Mechanics and Mechanical Engineering
Vol. 12, No. 1 (2008) 61–68

Possibilities of Application of Smart Materials in Vibratory Machines

Mariusz Giergiel
Department of Robotics and Mechatronics AGH University of Science and Technology
PL-30059 Kraków, Poland, al. Mickiewicza 30

Received (18 October 2007)
Revised (25 November 2007)
Accepted (30 February 2008)

Along with the increasing dimensions, speed, and output of modern machines, rising problems are being encountered in their design and construction. Based on the consideration made, the conclusion can be drawn that from the point of view of mechatronical design of vibratory machines it is necessary to determine precise model of machine including driving system, especially model of inductive motor. In this paper application of piezomaterials used as distributed sensor and actuator for active vibration control is shown. The research proved that piezoelectric patches could be applied to systems in which the control of dynamic features is required.

Keywords: vibratory machines, dynamics analysis, smart materials

1. Introduction

Along with the increasing dimensions, speed, and output of modern machines, rising problems are being encountered in their design and construction. Vibrations in the course of certain technological processes, create undesirable phenomena that limit durability, cause excessive dynamic reaction with the surroundings, or can increase the level of sound emitted. For the class of machines, however, vibration constitutes the primary factor ensuring that the desired technological process is performed correctly. This class of machinery hereafter referred to as vibratory machines. In this paper some concepts of application of smart materials in vibratory machines are presented, especially in. vibro isolation systems and suspension of vibratory machines. This kind of suspension can use "intelligent" springs made from smart metal NiTi known as Niniol or even piezoelectric materials. Parameters of machine suspension may by changed by digital control system. Because of phenomena of such "intelligent" suspension it is possible to greatly reduce problems concerned with passing thru resonance during start-up and breaking of over-resonant vibratory machines.

2. Considerations of modeling of vibratory machine

2.1. *Model of the electrical part*

The propulsion of vibratory machines used to be built with single run three-phased asynchronous engines. It is indispensably to use suitable model of propulsion, when performing simulation. The asynchronous machine is the most popular electromechanical converter, used in systems with various dynamical structures. Dynamical advances that exist during starting phase, braking, voltage-hesitation used to be basic gauge during mechanical, electrical and thermal selection of the engine for designed propulsion.

Nonlinear differential systems determine mathematical model of the engine. Nonlinearly these results from structures of connections of coordinates generalized mechanical and electric. The structure of engine electromagnetic configuration classified this type of machine to holonomical systems, where the equations of chains are integrated. The dependence between electromagnetic inductive in a slit and voltage systems are linear.

It is possible to use double or single-framed models of asynchronous machine. What is more we are able to explore Kloss model or statically characteristic as well. However from exchanged models only single and double-framed models suits dynamics of the asynchronous machine.

Using of double-framed model permits to describe wide group of asynchronous machines. This model gives an extraordinary precision of calculations including statically characteristics. The single-framed model posse quite big precision for annular and single-framed machines with ring-shaped bars cage about round rods [5].

Approaching the problem of mathematical construction for the asynchronous machine, it is necessary to take the care on basics purposes, which should be executed.

The first thing is that we expect mathematical models are the relations between voltages, currents, moments and rotary speed. We are especially interested in dynamical conditions like starting phase, braking, charge and discharge phases, and changes during switching the voltages and resistance or finally short-circuit. The most important conditions to obtain useful mathematical model are verifiability of its adequacy and possibility to fix parameters in the way of measurements.

Because of difficulty of identifying parameters of the system and restricted precision, especially when interested in electromechanical worth, we should use the plainest model as it possible. Defining parameters of double-framed needs conversant with engine characteristics and its construction parameters. Usually even if we all data of the engine are known it is necessary to use optimization methods, to obtain the right parameters. Moreover, the double-framed machine typical parameters are not defined in literature, as it goes with single-framed model. Summing up, there is no possibility to get accurate parameters, even if catalogue data of engine are well known.

Plain and the possibility of using catalogue data of engine are the reasons why the single-framed model used to be more popular. The stator of the machine has p poles pairs of clutches and the rotor is made as the single-framed. This model is described by equations [5]:

$$\dot{\Phi}_s = -a_s\left(\Phi_s - \frac{1}{k_w}\Phi_w\right) + U \tag{1}$$

$$\dot{\Phi}_w = -a_w\left(\Phi_w\frac{k_s}{k_w} - \frac{1}{k_w}\Phi_s\right) - j\omega_e\Phi_w \tag{2}$$

$$M_{el} = -CIm\left(\Phi_s\Phi_w\right) \tag{3}$$

$$C = \frac{2M_u\omega_o^2 p}{I_w} \tag{4}$$

The model of the asynchronous machine is resolved itself into equations, where linked streams of rotor and stator divisible by worth of carriage voltage marked as Φ_w and Φ_s, are co-ordinates of condition. What are more the parameters of the model are the relation's parameters.

Where:

M_{el} – electromagnetic moment,
M_u – breakdown torque,
k_w – coefficient of dispersing of rotor,
k_s – coefficient of dispersing of stator,
a_w, a_s – proportional parameters of engine model,
C – electromechanical constant of motor,
p – number of pairs of poles of asynchronous engine,
U – normalized excitation voltage,
ω_o – synchronous speed,
ω_e – electrical angular velocity,
I_w – rotor mass of inertia.

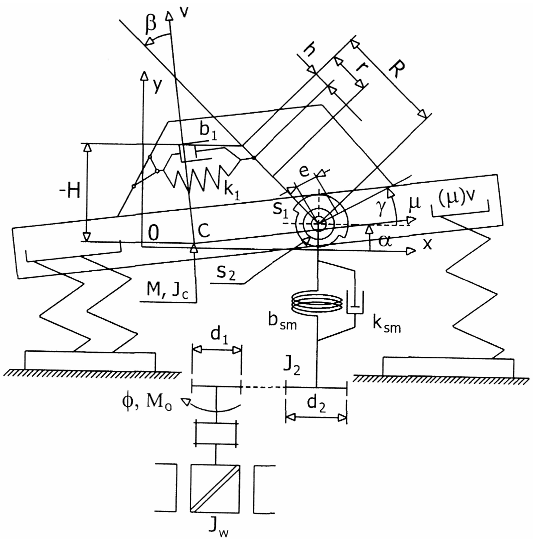

Figure 1 Physical model of vibratory machine with pendulum inertial vibrator

2.2. *The model of the mechanical part*

For the researching purposes, the mathematical models that describe relations between the body and the propulsion had been used during the digital simulation of dynamical events (ultra-resonance machine). These models afford possibilities for regenerating enormous part of difficult dynamical problems unable to explore on analytical ways. These dynamical events used to have considerable influence over rotary speed of the system.

Some of popular vibratory machines are the machines equipped with the pendular vibrator, which have got more latitude degrees than strong axis vibrator. If we are considering the machine with pendular vibrator, we would remember that model of the mechanical part include flat motion of the body, pendulum and pulp of the vibrator. Moreover, we would take care on relations between string-suspension and the body of the machine. Our calculation contains susceptibility of the clutch that keeps propulsion system with vibrator tighter.

$$\begin{aligned}&(M+m_h+m_n)\,x+[(m_h+m_n)\,H]\alpha+(m_hr+m_nR)\,\beta-[m_nesin\,(\gamma)]\\&=jj\sum[-k_{xx}\,(x-\gamma_i\alpha)-k_{xy}\,(y+\mu_i\alpha)-k_{x\alpha}\alpha]-b_xx+m_ne\gamma^2cos\,(\gamma)\end{aligned}\tag{5}$$

$$\begin{aligned}&(M+m_h+m_n)\,y+[m_necos\,(\gamma)]\gamma=jj\sum[-k_{yx}\,(x-\gamma_i\alpha)\\&-k_{yy}\,(y-\mu_i\alpha)-k_{y\alpha}\alpha]-b_yy+m_ne\gamma^2sin\,(\gamma)\end{aligned}\tag{6}$$

$$\begin{aligned}&[(m_h+m_n)\,H]\,x+\left[I_c+(m_h+m_n)\,H^2\right]\alpha+[(m_hr+m_nR)\,H]\,\beta\\&-[m_neHsin\,(\gamma)]\,\gamma=jj\sum-\{k_{\alpha x}\,(x-\gamma_i\alpha)+k_{\alpha y}\,(y-\mu_i\alpha)+k_{\alpha\alpha}\alpha\\&+\mu_i[k_{yx}\,(y-\gamma_i\alpha)+k_{yy}\,(y-\mu_i\alpha)+k_{y\alpha}\alpha]-\gamma_i[k_{xx}\,(x-\gamma_i\alpha)\\&+k_{xy}\,(y-\mu_i\alpha)+k_{x\alpha}\alpha]\}-b_\alpha+m_neH\gamma^2cos\,(\gamma)+b_1h^2(\beta-\alpha)\\&+k_1h^2(\beta-\alpha)\end{aligned}\tag{7}$$

$$\begin{aligned}&(m_hr+m_nR)x+[(m_hr+m_nR)H]\alpha+(m_hr^2+m_nR^2+I_h)\beta\\&-[m_nResin\,(\gamma)\,]\gamma=m_nRe\gamma^2cos\,(\gamma)-b_1h^2(\beta-\alpha)-k_1h^2(\beta-\alpha)\end{aligned}\tag{8}$$

$$\begin{aligned}&-m_nesin\,(\gamma)+m_necos\,(\gamma)-m_neHsin\,(\gamma)\,\alpha-m_neRsin\,(\gamma)\,\beta\\&+(I_n+m_ne^2)\gamma=-k_{sm}(\gamma-\phi\frac{d_1}{d_2})-b_{sm}(\gamma-\phi\frac{d_1}{d_2})-b_o\gamma^2sgn(\gamma-\beta)\end{aligned}\tag{9}$$

$$\begin{aligned}&\left[I_w+I_1+I_2\frac{d_1^2}{d_2^2}\right]\phi=-k_{sm}\left(\phi\frac{d_1}{d_2}-\gamma\right)\frac{d_1}{d_2}-b_{sm}\left(\phi\frac{d_1}{d_2}-\gamma\right)\frac{d_1}{d_2}\\&+M_{el}-M_osgn\,(\phi)\end{aligned}\tag{10}$$

Equations describes the flat motion of the machine body, the motion of pendulum, the motion of the vibrator, rotary motion of the engine and dynamical events in propulsion system.

Where:

x, y, α – horizontal coordinates, of perpendicular center of mass and angle of deviation from level of machine body,
β – angle of inclination of pendulum measured in relation to level,
φ – angle of rotation of rotor of engine,
γ – angle of rotation of vibrator measured in relation to level,
Φ_s, Φ_w – flux linkage of stator and rotor divided by reference voltage,
M – mass of machine body,
m_h – mass of pendulum,
m_n – unbalanced mass of vibrator,
I – referred to vibrator axis mass of inertia of driving system,
I_c – central mass of inertia of machine body,
I_h – central mass of inertia of pendulum,
I_n – central mass of inertia of unbalanced mass,
I_w – rotor mass of inertia,
I_1, I_2 – mass of inure of belt pulleys,
H – distance between axle of pendulum and center of mass of machine body,
h – distance between point of fastening of pendulum stabilization system and its rotation axle,
e – eccentric of vibrator,
R – distance between axle of pendulum and vibrator axis,
r – distance between center of mass of pendulum and its rotation axle,
d_1, d_2 – diameter of belt pulleys,
μ_i , ν_i – coordinate axes of fixing points of sets of elastic elements in relative central coordinate system $C_{\mu\nu}$,
k_1, b_1 – coefficients of elasticity and viscosity damping of pendulum stabilization system,
k_{sm}, b_{sm} – coefficients of elasticity and damping of rotation of coupling,
α_s , α_w – proportional parameters of engine model,
jj – number of elastic elements in set,
M_o – referred to vibrator axle anti torque moment of driving system,
M_u – breakdown torque,
M_{el} – electromagnetic moment,
ω_o – angular velocity of supply voltage,
b_o – resistance coefficient of bearings.

To realize all these purposes in designing vibratory machines it is necessary to use precise models of dynamic events in electromechanical system: vibratory machine - inductive drive. Presented model was verified and can be used for analysis and simulation of dynamics of vibratory machines in design process, virtual prototyping and mechatronical design. Presented model can be greatly useful for analysis of dynamics of machines with intelligent subsystems.

3. Piezoelectric elements in vibratory machines

Applying of piezoelectric elements in suspension systems of vibratory machines can give us possibilities for preparing ineligent solutions witch can greatly reduce problems of vibrations during passing thru resonance frequency. Folowing investigantins was made to test basic idea if such solutions can be usefull from practical point of view.

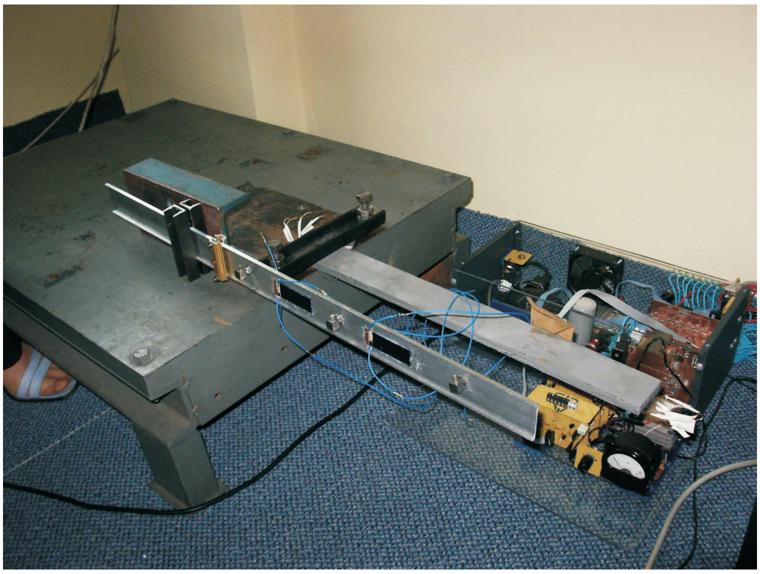

Figure 2 Test stand configuration with piezoelectric sensor / actuator

In this part it is described use of piezoelectric patches as actuator and sensor. To the experiment we used two patches attached to the top and button surface beam. One of the patches works as the sensor and operates due to the direct piezoelectric effect. The beam deflection develops the strain in the distributed sensor element. Since the sensor is relatively thin and perfectly bounded, the uniform stress distribution in the sensor cross section is assumed. and the stress value related to the beam. The second patches work as actuators. The tension and compression forces produced by the electrically activated piezoelectric actuator, are transferred to the structure by equivalent axial forces and bending moments responsible for a poor bending effect under consideration. The bending moment is distributed along the beam according to the shape of the effective electrode area, which usually coincides with the shape of the actuator. Piezoelectric patches was supplied with 50 V voltage. On the Fig. 2. natural frequency beam was shown. He has first natural frequency in 5 Hz and second 33Hz. When piezoelectric patches was used the beam reveals an increase in natural frequency during piezoelectric patches activation by 2 Hz as shown on Fig. 3. The vibration amplitude was decreased.

Measured but not presented here time characteristic shows that using of piezopatches gives possibilities of damping the vibration in 4 s. This characteristic prove that using piezopatches as sensor and actuator make the fast damping.

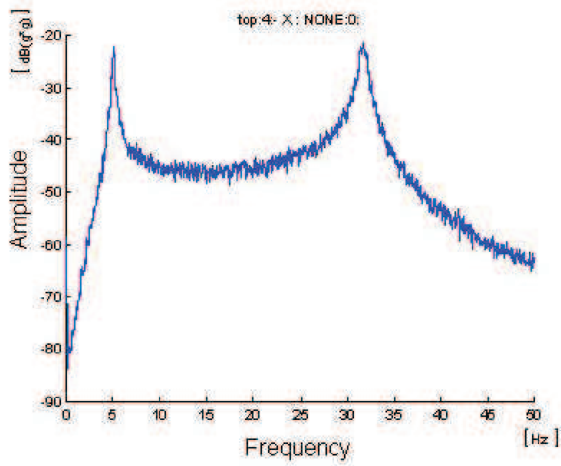

Figure 3 Natural frequencies of beam

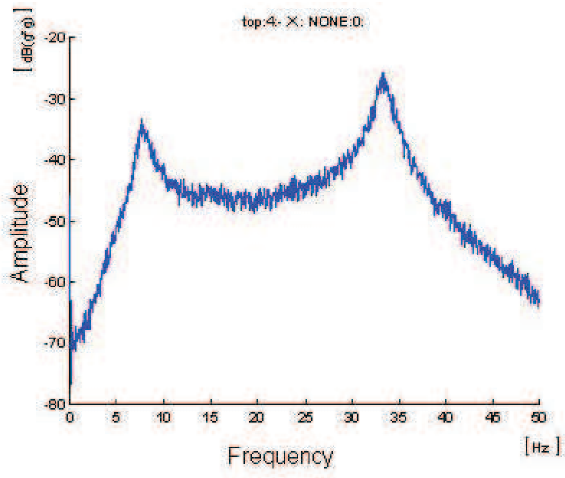

Figure 4 Vibration frequencies of beam witch piezoelectric patches

4. Conclusions

The research proved that piezoelectric patches could be applied in systems when control of dynamic features is required. Piezoelectric patches can be used in suspension systems of vibratory machines to prevent negative phenomenons concerned with passing thru resonance during starting and braking time.

References

[1] **Giergiel, M. and Sapiska-Wciso, A.**: ActiveVibration Control Using Piezoelectric Sensor and Actuator, ZN KMS Politechniki lskiej *Modelling and Optimization of Physical Systems*, No. 19, Gliwce **2006**.

[2] **Michalczyk J., Banaszewski T.**: Oddziaywanie dynamiczne na podoe mazyn stosowanych w przeróbce surowców mineralnych, **Wydawnictwo AGH**, Kraków, **2005**.

[3] **Srinivasan, A. V. and Mcfarland, D. M.**: Smart structures, *Cambridge University Press*, **2001**.

[4] **Michalczyk, J.**: Maszyny wibracyjne : obliczenia dynamiczne, drgania i haas. *WNT*, Warszawa **1995**.

[5] **Puchaa, A.**: Dynamika maszyn i ukadów elektromechanicznych. *PWN*, Warszawa **1977**.

This work is a part of the research project No. 4 T07A 026 30