \sum_i \overline{w}_i.f_i \;\; , \; for \;\; i = 1, 2.$$

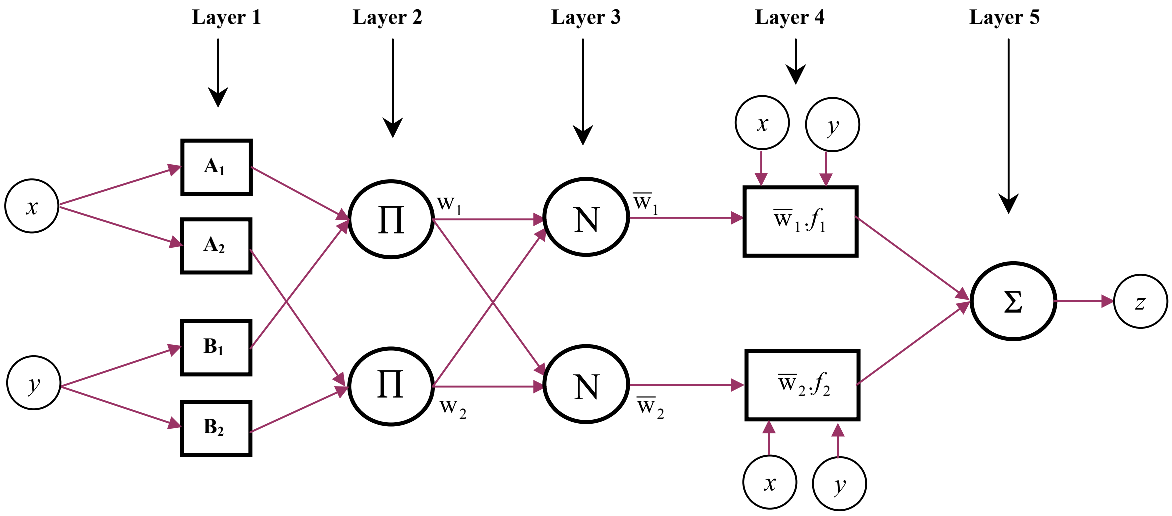

Figure 11. ANFYS equivalent architecture.

To solve this problem it is necessary to determine the nonlinear premises parameters (*a, b, c*) and the linear consequence parameters (*p, q, r*). Initially, the premise parameters are fixed, and the output *z* is calculated by the following equation:

$$z = \frac{w_1}{w_1 + w_2} f_1 + \frac{w_1}{w_1 + w_2} f_2 = \overline{w}_1.f_1 + \overline{w}_2.f_2 = \overline{w}_1.(p_1 x + q_1 y + r_1) + \overline{w}_2.(p_2 x + q_2 y + r_2)$$

The hybrid learning process consists of a first forward step where the premise parameters are set fixed and the network is calculated up to the fourth layer. The consequence parameters are estimated by a mean square procedure, and the output error can be evaluated. A second step is performed backpropagating the error, with the fixed values for the consequence parameters. The premise parameters are than updated, using a step descent gradient technique. This procedure is similar to that used to training a backpropagation neural network (Teixeira, 2001).

This method was applied to the forces data sets obtained by the numerical simulations of the finite element model, Two neural-fuzzy models were obtained, one for the damping force and the other for the force applied to the valve core. In each case, the inputs are Vx and GAP and the output is the force value. Five gaussian pertinence functions with 25 rules were used for the inputs. This configuration represents an optimization problem with 75 linear and 20 nonlinear parameters. Half of the data set was used in the training process and the inputs were normalized between –1 and 1. The remaining data was used to validate the models. The mean square error of all 2500 data points are calculated, resulting 0.06 N for the force at the valve core, and 0.57 N for the damping force. Figures 12 and 13 present the forces estimated by the neural-fuzzy models.

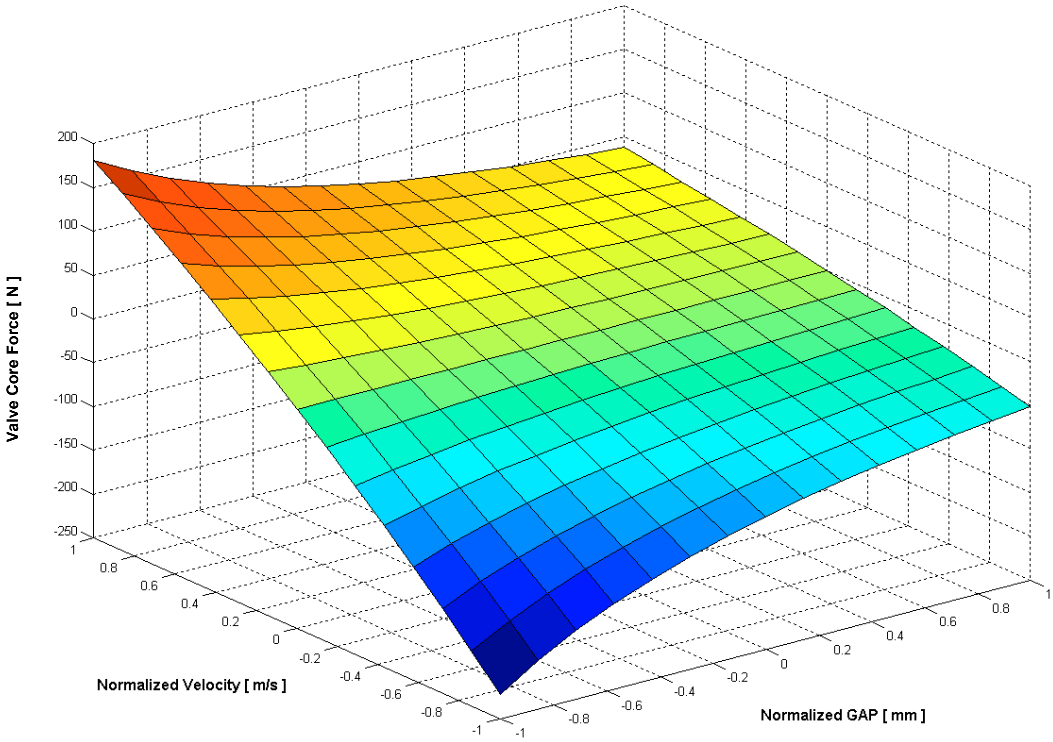

Figure 12. Neural-fuzzy model for the Force at the valve core.

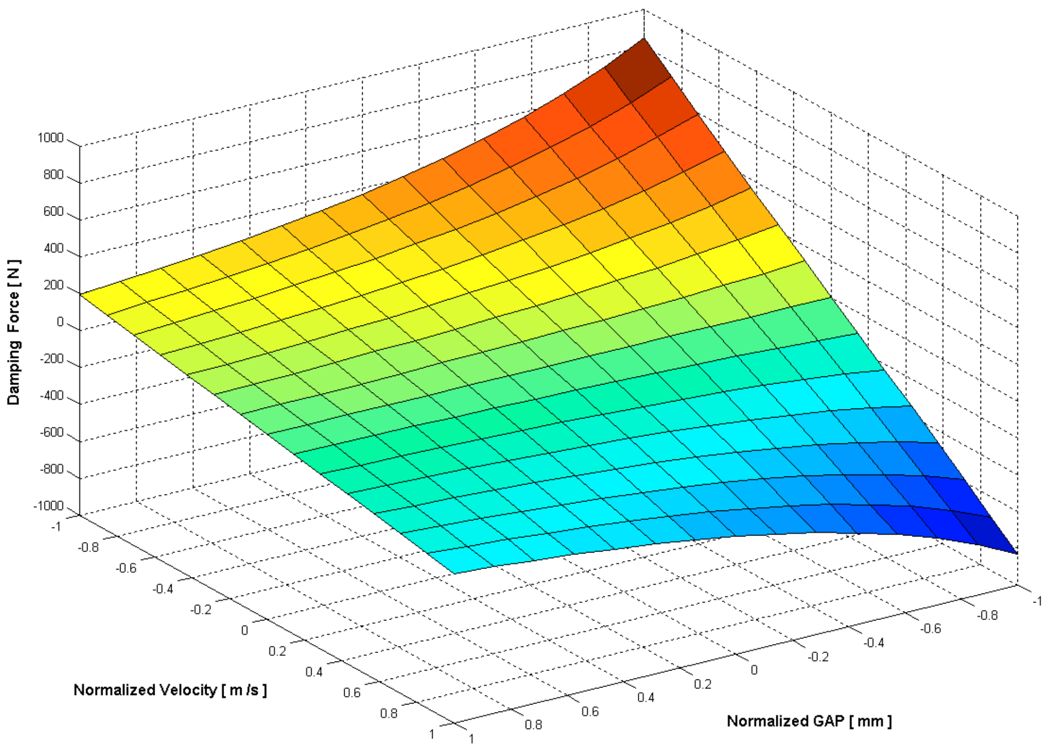

Figure 13. Neural-fuzzy model for the damping Force.

The non deterministic model errors are lower than those obtained by the analytical model of Equation 1, mainly for values of $|Vx|< 0.002$ m/s. Based on this technique, the forces direct model was detemined. In other words, given a GAP and Vx, it is possible to estimate both damping and valve core forces. However, inverse models are difficult to be determinied when the training data has ambiguity the training becomes impossible. This is the case of GAP estimation from the force and input velocity values. Then, the analytical model can be used to solve the inverse model.

5. Conclusion

The proposed design of this damping device has promising possibilities as a vibration absorber. It has simple and cheap construction and operation compared to the conventional devices that use hydraulic cylinders. Even for passive dampers, this design can produce high damping forces, without any expend of external energy. The physical limits are related to the mechanical resistance of the bellows material.

The simulation of the finite element model indicate that large damping force variation can obtained with small displacements of the valve core, as required by piezoelectric actuators. For the analyzed vibration relative velocity range the fluid flow could be considered as laminar, since the Reynolds Number are low at all model regions. The damping force global behavior indicates a linear dependence on the input velocity and exponential with the valve position. This fact is significant to the design of an active device, since small changes in the valve aperture provides large variations on the produced force.

The reduced analytical model for the damping force and for the force at the valve core can be applied to the dynamic model of the device coupled to a mechanical system, since the errors on the estimated forces are significant only for very low values of Vx. This approach produces a dircct and inverse model with high computational performance.

The neural-fuzzy model produces lower errors on the estimates of the forces and the increasing of computational effort required during the operation of a controlled system is marginal, but it can not solve the inverse problem that requires teh estimation of the GAP from the force and input velocity.

A mixed reduced model can be defined. The forces are estimated by the neural-fuzzy model and the GAP estimation is done by the analytical model. This way the controlled system can operate with high computational efficiency with minimum error on forces or GAP estimates.

The second part of this paper presents the simulation of a quarter vehicle model that includes this active damper on its suspension. An optimum quadratic regulator associated with a PID controller for the piezoelectric actuator is designed to control system vibrations.

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