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# Minimisation of resonance phenomena effect of piezoresistive accelerometer

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***Abstract*— In this paper, a suitable mathematical model of piezoresistive accelerometer is developed and validated by simulation tests. A new relationship is found between the movement relative frequency and piezoresistive accelerometer natural frequency. This relationship is represented by a new formula that selects for each frequency range a suitable piezoresistive accelerometer and thus, resonance effect is minimized.**

## I. Introduction

Monitoring by vibration analysis electromechanical systems of industrial plants is the preventive maintenance tool. This technique is largely used in most industrial system; it detects virtually any anomalies that may appear in rotating machines. Misalignment and bearing heating which is explained by a change of internal forces developed by the machine, this leads to a change in its vibration behavior [1].

It can track and monitor the device state, if vibration sensors are placed where these efforts are transmitted (on the bearings of machines). Measurements facilities, defects detection at an early stage, and the ability to conduct a diagnosis to determine the origin are the main advantages of vibration analysis technique. Measurement setup of this technique is composed from several devices such as accelerometer, amplifier and FFT analyzer. Improving the performance of one device of this setup can enhance greatly the vibration analysis technique. In this work, the accelerometer as the main and the most important element of measurement setup is improved and studied [1]. A new relationship links relative frequency of the movement and the piezoresistive accelerometer natural frequency to make an appropriate choice of accelerometer frequency range.

Several works were conducted on the optimization of piezoresistive accelerometer and the enhancements of various types of accelerometers are presented in [2-8].

## II. Piezoresistive Accelerometer Modelling

The operation principle of an accelerometer can be explained by a simple mass (m) attached to a spring stiffness (k) that in turn is attached to a casing, as illustrated in Fig. 1. The mass used in accelerometers is often called the seismic mass or proof-mass. In most cases the system also includes a spring to provide a desirable damping effect.

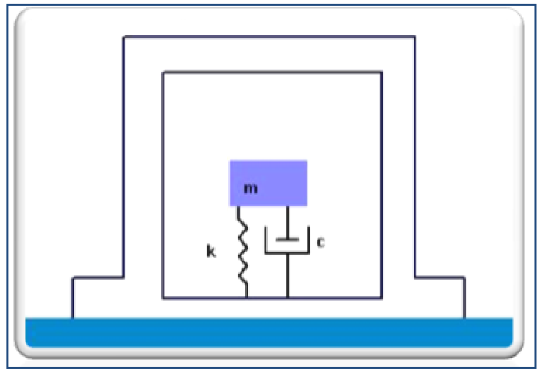


Figure 1. Piezoresistive accelerometer modeling.

The spring with friction coefficient (c) is normally attached to the mass in parallel to the spring. When the spring mass system is subjected to linear acceleration a force equal to mass times acceleration acts on the proof-mass, causing its deflect. This deflection is sensed by a suitable means and converted into an equivalent electrical signal. Therefore, a damping device is required; otherwise the system would not be stabile under applied acceleration.

To derive the system motion equation the Newton's second law is used, where all real forces acting on the proof-mass are equal to the inertia force on the proof-mass. Accordingly a dynamic problem can be treated as a problem of static equilibrium and the motion equation can be obtained by direct formulation of equilibrium equations.

The accelerometer is considered as a mass (m) mounted at the end of a beam producing a spring stiffness **k** and a damper providing a friction coefficient **c**.

The variables x and y are mass displacements. For an excitation force, where **Y** is the amplitude of vibration and **ω** the relative frequency, thus motion differential equation is:

$$z(t) = x(t) - y(t) \tag{1}$$

Where, z (t) represents the relative movement of the mass (m) according to the structure base, y(t) represents the amplitude of movement. By applying Newton law, the equation of the movement will be:

$$m\ddot{x}(t) = -k(x(t) - y(t)) - c(\dot{x}(t) - \dot{y}(t) \tag{2}$$

$$m\ddot{x}(t) + c(\dot{x}(t) - \dot{y}(t)) + k(x(t) - y(t)) = 0 \tag{3}$$

From Eq. 1 the following equations are obtained:

$$\dot{x}(t) - \dot{y}(t) = \dot{z}(t) \tag{4}$$

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