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# Application of smart materials in vibration control systems

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**Analysis and modelling**

## ABSTRACT

**Purpose:** The goal of this paper is to present application and method of numerical modelling smart materials in vibration control systems. Two methods of vibration control was presented in this work. First one is based on shape memory alloy absorber. Second method use magnetorheological bearing which was placed in revolute join of manipulator mechanism.

**Design/methodology/approach:** The numerical models of presented mechanical systems were created in APDL language, which is internal ANSYS language. Dynamic characteristics of shape memory alloy absorber were determined by using algorithm which automatically changes absorber's length. The manipulator mechanism with magnetorheological bearing was modelled by using multibody dynamics method connected with finite element method in ANSYS environment.

**Findings:** Through this study it was determined shape memory alloy absorber's length which eliminated specified resonance due to natural frequencies of mechanical system. The dynamic characteristics of mechanical system with magnetorheological bearing were also obtained.

**Research limitations/implications:** The main disadvantage of presented methods is the necessity to calculate parameters for each iteration step. In the case of shape memory alloy absorber this process significantly extends the calculation time.

**Practical implications:** Presented methods allowed to determine dynamic characteristics of vibration control systems using smart materials and enabled implementation of the method to commercial finite element method environment.

**Originality/value:** This work contains new aspects, which are: determination of shape memory alloy absorber's length, practical implementation of magnetorheological fluids in vibration control systems.

**Keywords:** Numerical techniques; Shape memory alloys; Magnetorheological fluids

## 1. Introduction

The vibration control system applied in machines construction can be divided into following groups [1],[2],[4],[13] – passive, active and semi-active. Passive vibration control systems can be depicted as systems characterized by dissipation of vibration energy without increasing total energy in primary system. These solutions are often designed as additional mass connected with a spring element to primary system. The optimal designing may be found in a number of sources, including [10],[12].

Active vibration control systems are in contradistinction to passive one. These systems act on structure by means of actuators generating forces depending on dynamic response from the structure. The energy applied in that way causes vibration reduction for the whole primary system. Semi-active vibration control system connects passive tuneable devices and active systems characterized by control abilities. These systems use shape memory alloys and magnetorheological fluids more and more frequently. That method allows taking advantage of positive features of passive and active systems characterized by very low energy consumption.

## 2. Shape memory alloys

Shape memory alloys are a group of material, which are characterized by the ability to undergo a large deformation about 8% to 10% without permanent deformation effect [6],[10],[11]. If recovering to initial shape is self-contained that effect is called superelasticity or pseudoelasticity. If the above mentioned recovery is a result of heating to the specified temperature, the shape memory effect is observed then. Thermodynamical properties of SMA are characterized by the four temperatures $M_s$, $M_f$, $A_s$, $A_f$. The first two temperatures are responsible for the transition from austenite to martensite phase during cooling. The other two temperatures are responsible for the inverse process during heating. The subscripts indicate respectively the start and the finish point of the above described transformation process. These alloys are classified as smart materials and are frequently used to construct semi-active absorbers [7],[14],[15].

### 2.1. Mathematic model

Model with assumed phase transformation kinetics presented in this section considers strain (ε), temperature (T), and internal variable (ξ), used to represent the martensitic volumetric fraction involved. The constitutive relation between stress and state variables, for SMA modeling, is considered in the rate form as follows [8]:

$$\sigma - \sigma_0 = E(\xi)(\varepsilon - \varepsilon_0) + \Theta(T - T_0) + \Omega(\xi)(\xi - \xi_0) \tag{1}$$

Where E represents the elastic tensor, Ω corresponds to the phase transformation tensor, and Θ is associated with the thermoelastic tensor. Due to martensitic transformation nondiffusive nature, the martensitic volumetric fraction can be expressed as function of current values of stress and temperature.

Brinson constitutive model [8] offers an approach to the phase transformation kinetics, in which, besides considering cosine functions, the internal variable ξ is split into two distinct martensitic fractions - one temperature induced $\xi_T$, and the other stress induced $\xi_S$, in such a way that:

$$\xi = \xi_S + \xi_T \tag{2}$$

In this model also considers different elastic moduli for austenite $E_A$ and martensite $E_M$, so that the elastic modulus is given by a linear combination such that [8]:

$$E(\xi) = E_A + \xi(E_M - E_A) \tag{3}$$

The martensitic transformation evolution is expressed by:

For $T > M_S$ and $\sigma_S^{CR} + C_M(T - M_s) < \sigma < \sigma_F^{CR} + C_M(T - M_S)$

$$\xi_S = \frac{1-\xi_{S0}}{2}\cos\left(\frac{\pi}{\sigma_S^{CR} - \sigma_F^{CR}}(\sigma - \sigma_F^{CR} - C_M(T - M_S))\right) + \frac{1+\xi_{S0}}{2} \tag{4}$$

$$\xi_T = \xi_{T0} - \frac{\xi_{T0}}{1-\xi_{S0}}(\xi_S - \xi_{S0}) \tag{5}$$

For $T < M_S$ and $\sigma_S^{CR} < \sigma < \sigma_F^{CR}$ :

$$\xi_S = \frac{1-\xi_{S0}}{2}\cos\left(\frac{\pi}{\sigma_S^{CR} - \sigma_F^{CR}}(\sigma - \sigma_F^{CR})\right) + \frac{1+\xi_{S0}}{2} \tag{6}$$

$$\xi_T = \xi_{T0} - \frac{\xi_{T0}}{1-\xi_{S0}}(\xi_S - \xi_{S0}) + \Delta_{T\xi} \tag{7}$$

Where for $M_F<T<M_S$ and $T<T_0$

$$\Delta_{T\xi} = \frac{1-\xi_{T0}}{2}[\cos(a_M(T - M_F)) + 1] \tag{8}$$

The reverse transformation is defined:

For $C_A(T - A_S) < \sigma < C_A(T - A_S)$ and $T > A_S$

$$\xi_S = \frac{\xi_{S0}}{2}\left\{\cos\left[a_A\left(T - A_S - \frac{\sigma}{C_A}\right)\right] + 1\right\} \tag{9}$$

$$\xi_S = \frac{\xi_{T0}}{2}\left\{\cos\left[a_A\left(T - A_S - \frac{\sigma}{C_A}\right)\right] + 1\right\} \tag{10}$$

The numerical implementation of Brinson model for different SMA temperatures is presented on figure 1. The material properties for the shape memory alloy in the following analysis are listed in table 1.

Table 1.
Material properties for the Nitinol alloy

| Modules | Transformation Temperatures | Transformation Constants | Maximum Residual Strain |
|---|---|---|---|
| $E_a$=67e9 [Pa] | $M_f$=9 [°C] | $C_M$=8e6 [Pa/°C] | $\varepsilon_L$=0,067 |
| $E_m$=26,3e9 [Pa] | $M_s$=18,4 [°C] | $C_A$=13,8e6 [Pa/°C] | |
| Θ =0,55 [Pa/°C] | $A_s$=34,5 [°C] | $M_f$=100e6 [Pa] | |
| | $A_f$=49 [°C] | $M_f$=170e6 [Pa] | |

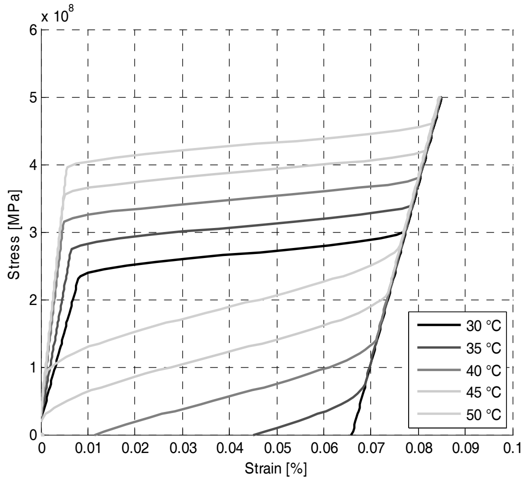


Fig. 1. Numerical implementation of Brinson model for different SMA temperatures

## 2.2. SMA absorbes

The different prototype SMA vibration control systems may be found in [13]. In order to modify dynamic characteristics of mechanical systems, the change of mass or the stiffness of system is indispensable. In designing process of such absorbers it is very important to choose proper physical features [16]. Two examples of absorber construction with SMA are presented on figure 2.

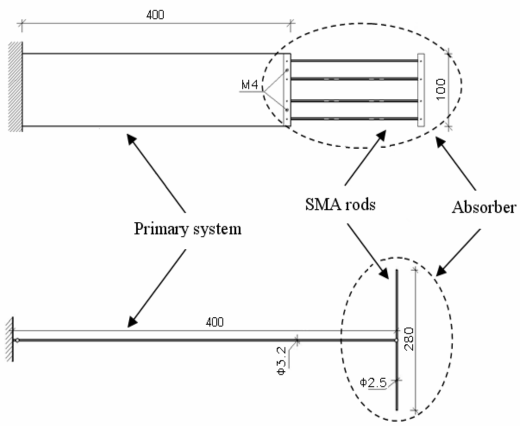


Fig. 2. Semi-active absorber with SMA rods

## 2.3. Numerical analysis

The APDL algorithm was used to analyse dynamic characteristics of the absorber by automatic change absorber's length. The data set which was obtained contains the absorber's length, frequencies and displacements of the specified mode shapes. These results were achieved with the assistance of modal assurance criterion (MAC) [5]. The comparison was indispensable owing to different absorber's lengths entailing the occurrence of different mode shape frequencies. Therefore the first set of mode shapes was compared by means of modal assurance criterion with the mode shapes of the beam without an absorber. Because this relationship is only appropriate for a few first absorber's lengths, it was decided that the updating eigenvectors should be applied. This method allows to compare calculated eigenvectors with the results which had been previously obtained. If the deviation between compared eigenvectors exceeds the limit, the calculated mode shape will be discarded. Otherwise, the calculated frequency and displacement for the specified mode shape will be saved to the file.

The model consists of the two connected parts, a steel beam as primary system and a aluminium rod with SMA actuator as an absorber. The figure 3 and 4 shows a geometrical model made in ANSYS of an analyzing mechanical system with the absorber. The dimension of the steel beam is 20x2 mm and the absorber rod is 5x1 mm. The model was fixed at the beginning of the beam. The numerical modal prestressed undamped analysis was used to determine dynamic system characteristics of the absorber's length function for two different SMA actuator states. In this process there were six main mode shapes obtained with their modal parameters.

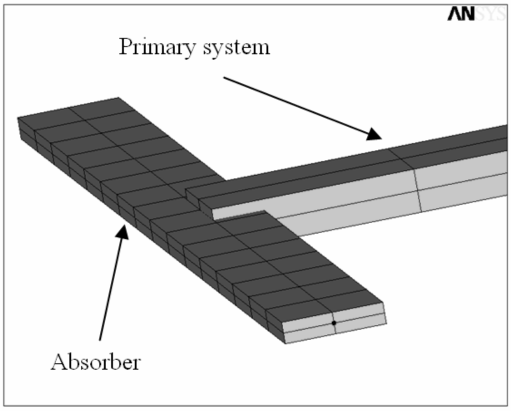


Fig. 3. The geometrical model of semi-active absorber

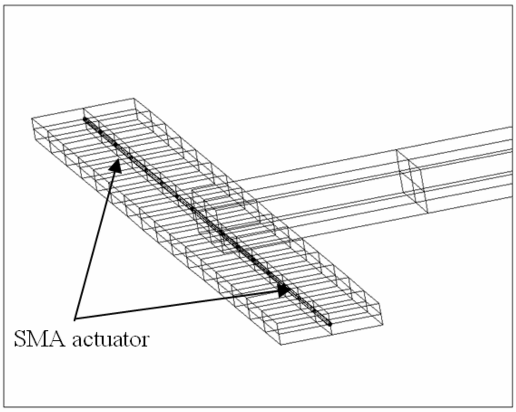


Fig. 4. Section of geometrical model of semi-active absorber

## 2.4. Results

The black lines in figure 5 correspond to the low temperature, and the grey lines show the high temperature of SMA actuator. The martensite curves are situated below austenite curves in particular ranges of absorber's length. This effect can be used to control dynamic properties of analyzing mechanical systems.

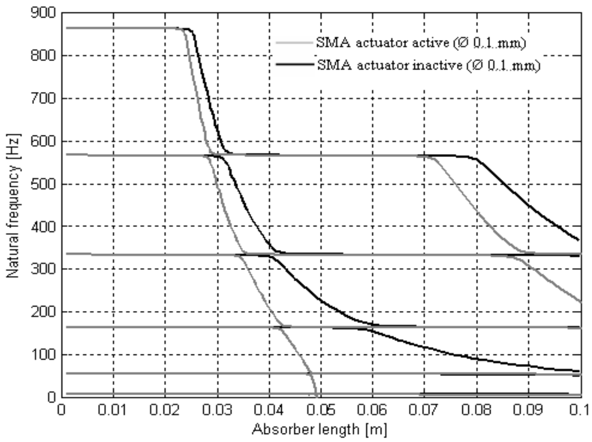


Fig 5. The eigenfrequencies in an absorber length function

In order to control vibration of primary system it can be use effect of detuning system resonance. It is possible to obtain absorber length which allows to detune two and more natural frequencies. It is necessary to notice that passive control systems can eliminate only one natural frequency. The following graph (Fig.6.) illustrate example of elimination resonance due to fourth and fifth eigenfrequencies of primary system for specified absorber length and SMA actuator dimensions. The similar operation can be done for others neighbouring eigenfrequencies by changing absorber length and SMA actuator force.

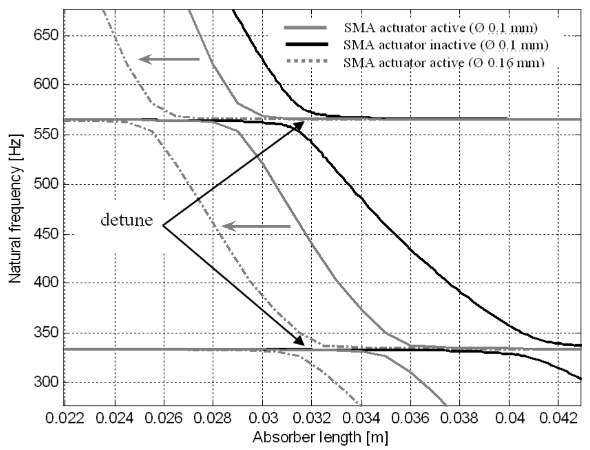


Fig 6. Detuning fourth and fifth eigenfrequencies by changing dimension of SMA actuator

## 3. Magnetorheolgical fluids

Magnetorheological materials are a class of material whose rheological properties (elasticity, plasticity, or viscosity) may be rapidly varied by applying a magnetic field. A typical magnetorheological fluid consists of 20–40 % by volume of micron sized (3–8 μm), magnetizeable, relatively pure iron particles suspended in a carrier liquid such as mineral oil, synthetic oil, water, or glycol. A variety of proprietary additives similar to those found in commercial lubricants are used to discourage gravitational settling and promote particle suspension, enhance lubricity, change initial viscosity, and inhibit wear. Under influence of magnetic field the suspended particles polarize and interact to form a structure aligned with the magnetic field that resists shear deformation or flow. This change in the material appears as a dramatic increase in apparent viscosity, or the fluid develops the characteristics of a semisolid state. Strength of the magnetic field controls the magnitude of this change. Increase of viscosity is immediately reversed upon removing the magnetic field. Besides the rheological changes in MR fluids under the influence of a magnetic field, there are often other effects such as thermal, electrical, and acoustic property changes. Gels, foams, powders, greases, and elastomers are other materials that are responsive to magnetic fields [9].

### 3.1. Mathematical models

A simple Bingham visco-plasticity model, as shown in Fig. 7, is effective at describing the essential field-dependent fluid characteristics. In this model, the total shear stress is given by [9]

$$\tau_{TOTAL} = \tau_0(H) \cdot \mathrm{sgn}(\dot{\gamma}) + \eta_P \dot{\gamma} \tag{11}$$

Apparent viscosity $\eta_a$ is defined as the total shear stress divided by the shear rate:

$$\eta_a = \frac{\tau_{TOTAL}}{\dot{\gamma}} \tag{12}$$

Where:

H - Magnetic field strength (a/m)
$\tau_{TOTAL}$ - Total shear stress of the material [Pa]
$\tau_0$ - Yield stress caused by the applied field [Pa]
$\eta_a$ - Apparent viscosity [Pa • s]
$\eta_p$ - Plastic viscosity [Pa • s]
$\dot{\gamma}$ - Shear rate [$s^{-1}$]

In the Bingham model the fluid post-yield viscosity is assumed to be a constant. Because MR fluids exhibit shearing thinning effect which is shown in figure 2 the Herschel-Bulkley visco-plasticity model can be employed to accommodate this effect. In this model, the constant post-yield plastic viscosity in the Bingham model is replaced with a power law model dependent on shear strain rate.

Therefore,

$$\tau_{TOTAL} = \left( \tau_0(H) + K\left|\dot{\gamma}\right|^{\frac{1}{m}} \right) \cdot \mathrm{sgn}(\dot{\gamma}) \qquad (13)$$

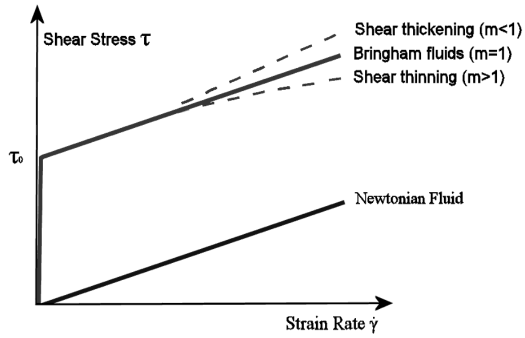


Fig. 7. Bingham visco-plasticity model [2]

Where:
K, m –fluids parameters greater than zero

Comparing equation 11 with equation 13, the equivalent plastic viscosity of the Herschel-Bulkley model is:

$$\eta_P = K\left|\dot{\gamma}\right|^{\frac{1}{m}-1} \qquad (14)$$

Equation 9 indicates that the equivalent plastic viscosity decreases as the shear strain rate increases when (shear thinning). Furthermore, this model can also be used to describe the fluid shear thickening effect when. The Herschel-Bulkley model reduces to the Bingham model when m = 1 and $\eta_P = K$ [9].

## 3.2. MRF bearing

One of the ways to control damping in kinematic pairs of mechanical system is application of element with magnetorheological fluid [3]. The prototype MRF bearing is presented on figure 8. This device was applied in manipulator mechanism which is shown on figure 10. Principle of operation is increase torque when accelerations exceed the limit.

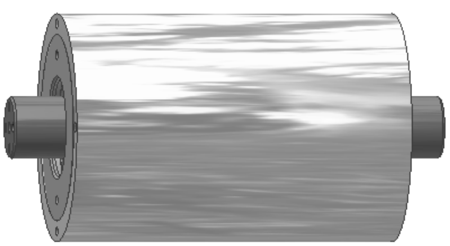

Fig. 8. The geometrical model of MRF bearing

## 3.3. Analysis method

The method which was used to analyse mechanical system with MRF bearing is describing below. The mechanical system was divided in two parts: rigid parts and elastic parts. The multibody dynamics method was used to obtain accelerations, velocities and positions of manipulator mechanism. The MRF bearing torque value was depended of acceleration and time. The calculations were done in ANSYS environment. On the basis of evaluated accelerations there were calculated forces which acted on each finite element modelling elastic part of manipulator (Fig.9). The described process was repeated for each iteration step.

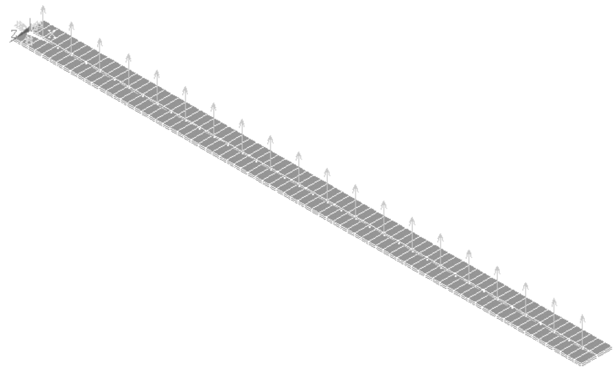

Fig. 9. The finite elements model of manipulator elastic part.

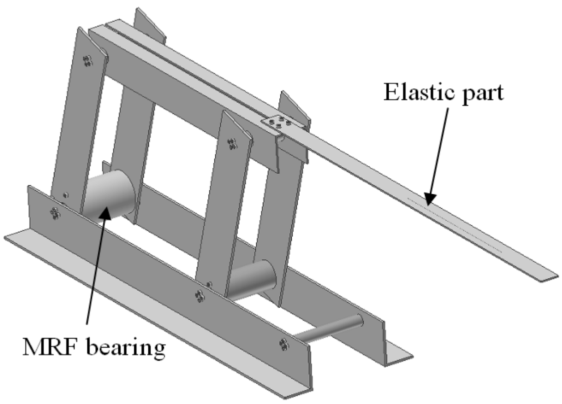


Fig. 10. The mechanical system with MRF bearing

## 3.4. Results

Simulation of dynamic characteristic of mechanical system with MRF bearing is presented on figure 11. The acceleration of elastic part tip are dependent of MRF bearing torque value. The algorithm which control MRF bearing torque value should be optimized for minimization vibrations of elastic part tip.

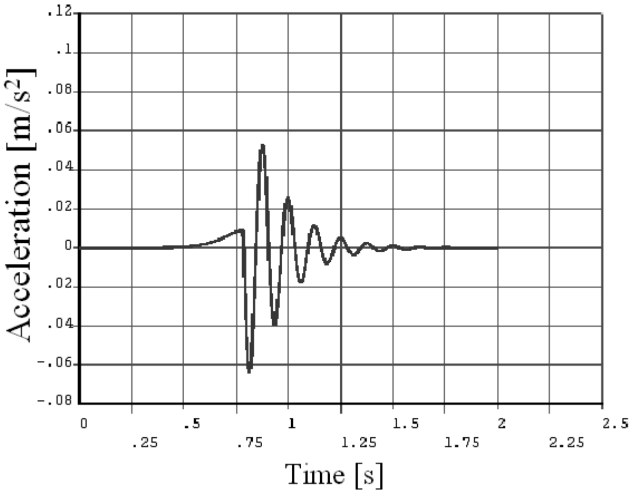


Fig. 11. The accelerations of elastic part tip

## 4. Conclusions

The shapes memory alloys and magnetorheological fluids give many possibilities of control vibration systems. In the case of SMA absorber absolute differences between frequencies are enough to change dynamic properties of primary systems. The method which was shown in this paper is proper for simple mechanical systems. The mainly disadvantage of this method is the necessity to calculate modal parameters for each absorber length. The method which was used to determine dynamic characteristics of manipulator mechanism with MRF bearing is simply in application. This approach allows to decrease time needed to calculate solution by choosing vibrating part of mechanical system. The implementation of this method is possible almost for all available commercial finite element method environments. The very important thing in the future works is taking into consideration the damping of mechanical system. There is also a necessity to verify numerical results by experimental analyses.

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