Forced Vibration Analysis of Dashpot using Rotating Unbalance

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Abstract— Two different paths are followed for the analysis of dashpot. In the first path, the concept of coefficient of restitution is used assuming two impacts per cycle symmetrically. In the other path impact forces are split assuming periodic impacts into harmonic functions through Fourier series expansion. The main concept of vibration analysis of impact damper is based on impact duration & impact interval. Study of impact damper is essential to reduce the amplitude of vibration so that the structure can be safe against vibration hazards. Mostly researchers adopt duration of contact to be zero. There is no experimental analysis available for the same. So, we made an experimental setup to determine the characteristic of different parameter experimentally for the case of periodic steady state vibration. For the experimentation forced vibration is created by unbalance rotating masses. Experimental outputs were studied on Digital Storage Oscilloscope. (DSO) The results of experiment can prove that the assumption of impact damper is wrong. And this work can give new approach to solution of vibration analysis in future.

Key words: Dashpot (damper); forced vibration; Optimum clearance; Rotating Unbalance

I. Literature Review

M.M Sadek [1] assumed two un–equi-spaced impacts per cycle in the steady state and resolved the impact forces into harmonic forces by use of Fourier series on the assumption that the impact duration is nearly zero and thus found out the amplitude of steady state vibration through the concept of conservation of momentum. He concluded that two equi-spaced impacts are not possible for the efficient behaviour of impact damper. All other authors have assumed two equi-spaced impacts and analyzed the system using the concept of coefficient of restitution. S F Masri [2] analyzed the motion of the primary system for the number of impact dampers in parallel by assuming the system as a piecewise linear process. He showed that the system is stable if modulus of all Eigenvalues of a matrix is less than unity. Grubin [3] assumed a steady state and a linear system and expanded the motion between impacts into second order polynomials and found out an answer irrespective of damper parameters. Cheng and Weng [4] discussed the behaviors of a resilient impact damper in free damped vibration. The results show that the clearance between two masses in an effective impact damper is smaller than twice the initial displacement of the main mass in the vibration system if the system is simulated by an initial displacement only. Jianlian Cheng and HuiXu [5] investigated the behaviour of vibration system suppressed with an impact damper where the impact damper is simplified as a combination of spring and viscous damping. The results clearly show that the reduction of vibration response does not depend on the number of impacts, but primarily on the collision that occurs while the impact mass and main mass are moving towards each other. Peterka Frantisek [6] examined the dynamics of impact dampers using analogue computer simulation which consists of small mass which can decrease resonance amplitudes of vibrating system due to mutually impact interactions. He concluded that the system wholly eliminates the main resonance amplitudes of the fundamental vibrating system with one degree of freedom system owing to the addition of damper. Fred Akl [7] performed multiple linear regression analysis using Stat View software. From the numerous tests conducted, it was observed that in order for the damper to be active the test structure should be driven at a certain level of excitation necessary to overcome friction between the damper and guide rods on which it travels. He concluded that impact dampers are effective in increasing damping ratio of lightly damped flexible structures which attributes to the impulse and momentum equation of the test structure. K.Li, A.P. Darby [8] investigated the impact damper which is a freely moving mass. He concluded that the buffered impact damper is less sensitive to variation of excitation type and clearance and mass parameters of the damper itself, and results in quicker attenuation in the free vibration response. A higher coefficient of restitution given by elastic buffered impact damper results in higher impulse momentum and increased transfer of kinetic energy from the structure to the damper mass. E- Dehgan Niri and SM. Zahrai [9] studied the performance of a single conventional impact damper in both wide range frequency and resonance excitations. He concluded that larger the coefficient of restitution, the more efficient and reliable is the impact damper. S. Chatterjee [10] discusses a new principle of active vibration control of lightly damped flexible structural members. He proposed a dynamic control to generate a hysteric control commands for expanding and contracting the actuators. He concluded that the control laws are robust in a sense that the success of these laws do not depend on any explicit model of system dynamics. Satoshi Ema, Ema Marui [11] developed an impact damper to improve the damping capability of long, thin cutting tools, such as drills or boring tools. It was concluded that frequency of a vibratory system with an impact damper is approximately decided by the natural frequency of the system and mass ratio. These assumptions need to be further investigated. Hence, an experimental setup is designed to analyse the differences between the assumptions of researchers in this area.

II. Introduction

An impact damper consists of mass freely moving inside a cavity of main vibrating system. It is used to reduce vibration of main mass system for both steady state and transient motion. The cases of failures of structures and machines due to different types of loads motivate further research on how to reduce the effects of vibration. This effect may be expressed in terms of different vibration levels which cause fatigue failure of machine parts subjected to cyclic loadings. An effective reduction of the excessive oscillations can be obtained by the well- known tuned mass damper. This type of damper is successfully used in several civil engineering structures. Impact damper basically consists of single mass in cavity to improve stability and reduced vibration. It also holds a small clearance to the structure. The impact damper is usually a mass placed inside the structure and holds a small clearance to the structure. There is not yet any setup made to find out optimum clearance i.e. minimum clearance so that vibration is reduced and impact damper will be effective. The topic chosen is the "FORCED VIBRATION ANALYSIS OF DASHPOT USING ROTATING UNBALANCE".

The experimental setup is prepared for the following situations:

- To manufacture a standard setup which will be used to find out operation of dashpot,
- The setup will also be used to find optimum clearance of dashpot.
- Experimentally we will also find out the number of impacts for the steady state motion.
- Comparison of experimental readings with Theoretical calculations.

III. Problem Definition & Methodology

Vibration has its own merits and demerits, demerits of vibration include engine, equipment's etc.

The structures designend to support heavy centrifugal machines, like motors and turbines, or reciprocating machines, like steam and gas engines and reciprocating pumps, are also subjected to vibrationn. In all these situations, the structure or machine component subjected to vibration can fail because of material fatigue resulting from the cyclic variation of the induced stress.

Damping the vibration to the maximum extent is the major problem. The performance of the damper depends on its clearance. Finding the optimum value of the clearance at which damper gives maximum damping effect is the ultimate aim. At optimum clearance vibration will be damped to the maximum extent.

In methodology First step is to fabricate experimental setup in which damper can be incorporated and second is to fabricate damper along with damper slot having adjustable clearance mechanism. Analysing experimental results using Digital Storage Oscilloscope (DSO).

IV. Generation Of Forced Vibration

$F_o = m_o e\omega^2$

$F = F_o \sin(\omega t)$

Equation of motion

$\mathbf{m\ddot{x}+c\dot{x}+kx=F_o\sin(\omega t)}$

HERE $F_o = m_o e\omega^2$

Hence Equation of motion becomes

$m\ddot{x}+c\dot{x}+kx = m_o e\omega^2 \sin(\omega t)$

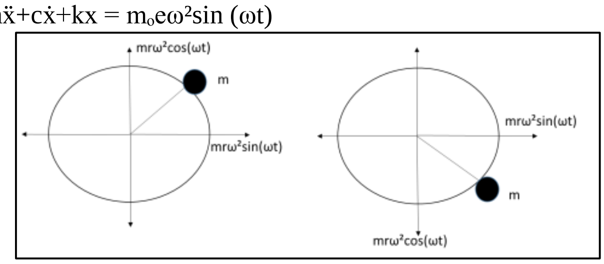

Fig. 1:

$\Sigma F_y = 0$ (1)

$\Sigma F_x = 2mr\omega^2 \sin(\omega t)$ (2)

Validation of about results:-

(↑ +ve)

$\Sigma F_y = mr\omega^2\cos(\omega t) - mr\omega^2\cos(\omega t)$

$= 0$

(→ +ve)

$F_x = mr\omega^2\sin(\omega t) + mr\omega^2\sin(\omega t)$

$= 2mr\omega^2\sin(\omega t)$

*If ωt is 90° then

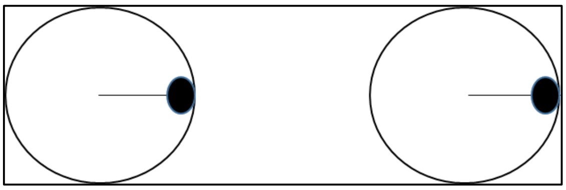

Fig. 2:

There will be no vertical component and hence

$\Sigma F_y = 0$ And

$\Sigma F_x = 2mr\omega^2\sin(\omega t)$

Since both forces acting from left to right

Same is the case for ωt =270°

If ωt is 180° then

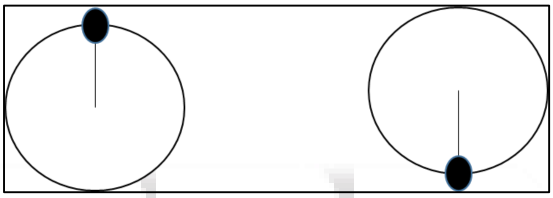

Fig. 3:

There will be no horizontal component

$\Sigma F_x = 0$

And there will be only vertical components are present but they are opposite in direction with each other

Therefor $\Sigma f_y = 0$

Same in the case for ωt =360°

In this way, we will get the transverse vibration which is requirement of the project

V. Calculation Of Natural Frequency

$(W_n)^2 = K/m$ and $K = W/\delta = 3EI/l^3$

$W_n = 10$ Hz $= 20\ \pi$ rad/sec

$400\ \pi^2 = K_1/\ 16.5$ $K_1 = 65.13*10^3$ N/m

FOR CANTILEVER PLATE

$K_1 = 3EI/l^3$ $\quad 65.13* 1000 = 3* (2*10^{11})\ I/(0.3)^3$

$I = 2.93*10^{-9}\ m^4$

Now we want the vibration in the transverse direction so base should be more than the height because moment of inertia should be less if we want the vibration to be more.

Hence we take b= 5h and $I = bh^3/12$

$2.93*10^{-9} = 5h*h^3/12$

Hence h= 9.15*10^-3 m = 9.15 mm ~ 10mm

For cantilever plate b= 50 mm and h= 10mm

For main plate b= 3mm, h= 200mm and l= 400mm

FOR MAIN PLATE

$I = bh^3/12 = 0.003 * 0.2^3/12 = 2*10^{-6}\ m^4$

$K_2 = 3EI/l^3 = 3*2*10^{11}*2*10^{-6}/0.4^3$

$= 18.75*10^6$ N/m

The two plates are in series, therefore equivalent spring stiffness is equal to

$1/K_{eq} = 1/K_1 + 1/K_2$

$= 1/65.139*10^3 + 1/18.75*10^6$

$K_{eq} = 64.935*10^3$ N/m

$w_n^2 = K/m$

$w_n^2 = 64.935*10^3/20$

$w_n = 56.978$ rad/sec $= 56.978/2\pi = 9$ Hz

VI. EXPERIMENTAL SETUP

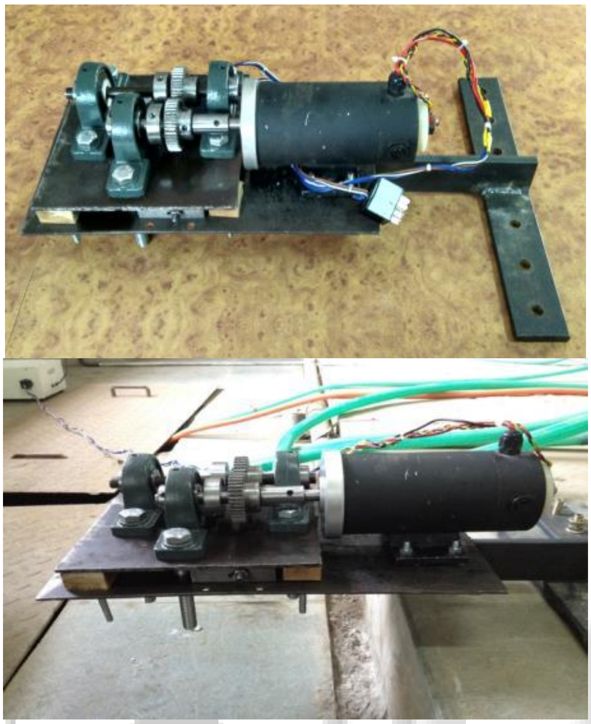

Fig. 4: Experimental Setup

The primary objective of this project is to find out impact duration and impact interval of the damper. As we know that the 3 regions in vibrations are of utmost important that is at resonance, below resonance and above resonance. So keeping this in mind, we have selected the resonance speed at 600 rpm. After selecting the resonance speed we calculated the dimensions of the main plate and cantilever plate. After calculating the dimensions, experimentally we got resonance speed at 600 rpm.The setup consists of a base plate welded to the free end of a cantilever beam. The beam has a very high stiffness in vertical direction as compared to its stiffness in the horizontal direction. A DC motor ranging from 0 to 1500 rpm mounted with a converter and a dimmer stat to vary the speed. The motor is connected to a flexible coupling to take care of the misalignment between the gear and the motor. Two spur gears are then used for speed reduction on two different shafts. One gear is on the same shaft on which motor is attached and those spur gears are then coupled with two roller disks having eccentric masses one on the top side and other one on bottom side of the disks. If we want more than one impact in one cycle, then the horizontal displacement should be more. Some part of it is developed by a spring to a certain amount. But the rest horizontal force can be developed by taking help of centrifugal force which means we need to develop thrust in the direction of motion. This is possible by taking two roller disks with eccentric masses one having eccentric mass at the top side and other at the bottom side in such a way that the vertical forces balances each other and we get a resultant horizontal force of twice the magnitude which is centrifugal force. We have found out the time interval between two impacts and also we have seen that the impacts are equally or unequally spaced. The free mass will slide on a linear slider bearing which has low friction.

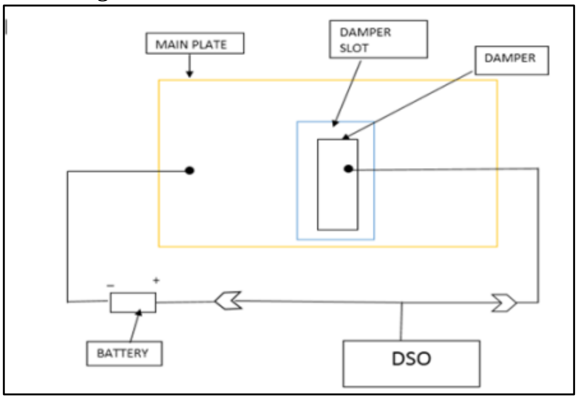

Fig. 5: Schematic block diagram of damper connection

In the above diagram, the blue colour block is the rectangular slot which is covered with acrylic plate for the circuit purpose and the black colour block is the damper which is placed in the slot. The damper strikes the slot on both sides when the vibration occurs. Readings are being taken on Digital Storage Oscilloscope (DSO) attached to one end of the battery. Since the motor speed range is from 200 rpm to 1800 rpm so to take at least 2 readings above resonance and 2 readings below resonance we have taken the natural frequency of the system at around 600 rpm. $\omega n =$ 10 Hz. Readings have been taken on one damper which is 3 percent, of the main mass.For every rpm, clearance has been changed 0.08, 0.24, 0.4, 0.8 mm and readings have been taken on DSO (Digital Storage Oscilloscope) and variation of clearance and rpm have been plotted.

VII. PROCEDURE FOR CLEARANCE CHANGING

Now for changing the clearance we have taken small circular disc and divided it in 10 parts each part at an angle of 360 so that the total circumference comes out to be 3600. But first we have made a hole at the centre with pitch 0.8 mm so that each division automatically becomes 0.08 mm and one complete round of the disc comes out to be 0.8 mm. So we have inserted a M6 screw of nearly 3 inches in that disc hole and in this manner we are changing the clearance manually.

Fig. 6:

VIII. GRAPHS FROM EXPERIMENTAL READINGS

A. *Optimum clearance vs RPM for Damper*

The set-up is vibrating & and then external force is applied to the setup. After the application of force the setup will continue to vibrate & then it will come to steady state motion. The clearance at which the steady state motion is obtained is called as optimum clearance. The optimum clearances for particular RPM are shown in below graph.

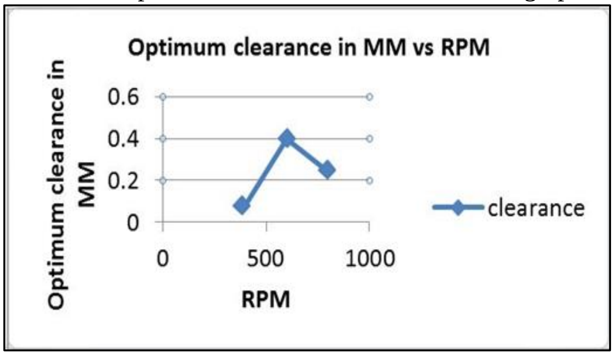

Fig. 7:

It is the essential parameter which have evaluated experimentally, since at optimum clearance effectiveness of damper is maximum as it is absorbing the energy from main system & bringing it to the steady state.

B. Variation of rpm with Number of Impacts of damper mass 3 present of main mass

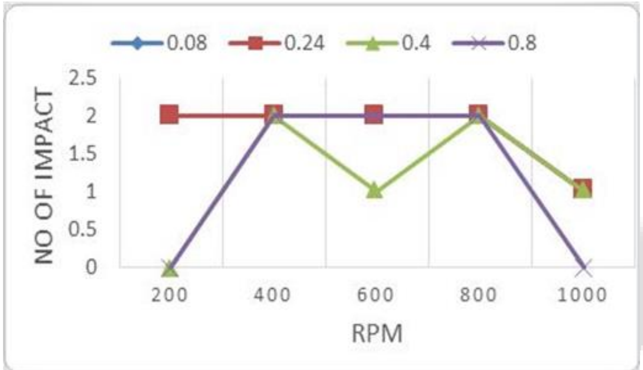

Fig. 9: Variation of rpm with Number of Impacts of damper for 0.08 mm,0.24mm, 0.4 mm and 0.8 mm clearance

C. Variation of rpm with Impact Duration of damper mass 3 percent of main mass

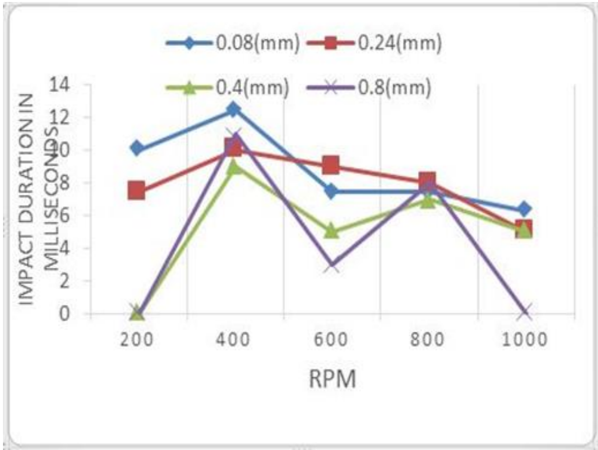

Fig. 10: Variation of rpm with Impact Duration of damper mass 3 percent of main mass for 0.08 mm, 0.24mm, 0.4 mm and 0.8 mm clearance

IX. INFERENCES FROM GRAPHS

For damper mass, optimum clearance initially increases as speed increases and then after the resonance it decreases as speed increases . The maximum value of optimum clearance is only at the resonance

For damper, the duration in which the impact does not occur in one cycle continuously decreases as the rpm increases because the impact duration changes with the impact interval in such a way that the non-contact duration goes on decreasing with increase in rpm. At large clearance and low speed, duration and interval of impact is zero. Similarly at large clearance and at high speeds also duration and interval of impact is zero.

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