LABORATORY WORKPLACE FOR LOW FREQUENCY VIBRATION MEASUREMENT

Petr Fidler, Miroslav Uher
Doctoral Degree Programme (4, 2), FEEC BUT
E-mail: xfidle00@stud.feec.vutbr.cz, xuherm02@stud.feec.vutbr.cz

Supervised by: Petr Beneš
E-mail: benes@feec.vutbr.cz

ABSTRACT

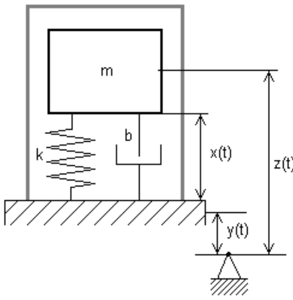
The target of our project is to create an illustrative stand for vibration measurement. The stand will be used in laboratory of non-electric quantities measurement at Department of Control and Instrumentation, FEEC BUT Brno. The workplace will be used for better understanding to the function of vibration sensors. This article submits statement of project solving and documents used solutions.

1. INTRODUCTION

Project is designated for variegating of education of non-electric quantities. The main target of the project is construction of a new laboratory stand, which is able to generate mechanical oscillations with relatively low harmonic frequencies. This stand will demonstrate problematic of vibration measurement to students in an illustrative form. Principle of vibration sensors, way of vibration measurement and relationships between displacement, velocity and acceleration of harmonic oscillations are shown in practice by the new workplace. Further, this stand aids the process understanding to terms such as absolute and relative vibration measurement and sensor behavior (inertial mass) in whole frequency range including resonance. The stand represents sensor such as vibrating element of second order and it will help in understanding to a sensor as a system with one degree of freedom. The use of this workplace will allow measuring through the use of several different types of sensors that can be based on different physical principles. Measuring control, data acquisition and data processing will be accomplished by use of USB measuring card and National Instruments LabView software.

2. ACCELEROMETERS – THEORY OF OPERATION

General model of absolute vibration sensor is shown on Figure 1. It consists of seismic mass m, spring with constant k and damping of value b. System is described by equilibrium of inertial, directive and damping forces, which is expressed by Equation 1. On frequencies lower than resonance there is a displacement of seismic mass proportional to acceleration of basis. This knowledge is used in accelerometers (e.g. in piezoelectric sensors). On frequencies higher than resonance there is displacement of mass proportional to vibration displacement and seismic mass creates referential point inside the sensor [1, 2].



$$m\frac{d^2z}{dt^2} + b\frac{dx}{dt} + kz = 0 \qquad (1)$$

Figure 1: General model of absolute sensor of acceleration [1, page 37].

On Figure 2 there are shown examples of typical bode magnitude and phase plot for vibrating system with mass $m = 0{,}1\ kg$, spring constant $k = 340\ N.m^{-1}$ and damping coefficient $b = 0{,}2\ N.s.m^{-1}$. The presented characteristics are displayed in dependence on normalized frequency ψ.

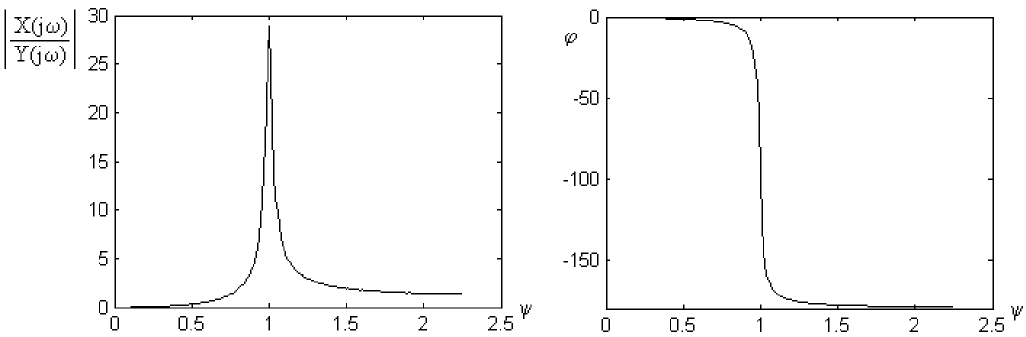


Figure 2: Bode magnitude and phase plot for general model of sensor.

3. DESIGN OF SECOND ORDER OSCILLATING SYSTEM

The most important part of the system is a spring, which affects usable frequency range and resonance frequency. The range of frequency is between 0 and 22 Hz. The choice of the spring was made in accordance with required frequency range (resonance in this frequency band and compliance to maximal force in tension and compression that can evoke its degradation). Due to simplification of design was created a numerical model which employs differential method for solving of second order differential equation. Optimal combination of spring and load is chosen on the base of this numerical model. The process of design of workplace is divided into steps represented by the following paragraphs.

3.1. FREQUENCY RANGE DETERMINATION

Frequency range was determined between 0 and 22 Hz, higher frequencies were not studied. Lower frequencies are more illustrative for students and facilitate visual understanding of the vibrating second order system close to resonance frequency, especially phase shift of mass displacement.

3.2. CHOICE OF SPRING RATE AND LOAD MASS

The spring constant in numerical model was chosen on the base of catalog values from spring manufacturer (Alcomex). Maximal occurring forces were checked in the numerical model with $m = 0{,}1\ kg$ and following values of resonance frequency were found: 8,2 Hz

(D1580), 9,5 Hz (D1540) and 11,6 Hz (D1580). The resonance frequency could be influenced by the change of seismic mass value. On Figure 3 there are shown calculated frequency dependencies of displacement for each of three studied springs with amplitude of excitation 10 mm.

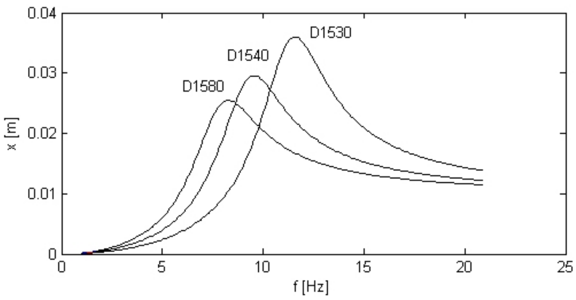


Figure 3: Frequency dependence of displacement seismic mass.

3.3. DAMPING OF SYSTEM

Determination of damping coefficient is difficult (friction, air resistance) and exact value should be measured by experiment. Damping is necessary for spring protection against exceeding of maximal force in system with high circuit grade. Magnetic brake (based on eddy-current) should be used for additional damping.

3.4. DYNAMIC RANGE OF SYSTEM

Dynamic range of the system is restricted by maximal compressibility (expandability) of spring, which corresponds to maximal force occurring on spring. These values can not be used from possibility of spring degradation. Values of dynamic range are connected with values of difference between normal length of spring and length during maximal compression. The differences for the springs are the following - spring D1530 24,2 mm, D1540 36,3 mm and D1580 40,7 mm.

3.5. DESIGN OF CONVERSION FROM ROTATING TO TRANSLATION MOTION

The basic part of the design driving segment is how to solve the conversion of harmonic rotating motion to harmonic translation motion. The problem is solved by the use of a flywheel (Figure 4 - 2) with fixed eccentric sliding hinge located in coulisse (Figure 4 - 1). The coulisse slides vertically on a conducting stake (Figure 4 – 3). Further, the seismic mass moves on the same stake. The coulisse is conducted in linear tracked ways (Linrol – Figure 4- 4).

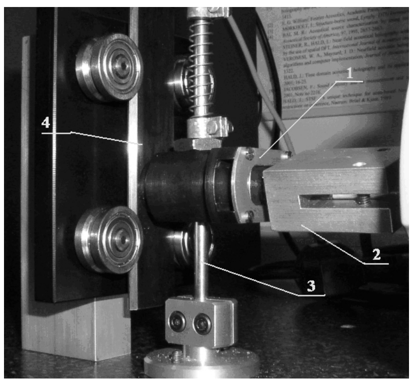


Figure 4: Conversion from rotating to translation motion.

3.6. DRIVING BLOCK

The actuation of the system is guaranteed by an asynchronous 3-phase motor with a short-circuit armature, produced by Siemens, with a corresponding frequency converter from the same manufacturer. The frequency converter is supplied by a one phase. Frequency control will be realized through executed from control software.

4. PRACTICAL REALIZATION

By reason of design simplicity, the vibrating system is placed on the same conducting stake as the driving coulisse. A value of seismic mass was determined by requests of minimal value (0,1 kg) and maximal dynamic range. The material of seismic mass and conducting stake is not allowed to be magnetically conductive. This request reflects supposed usage of magnetic damping. A hollow cylinder made of dural was chosen for seismic mass and stainless steel was chosen for conducting stake. There is a bronze element inserted into the seismic mass and there is a teflon element fixed on the conducting stake. These elements provide a sliding contact in two places. The growth of damping coefficient is caused by these elements.

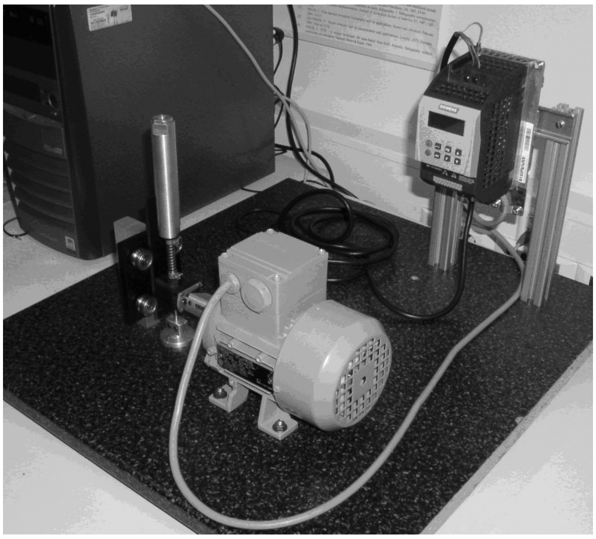


Figure 5: Completed workplace for vibration measurement.

4.1. MEASURED QUANTITIES AND USED SENSORS

Four sensors may be attached to the stand. Displacement acceleration of seismic mass and displacement and acceleration of basis (conducting coulisse) could be measured at the workplace. The main importance of the workplace in the sensor theory is explanation of accelerometer principle working in under-resonance band. The displacement between basis and seismic mass is proportional to acceleration of vibration in this band. The phase shift between both of displacements nears to zero. It is possible to obtain the constant of sensor (sensitivity) from the difference between output signal of displacement sensors and reference accelerometer placed on the basis. The displacement of seismic mass is proportional to displacement of basis on frequencies higher than resonance. In this case the phase shift nears to π. The seismic mass is damped out and its behavior is drawn closer to a reference point in which is measured the displacement of vibration. Inductive distance sensors Balluff BAW MKV-020.19-S4 for displacement measurement and accelerometers acceleration measurement are used on the workplace. The displacement measurement is adapted to the principle of fall sensor, respectively to mutual position of moving cone and hole inside sensor. Dynamical range and sensitivity of sensor is proportional to skewness of cone.

4.2. SIGNAL PROCESSING IN NI LABVIEW

National Instruments LabView is used for signal processing on the workplace. The studied quantities (displacements and accelerations) are captured by measuring card NI USB-6221. This card has 16 analogue (8 differential) inputs with 16bit A/D converter. The sample rate is 250 kS/s for all measurements. The single channel sample rate over 60 kS/s (on condition of using four channels) is suitable for designated frequency range. The measuring application vibrace.exe was developed in LabView. The application controls analogue inputs and displays measured data.

5. CONCLUSION

The main target of the project is the improvement of the quality of vibration measurement education in subjects Measurement of physical quantities (BMFV) and Sensors of non-electric quantities (MSNV). The current workplace consists of two shakers with piezoelectric accelerometers and a car model with capacitive impulse accelerometer. The designed stand exemplifies current education in bachelor course BMFV and creates new exercise in magister course MSNV. The stand is an ideal link between difficult theory and completed sensors that are presented on the current workplace like black boxes.

ACKNOWLEDGEMENT

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