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Study and Design of Eccentric Mass Absorber for Vibration System

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Abstract: Eccentricity vibration: - Eccentricity is defined as the offset between the axis of rotation and the axis of symmetric. Dynamic vibration absorber: It is a tuned spring mass system, which reduces or eliminates the vibration of a harmonically excited system. The concept behind these passive components is simply to add a spring and mass that have a natural frequency tuned to that of the resonant excitation frequency of the system. Doing so transfers all of the resonance energy of the system to the Dynamic Vibration Absorber, leaving the original system undisturbed. This study focuses on the optimum design of the damped dynamic vibration absorber (DVA) for damped primary systems. This design in the future to further characterize or improve the system performance

I. INTRODUCTION

We continuously experience vibration in our day-to-day life: the reason whywe can hear is due to the ability of our ear drum to oscillate due to sound waves and transmit those vibrations to our brain via the internal ear. Music instruments work on the principle of vibration of strings or diaphragms. Earthquakes can be felt and recorded because of the massive vibrations that are produced as a consequence. It can be safely concluded that movement ofconstituent particles is the origination of vibration in a material.

Similarly, vibrations also exist in machineries. However, vibration in this casecauses wear and tear to the system. Also, resonance is induced on approaching the natural frequency of vibration of the system which causes intense damage. Hence, the need arises to eliminate or at least reduce the vibrations in a machine to a minimum. A dynamic vibration absorber is sucha device. An undamped dynamic vibration absorber is basically a spring-mass system that transmits the vibration from the main system to an auxiliary system, preventing damage of the main system.

Vibration being an important part of our curriculum, our laboratory lacked an apparatus to enable us to observe and study the transmission and/or absorption of vibrations. Our project deals with equipping our laboratory witha simple apparatus that will enable the observation of dynamic vibration absorption and affect a better understanding of the phenomenon.

II. METHODOLOGY

A. Problem Review:

With the wide use of variable frequency drives, it is becoming more difficult to design mechanical systems free from natural frequencies within operating speed range. If such an occurrence is allowed in the field, a resulting resonance condition threatens to significantly impact performance and longevity of the equipment.

Since machines are made up of metallic parts, they have mass and elasticity both. Further if a machine contains any rotating of moving member; it is subjected to forces which vary periodically with time. These forces may or may not be harmonic in nature and result into „Forced- vibrations" in those machines. An electric motor or any other device with a rotor as its working component is called „rotating machine. The machine is said to have unbalance when the center of gravity of the rotor does not coincide with the axis of rotation.

Many systems, such as an internal combustion engine; a turbine in a power plant, operate at constant angular speeds. There is always a possibility that the frequency of excitation due to unbalance may match the natural frequency of the machine (main system). In such a case resonance will occur resulting in undesirable and harmful vibrations and loss of performance. This also limits the speed domain of a machine such that it can't be operated at a particular speed which is close to resonant speed though at that particular speed the performance of machine may be high.

On the other hand there are a number of practical situations in which the dynamic system is excited due to the motion of the base. A vehicle moving on a wavy road, a locomotive running on a rail track with gaps between the adjacent rails, a panel of measuring instruments subjected to excitation from the vibrating structure etc. In such cases also there is a chance of resonance to occur when the frequency of base excitation matches with the natural frequency of the machine.

Therefore, some viable solution to this problem is needed, which can eliminate or reduce the harmful and intolerable vibrations of machine so that it can perform as expected.

Let be the total mass of the machine including the rotor and let represents the amount of unbalance. Assuming that the machine is constrained to move in vertical direction, mainsystem has only 1 degree of freedom. The unbalance mass revolves with angular velocity in counter clockwise direction. Then will be the non-rotatingmass of the machine.

B. Fabrication Method

Following steps were followed for fabrication of setup for force excitation and the vibration absorber:

Selection of Motor:

Selection of a small sized, low hp motor whose speed can be varied by varying voltage supply. The motor that was used has following specifications

- Type – AC/DC (universal)
- Rated speed = 1000rpm
- Voltage supply = in put 230V single phase -out put 12+/-0.5
- Current = 300mA
- Mass = 150grm
- Diameter = 2.5cm; length = 4 cm.

C. Collection of Data and Calculations

Given data excitation=7hz=2pi7 red/sec stiffness =8000n/m Mass(m)=5kg

First we find the natural frequency (wn)=square root 8000/5 =40 red/sec=6.4hz

Design the dynamic vibration observer for natural frequency (or) excitation frequency

Wf=Wn=square root K/m where: k=2000n/m Wn=2*3.142*7

therefore;2*3.142*7=square root 2000/m m=44.72/43.98=1

m=1…. primary system

w1=7hz

Add secondary system Wa=4.5h and Wb=7.8hz

The advantage of the dynamic vibration obsorber increase the range of the vibration to a certain value

D. Determining the Minimum Motor Speed

With the help of auto-transformer available in the laboratory minimum speed of motor was checked by varying supply voltage. Minimum speed of motor was found to be 300 rpm.

E. Wooden Base and Unbalance Disc

A wooden base for clamping the motor and an unbalance disc were purchased in which amount of unbalance can be varied by increasing no. of nuts. Motor was centrally located on wooden base and clamped with nut-bolts.

F. Purchase of Springs

Now any combination of mass and spring would be useful if the natural frequency of the system made of them lies in speed range 300-4000rpm. Since main mass would be including motor, rotor, base etc ( ) adopting the speed range of 400-3000 rpm for ease, some springs of certain stiffness were purchased.

G. Determining the Spring Stiffness

To check the stiffness of springs, each spring was loaded progressively and change in length was recorded. Now plotting „load v/s deflection curve" which is a straight-line stiffness was obtained by slope of the line.

Stiffness of one of the main springs = 3270N/m

H. Calculation of Main Mass

Now total main mass was obtained by adding all sprung mass Main mass = (mass of motor + mass of wooden base + mass of rotor + mass of clamping accessories). It was found to be = 3.463 kg.

I. *Selection of Number of Springs*

Selection of no. of main springs would perform two tasks. Firstly, it would decide the combined stiffness of main spring in parallel arrangement; and secondly these springs would support the wooden base on which motor has been clamped.

- No. of springs used = 3
- Combined stiffness of main spring.

J. Calculation of Natural Frequency of Main System

Natural frequency of the main system was calculated and checked whether it lies in the adopted speed range otherwise no. of springs had to be changed.
Natural frequency, @ N = 500 rpm.
Arrangement of the Wooden Base
A wooden platform was arranged the top plate of which has a circular hole large enough to allow free movement of auxiliary mass across it, and that would support motor base by main springs.

K. *Mounting of Motor*

Wooden platform and motor base were connected together by 2 main springs using nut-bolts.

L. *Preparation of Auxiliary Mass*

Since auxiliary spring (stiffness = 3270 N/m) has been already purchased, an auxiliary mass was prepared with a hook welded on its top such that the natural frequency of auxiliary system matches with the natural frequency of main system, making it a tuned vibration absorber.

- Material used- Mild steel
- Auxiliary mass

M. *Mounting of Unbalance Disc*

Unbalance disc was tightened on the motor shaft.

N. *Arrangement of Auxiliary System*

Motor base, auxiliary spring and mass were provided with end hooks to couple & decouple main system and auxiliary system at will.
This completes the fabrication process of force excitation setup coupled with Dynamic vibration absorber.

O. Testing and Modifications

When both the machines are ready, the following testing procedure was conformed to:

P. *Testing of force Excitation Setup*

- Note down the specifications of machine, e.g., stiffness of main spring, main mass (including rotor), unbalance mass etc. and calculate the natural frequency of main system thus find speed of motor at natural frequency.
- Weight the auxiliary mass and find stiffness of auxiliary spring by plotting load deflection curve‟.
- Pick unbalance disk and load some known unbalance mass on it by applying nuts on the bolt provided on disk.
- Mount this unbalance disk on motor shaft by screw and connect motor to auto transformer.
- Set the auto transformer to zero voltage supply and switch it on
- Now slowly increase the voltage supply from auto transformer so that motor starts to rotate. Speed of motor increases as the supply voltage increases.
- Set the voltage supply to a certain value and measure the speed of motor using tachometer.
- At some particular speed of motor the vibration amplitude of main system will be highest. Set the voltage supply to a certain value where maximum amplitude of vibration is observed.
- Now machine (main system) is operating at a speed that corresponds to its natural frequency. Couple the auxiliary spring and mass with the main system and observe the absorption of vibrations.

- Calculate natural frequency shifts using formula and obtain two new speeds of motor which correspond to 2 new natural frequencies of the 2 DOF system (main system coupled with auxiliary system).

Q. Precautions

- Check for all fasteners are tight, and ensure there is no loose joint anywhere.
- Before switching on the auto transformer set it to zero voltage supply, then progressively increase supply voltage.
- Tight the screw which holds unbalance disk on motor shaft firmly, otherwise at higher speeds it may be thrown away.
- Calibrate auto transformer for motor speed also, since it would be difficult to measure motor speed when it is vibrating.

R. Modifications

When the machine was run first it was found that vibrations due to unbalancing force were occurring in horizontal plane also. It was due to the fact that motor was centrally clamped to wooden base and plane of rotation of unbalance mass was located at some distance from clamping, therefore unbalance force was generating a moment on wooden base resulting in horizontal oscillations of it. Since we have adopted single degree of freedom system, main system has to be made to vibrate in one direction only (1-DoF). To ensure vibrations of the main system would be in vertical plane only, some bearing surfaces were provided in the form of motion blocking rods clamped to wooden platform surrounding motor base and preventing vibration in horizontal plane.

S. Testing of Base Excitation setup

- Note down the specifications of machine, e.g. minimum and maximum speed available.
- Ensure for the variable speed drive that belt is tight enough on cones to avoid mutual slipping.
- Weight main and auxiliary mass. Also obtain stiffness of main and auxiliary spring by plotting load v/s deflection curve.
- Calculate natural frequency of the main system (mass & spring) and corresponding speed of rotation after reduction through variable drive.
- Connect motor to power supply and note down the speed available after reduction of speed through variable drive at the suspension end of cam motion generator.
- Suspend main spring & mass at the suspension end of cam motion generator, and vary the frequency of excitation (speed) by shifting of belt on variable drive and observe the vibration amplitude of main system.
- Set the speed of drive at which the vibration amplitude of main system is maximum. At this speed the machine is operating at its natural frequency
- Now couple the main system with auxiliary system (mass & spring) and observe vibrations of the system as a whole.
- Calculate natural frequency shifts using formula. Vary the frequency of excitation (speed) to lower and upper side and observe the vibration amplitude. Interpret results by graphs of „vibration amplitude v/s frequency ratio.

T. Precautions

- Clean cone surfaces to avoid slipping of belt on them.
- Tight the transmission belt so that it may not slip during operation.
- Do not shift the belt when machine is not running.

III. RESULTS AND DISCUSSION

A. Interpretation of Results

After performing tests on both the machines and taking out the experimental observations following results were obtained:

1) Undamped dynamic vibration absorber, when coupled to main system in„tuned" condition completely absorbs the vibration of main system at its natural frequency.

2) At an operating speed at lower or higher side of zero amplitude condition (r
= 1), vibration amplitude of the combined system increases and finally meets resonance condition at lower & higher natural frequencies of the system as a whole.
New resonance frequencies of the combined system subjected to force excitation
New resonance frequencies of the combined system subjected to base excitation

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3) Undamped dynamic vibration absorbers are extremely effective for constant speed machineries in a certain speed range. The spread of this working speed range depends on the mass ratio of the system. As in the testing of force excitation setup, working range of DVA is r = 0.82 to 1.25, for mass ratio μ =0.35. Whereas in testing of base excitation setup, working range of DVA was found to be r = 0.68 to 1.52
4) Auxiliary mass vibrates with low amplitudes which are in tolerable limits absorbing undesirable vibrations form main system. At an operating frequency away from natural frequency lower or higher side vibration transmission problem is insignificant.

IV. DISCUSSION & FIELDS OF APPLICATION

As the working principle of Undamped dynamic vibration absorber suggests, it can be applied to a machine or structure which is subjected to external excitation at a constant frequency. The experimental results obtained from the machines on which work is done verify the phenomenon of absorption of vibrations using DVA. Therefore, the undamped dynamic vibration absorber can be successfully employed in turbines & compressors of a power plant, motor of flour mill, on bridges when tuned to its natural frequency so that 56 when excitation frequency due to passing of vehicles or troops matches with natural frequency of the bridge so it should not collapse.It should be noted that the use of DVA with machinery should not be viewed only as a fix to a vibration problem. In some cases, when a machine structure has to be tall with a high center of gravity, a DVA can be designed to be built into a machine, very much like it would be installed in a tall building. For example, heavy motors designed for vertical installation often have their fundamental natural frequency (often called reed frequency) just slightly above the operating speed. If such a motor is installed above a pump on a pedestal, the system natural frequency can get dangerously close to the operating range, causing a resonance. A DVA could be incorporated into the motor structure and tuned appropriately to prevent resonance vibration. For this purpose an element of the motor structure, such as a fan cover, may serve as an absorber mass if it is mounted to the motor by elastic springs.

V. CONCLUSION AND FUTURE SCOPE

This project work covers the utility of „Undamped Dynamic Vibration Absorber‟ (UDVA) in its simplest form which comprises of an auxiliary mass and a spring. UDVA can be of some other forms also which depends on its application. Such as in a vertical motor it can be a fan or flywheel connected through elastic spring, it can also be in the form of an auxiliary mass with enclosed air or a beam of metallic strip as variable spring element.

However, with incorporation of a damper in auxiliary system a better attenuation of vibration in a wider range of excitation frequency can be achieved by selecting a damping element with suitable damping coefficient. DVA‟s can be designed for tall buildings and structures also in suitable form which will keep safe those structures from earthquake and wind induced vibrations

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