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Design of Spring Mass Experimental Setup for Mechanical Design Laboratory

H. M. Naveen¹, Janardhan Reddy², Karthik Kundagoal³, Hugar Siddesh⁴, Keerthi K⁵

Assistant Professor, Department of Mechanical Engineering, RYM Engineering College, Ballari, Karnataka, India¹

UG Students, Department of Mechanical Engineering, RYM Engineering College, Ballari, Karnataka, India²³⁴⁵

ABSTRACT: The paper deals with design of experimental setup of spring mass system for mechanical design laboratory. Spring required is designed based on the fundamental design procedure and experiments are conducted. The effect of longitudinal vibration on an undamped condition and force exerted on the spring will be studied out in the preset context. The required is also designed in solid edge software and represented.

KEYWORDS: Design, Spring Mass System, Design Laboratory

I. INTRODUCTION

The spring mass experiment is used worldwide in all laboratory courses because it is physically and pedagogically rich, and its apparently simple contents lead students to face the gap between theory and practice, to program and master experimental techniques and to treat data statistically. In the course, spring-mass is one of the setup required in a sequence of six experiments, as follows: the pendulum, vertical oscillations with the spring-mass, spring-mass with two degrees of freedom, horizontal oscillations with two masses and three springs, vibrations in an elastic string, vibrations in a gas tube and the ripple tank. The idea is to study the single degree spring mass given their importance in many fields such as electronics, accelerators, quantum-mechanics.

In this context, mass and spring experiment is connected to the development towards frequency and stiffness. The course organization is based on the idea of creating groups of about twenty students at a time in a laboratory, working together as in a research group studying a certain topic. As such, the group head guides the experimental research project, presents the theory in a few lectures, describes the test strategy, defines steps and, very importantly, creates the occasions for discussions at crucial points. In the laboratory sessions sub-groups of four students are helped by assistants. Spring mass system is fundamental system which plays an important role in mechanical vibration hence considered.

II. LITERATURE REVIEW

The study of the movement experienced by a mass suspended from the free end of a spring is a topic discussed in most introductory physics courses, from both theoretical and experimental outlooks. The physics of the spring-mass system oscillations have been widely studied in a great variety of texts, in which the relationship between the period and the oscillation frequency is shown in detail [1]. Similarly experiments allow observing the dependence of the oscillatory systems on the mechanical forces as established by Hooke's law [2]. Some studies included corrections in order to take into account the influence of the spring mass on the oscillations of the spring-mass system.

The spring-mass experiment is a step in a sequence of six increasingly complex practicals from oscillations to waves in which we cover the first year laboratory program. Both free and forced oscillations are investigated. The spring-mass is subsequently loaded with a disk to introduce friction. The succession of steps is: determination of the spring constant both by Hooke's law and frequency-mass oscillation law, damping time measurement, resonance and phase curve plots. All cross checks among quantities and laws are done applying compatibility rules. The non negligible mass of the spring and the peculiar physical and teaching problems introduced by friction are discussed. The intriguing waveforms generated in the motions are analyzed. We outline the procedures used through the experiment to improve students' ability to handle equipment, to choose apparatus appropriately, to put theory into action critically and to practice treating data with statistical methods [1].

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Several mass–spring–damper models have been developed to study the response of the human body to the collision with the ground during hopping, trotting, or running. The mass, spring, and damper elements represent the masses, stiffness properties, and damping properties of hard and soft tissues. The masses that models are composed of are connected to each other via springs and dampers. The present paper reviews the various types of mass– spring– damper models including one-body and multi-body models. The models are further categorized as being either passive or active. In passive models, the mechanical properties (stiffness and damping) of soft tissues remain constant regardless of the type of footwear, ground stiffness, etc. In active models, the mechanical properties adapt to external loads. The governing equations of motion of all models as well as their parameters are presented. The specific ways that the models take account of the shoe–ground interactions are discussed as well. The methods used for determination of different modelling parameters are briefly surveyed. The advantages and disadvantages of the different types of mass–spring–damper models are also discussed. The paper concludes with a brief discussion of possible future research trends in the area of mass–spring–damper modelling [2].

The exact natural frequencies, mode shapes, and the corresponding orthogonality relations are important in forced vibration analysis via modal expansion. In the present paper, a free vibration analysis is conducted to determine the exact natural frequencies and mode shapes of an axially loaded beam carrying several absorbers. An explicit expression is presented for the generalized orthogonality relations. These generalized orthogonality conditions are employed along with the assumed modes method to perform forced vibration analysis. The present approach is compared to other approximate methods in the literature with the classical orthogonality relations and different choice of mode shapes. The results indicate that the use of the generalized orthogonality relation with the exact mode shapes is required for a precise investigation of the dynamic response of a beam with mass-spring-mass-damper system [3].

This paper presents a deformable model that offers control of the isotropy or anisotropy of elastic material, independently of the way the object is tiled into volume elements. The new model is as easy to implement and almost as efficient as mass-spring systems, from which it is derived. In addition to controlled anisotropy, it contrasts with those systems in its ability to model constant volume deformations. We illustrate the new model by animating objects tiled with tetrahedral and hexahedral meshes [4].

The frequency-response curve is an important information for the structural design, but the conventional timehistory method for obtaining the frequency-response curve of a multi-degree-of-freedom (MDOF) system is timeconsuming. Thus, this paper presents an efficient technique to determine the forced vibration response amplitudes of a multi-span beam carrying arbitrary concentrated elements. To this end, the "steady" response amplitudes |Y(x)|s of the above-mentioned MDOF system due to harmonic excitations (with the specified frequencies ωe) are determined by using the numerical assembly method (NAM). Next, the corresponding "total" response amplitudes |Y(x)|t of the same vibrating system are calculated by using a relationship between |Y(x)|t and |Y(x)|s obtained from the single-degree-offreedom (SDOF) vibrating system. It is noted that, near resonance (i.e., $\omega e/\omega \approx 1.0$), the entire MDOF system (with natural frequency ω) will vibrate synchronously in a certain mode and can be modeled by a SDOF system. Finally, the conventional finite element method (FEM) incorporated with the Newmark's direct integration method is also used to determine the "total" response amplitudes |Y(x)|t of the same forced vibrating system from the time histories of dynamic responses at each specified exciting frequency ωe . It has been found that the numerical results of the presented approach are in good agreement with those of FEM, this confirms the reliability of the presented theory. Because the CPU time required by the presented approach is less than 1% of that required by the conventional FEM, the presented approach should be an efficient technique for the title problem [5].

This paper deals with some dynamical behavior of Damped Non Homogeneous mass spring system. Both ordinary differential equations and discrete fractional order equations versions are considered. Stability at equilibrium positions are discussed with the analysis of Jacobian matrix. Also the results are illustrated with numerical examples with suitable parameters showing the rich dynamical behaviour [6].

The interaction between human and structure can produce significant dynamic effects. This has been demonstrated in several occasions including the closing of the Millennium bridge in London shortly after being open to traffic. Models based on springs, dampers and lumped masses have been widely accepted by the scientific community to model the human in human-structure interaction (HSI) problems. Recently, models of the human body based on control theory have been proposed. This paper provides a comparison between two traditional models using spring, dampers and lumped masses and those using control theory. The models are updated in a probabilistic sense using Bayesian inference. The experimental data used for the comparison is obtained from a laboratory test structure specially designed for HSI studies [7].

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Most of the applications today need dynamic systems modelling and their control. Natural systems that show the dynamic behavior of dynamic systems are simply a mass spring system, further we add damper for system flexibility. Currently most of the systems have multi Mass Spring Dampers like robot manipulator and Vehicle suspension systems use Double Spring-Mass-Damper system in planner and vertical configuration respectively. In this paper we will first model a general Double Spring-Mass-Damper System then we will analyse it using different methodologies and then control by the use of many continuous & digital control techniques. A Graphical User Interface is introduced that models, controls and analyzes the system with the application of many control techniques including Closed Loop, P, PI, PID, State Space, Bode Plot, Nyquist, Root Locus, and stability with the applications of both digital and analogue modes [8].

System Dynamics is a well formulated methodology for analyzing the components of system including cause effect relationships and their underlying mathematics and logic, time delays, and feedback loops. It began in the business and manufacturing world, but is now affecting education and many other disciplines. Having inspired by successful policy changes in lots of fields, the system dynamics researchers targeted to apply the system dynamics approach in the educational fields too [9].

This paper considers an inverse mixed eigenvalue problem for a damped spring-mass system. The problem of reconstructing the vibration system from its partial physical parameters, complex mode and a real mode of the corresponding undamped system is considered. By solving a kind of quadratic inverse eigenvalue problems, the necessary and sufficient condition for the solvability of the problem is obtained, and a numerical method for solving the problem is proposed. The numerical example shows feasibility of the method [10].

III. DESIGN OF SPRING

A spring is an elastic object that stores mechanical energy. Springs are typically made of spring steel. There are many spring designs. In everyday use, the term often refers to coil springs. They can return to their original shape when the force is released and it is defend as a elastic bodies.



Figure 1 Nomenclature of Spring

3.1 Design Procedure for Helical Spring [11]

Nomenclature of Helical Spring

c = Spring Index d = wire diameter D = Mean diameter of coil D_i = Inner diameter of coil D_o = Outer diameter of coil E = Young's Modulus F = Axial Force G = Modulus of Rigidity k = Wahl Factor L_0 = Free

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Length $L_s =$ Solid Length i' = Total number of coils i = Number of active coils p = pitch y = deflection $\tau =$ shear stress τ max = Max shear stress n= additional coils a= clearance

1. Diameter of wire

Shear stress $\tau = \frac{8FDK}{\pi d^3}$ Eq. 20.22 [12] Shear stress $\tau = \frac{\pi d^3}{\pi d^3}$ Eq. 20.22 [12] Stress factor K = $\frac{\frac{4C-1}{4C-4}}{\frac{4C-4}{C}} + \frac{0.615}{C}$ Eq. 20.23 [12] Spring index C = $\frac{D}{d}$

- 2. Mean diameter of the coil Inside diameter of the coil Di = (D-d)Outside diameter of the coil Do = (D+d)
- 3. Number of active turns or coils Mean deflection by $\frac{8FD^3i}{Gd^4}$ Eq. 20.29 [12]
- Free length of the spring 4. $l_0 = (i+n) d + y + a Eq. 20.53 [12]$ where a is the clearance between working coils under max load take a as 25% of max deflection 5. Pitch
- $P = \frac{l_0 2d}{i}$ Table 20.14 [12]
- Stiffness or rate of spring 6.

 $F_{0} = F/Y Eq. 20.30 [12]$

3.2 Illustration

Design a helical spring to support an axial load of 3000 N, the deflection under the load being limited to 60 mm. The spring index is 6. The spring is to be made of chrome-vanadium steel & a factor of safety is 2.

Given Data:

F=3000N, y=60 mm, c=6, FOS=2 where τ_e is elastic limit in torsion for chrome-vanadium steel for chrome vanadium steel, (Column. No 9 & 10) Elastic limit in torsion τ_e =690 MPa Rigidity modulus G=78450 MPa. Applying Factor of safety of 2, the allowable shear stress $\tau = 345$ MPa.

1. Diameter of the wire: Shear stress $\tau = \frac{8FDK}{\pi d^3}$ where c is the (C = $\frac{D}{d}$) K Wahl's stress factor K = $\frac{4C-1}{4C-4} + \frac{0.615}{C} = 1.2525$ 345 = $\frac{8\times3000\times6\times K}{\pi d^2}$ = d = 12.81 mm \approx 13 mm. πd^2 2. Mean diameter of the coil : $D = Cd = 6 \times 13 = 78mm$ Inside dia of the coil $D_i = (D - d) = (78 - 13) = 65 \text{mm}$ Outside dia of the coil $D_0 = (D+d)=(78+13) = 91$ mm Number of active turns (coils): 3.

Deflection = $\frac{8FD^3i}{Gd^4} = \frac{8 \times 3000 \times 78^3 \times i}{78450 \times 13^4}$ $i = 11.8 \approx 12 \text{mm}$

Free length of the spring: 4.

For squared & ground ends, free length $l_0 = (i+2) d + y + a$

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where a \approx clearance between working coils under maximum load Take a 25% of maximum deflection = 25% of 60 = 15mm $l_o \geq (12+2)13+60+15 = 257 mm$



Figure 2 Types Coil Ends

5 Pitch:

$$P = \frac{l_0 - 2d}{i} = \frac{257 - 2 \times 13}{12} = 19.25 \text{ mm}$$

6 Stiffness of spring rate: $F_0 = \frac{F}{y} = \frac{3000}{60} = 60 N/m.$

Specification:

- 1. Material = chrome, vanadium
- 2. Wire diameter d=13mm
- 3. Mean coil diameter D=78mm
- 4. Free length $l_o = 257$ mm
- 5. Total number of turns i = 14
- 6. Style of ends: Squared & ground
- 7. Pitch =19.25 mm
- 8. Spring rate = 50N/m

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3.3 Considered Experimental Setup



Figure 3 Experimental Setup



Figure 4 Masses for Conduction of Experimental

Figures 2 and 3 represent the experimental setup and masses taken into consideration for conduction of experiment.

IV. DESIGN OF SPRING IN SOLID EDGE SOFTWARE

Considered spring is designed in solid edge software and are represented in following figures 5, 6 and 7



Figure 5 Designed Spring in Front View

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Figure 6 Designed Spring in Isometric View



Figure 7 Designed Spring in Side View

V. CONCLUSION

By using the spring mass system the longitudinal vibrations of helical spring and frequency oscillations can be determined experimentally. The obtained values can be compared to theoretical values and validate the results.

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