AN EXPERIMENTAL STUDY ON IMPACT DAMPERS

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Abstract: Effect of an impact damper to reduce the vibration amplitude of the main mass system is well established. Two different paths are followed for the analysis of impact damper. 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. Using the second path Sadekpoints out that symmetric impact with equal time intervals is not possible. However most of the literature follows the first path. Duration of contact is assumed to be nearly zero. However no theoretical analysis is available to determine the contact duration between the masses or to determine symmetry/non-symmetry of interval between the impacts. This work aims at determining the two quantities experimentally for the case of periodic steady state vibrations caused by the unbalanced rotary masses. Effect of speed (frequency), clearance and mass ratio on impact interval and impact duration is studied.
Keywords: Impact damper, number of impacts, experimental verification of impact duration and time interval, optimum clearance.

1. INTRODUCTION

1.1 An impact damper consists of a mass freely moving inside a cavity of the main vibrating system. It is used to reduce the amplitude of vibration of the main system either for the steady state periodic motion or for the transient vibration of the main system. Its effectiveness is now accepted and depends on the mass of the damper, clearance and amplitude of the motion. The cases of failures of structures and machines due to different types of loads motivate further research on how to reduce the vibration. The effect may be expressed in terms of different vibration levels which cause fatigue failure of machine parts subjected to cyclic loadings. In order to suppress chatter vibration and to improve the stability, many types of dampers have been used, and viscous materials and frictional force in contacting parts have been utilized. 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 where viscous damping of tuned mass dampers is used with different

hydraulic mechanisms. The impact damper is usually a mass placed inside the structure and holds a small clearance to the structure. There is not yet any experimental setup mentioned to find out whether the impacts are equally or unequally spaced and what are the numbers of impacts per cycle.

M.M Sadek [1] assumed two un–equi-spaced impacts per cyclein 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 analysed the system using the concept of coefficient of restitution. **S F Masri [2]**analysed 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 Eigen values of a certain 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 irrespective of damper parameters. **Cheng and Weng [4]** discussed the behaviours 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 behaviours 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- DehganNiri 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. **Satoshi Ema, Ema Marui [10]** 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.

2. EXPERIMENTAL SETUP

2.1 The primary objective of this work is to find out impact duration and impact interval of the dampers. The effect of clearance on amplitude, effect of mass ratio on optimum clearance and effect of damping coefficient on clearance is also seen by plotting graphs between them. Also the results are supported by numerical analysis in ADAMS software by simulating the model in it. 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, the resonance speed is selected at 600 rpm. After selecting the resonance speed, the dimensions of the main plate and cantilever plate are calculated. Experimentally, the resonanace speed is observed at 500 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 with speed range from 0 to 1500 rpm of 0.5 hpis mounted on the plate with a converter and a dimmerstat 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 placed diametrically opposite to each other.If we want more than one impact in one cycle, then the vibratory displacement should be more. The horizontal force can be

developed by taking help of centrifugal force so that we need to develop thrust in the direction of motion. This is possible by taking two roller disks with eccentic masses having eccentric mass at diametrically opposite position in such a way that the vertical forces balances each other and we get a resultant horizontal force of twice the amplitude ofthe centrifugal force.The free mass will slide in a slot in the direction of motion which has a low friction.
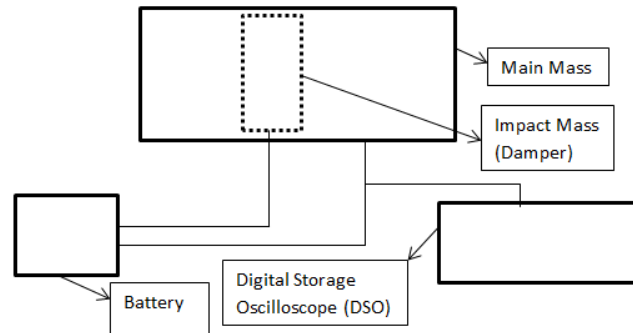

Fig.1 Schematic block diagram of damper

In the above diagram, the thick line block is the rectangular slot which is covered with acrylic plate for the circuit puropse and the light line 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.

Fig.2 Experimental Setup

Since the motor speed range is from 200 rpm to 1800 rpm, at least 5 readings above resonance and 5 readings below resonance have been taken. The natural frequency of the system is at 500 rpm. w_n = 8.33 Hz Data: Total mass of the system = 16.45 kg, Eccentric disc outer Diameter = 46 mm, Thickness of Disc t = 12 mm, Eccentricity r = Distance from centre of disc to centre of eccentric hole = 14 mm, Eccentric hole diameter = 12 mm

Readings have been taken on four dampers which are 3 percent, 2 percent, 1 percent and 0.6 percent of the main mass. For every damper mass and at every rpm, clearance has been

changed from 0.08 mm to 1.12 mm at a gap of 0.08 mm and readings have been taken on FFT (Fast Fourier transform) analyser and variation of clearance and amplitude have been plotted.

3. MODELING IN ADAMS

3.1 To compare the experimental results with software simulation, a model is made in Adams as shown in the above figure. In the figure, the red colour solid block is the main mass of the system. The red colour spiral rod on the left side is a translational spring damper. The red arrow on the right side is the harmonic force applied to the system for the excitation. The blue colour base on the bottom is the base for the main mass to slide on it and thereby give the support to the main mass. The green colour pendulum bob is the damper of the system. According to hertz contact theory in Adams, if we make a rectangular slot in Adams, then the point of contact changes in the simulation from point to line and then to surface. To avoid that we have made a circular damper so that the contact problem does not occur. A translational joint, rotational joint, fixed joint and a contact joint is applied to the model. In this way, a model is built in Adams so that the experimental and theoretical results are matched with the simulation results. For the proper modelling in Adams, the centre of gravity of the green colour damper, the red colour main mass, the red colour translational spring damper on left side and the harmonic force on right side all should be in one line.
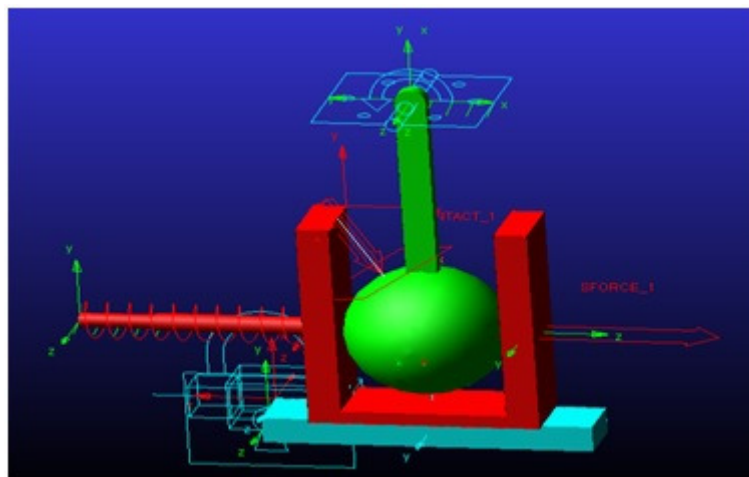

Fig. 3 Adams Model

4. EXPERIMENTAL OBSERVATIONS AND DISCUSSIONS

The following results are observed during experimentation of the damper which is 3 percent of the main mass where amplitude ratio is defined as the ratio of amplitude with damper to amplitude without damper expressed in percentage.
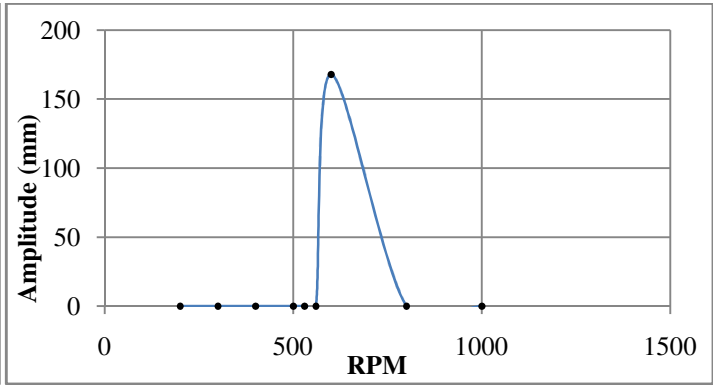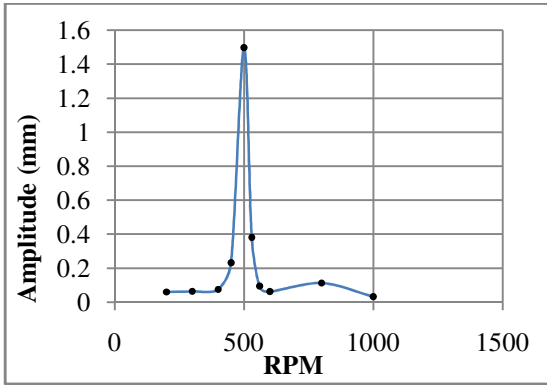
Fig.4 Amplitude without Damper **Experimental**   **Fig.5** Amplitude without Damper **Theoretical**
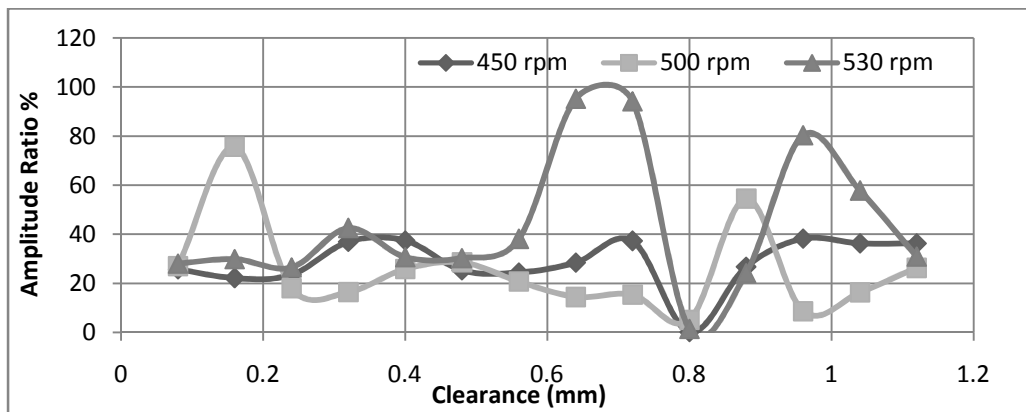

Fig 6: Variation of Amplitude Ratio with Clearance of damper mass (492 grams) 3 percent of main mass for 450 rpm,500 rpm and530 rpm.

In the above graph amplitude ratio is sensitive above resonance and it is least sensitive below resonance because below resonance amplitude is less as compared to resonance and above resonance.
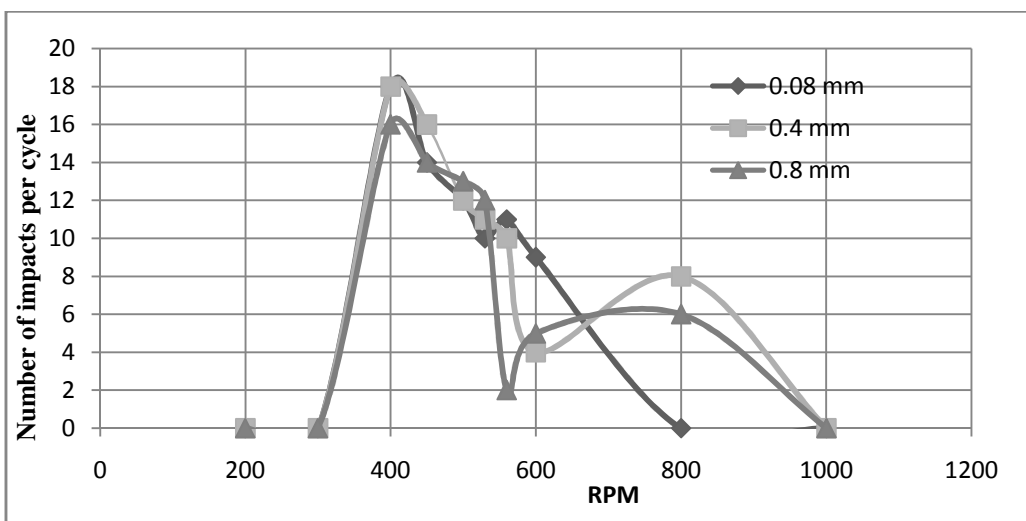

Fig 7: Variation of Number of Impacts with rpm of damper mass (492 grams) 3 percent of main mass for0.08 mm, 0.4 mmand 0.8 mm clearance.
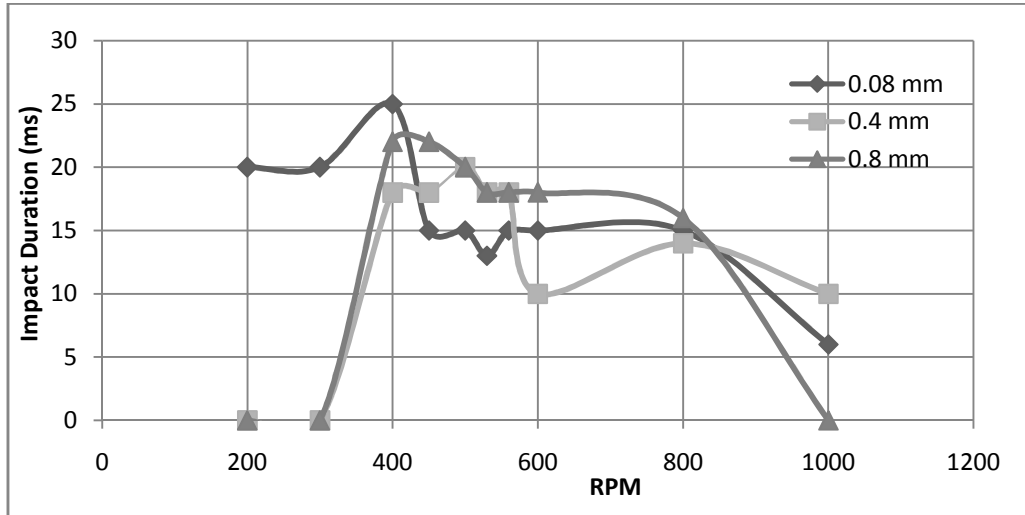
Fig 8: Variation Impact Duration with rpm of damper mass (492 grams) 3 percent of main massfor 0.08 mm, 0.4 mm and 0.8 mm clearance.

For the system under consideration, without any damper its amplitude is likely to be as follows:

450 rpm $\qquad$ X = 7.68* 10^-2 mm
530 rpm $\qquad$ X = 0.1368 mm
For $\dfrac{\omega}{\omega_n} = 0.99$ $\qquad$ X = 0.95 mm
For $\dfrac{\omega}{\omega_n} = 1.01$ $\qquad$ X = 0.91 mm

Below 450 rpm, X will be less than 0.05 mm while above 530 rpm, X will be nearly 0.1 mm. Thus below 450 rpm, the main system has very little energy to drive the damper while above 530 rpm it has a constant energy which is still small. Only in the resonance region, the main system has amplitude which is nearly 1 mm. These amplitudes should be compared with the clearance values.

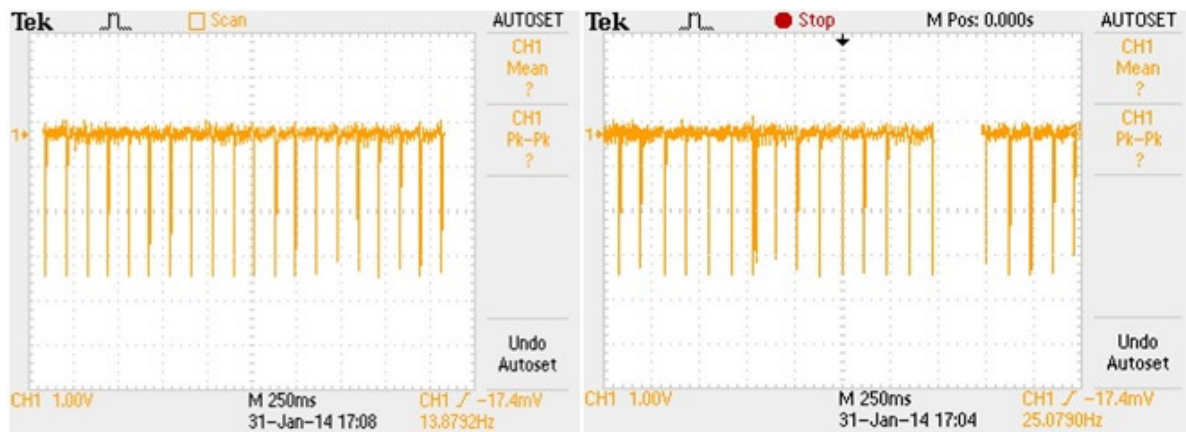A sample graph of the circuit output on the oscilloscope is shown in the figure.

Fig.8 Oscilloscope Sample Reading at resonance

Circuit was so arranged that the contact between the damper mass results into full battery voltage while the circuit voltage is zero when there is no contact. As this figure shows there is an appreciable time of contact. This phenomenon is particularly observed at low speeds when the amplitude of vibration of the main system is zero.From the graph of amplitude ratio and clearance it is seen that the damper is effective at all speeds and clearances. However at low speeds, when vibration amplitude of the main system is small, there is no particular optimum clearance near and above resonance speed there is a region of optimum clearance where the amplitude ratio is minimum. This is observed at all mass ratios.However at low speeds and low mass ratio the damper is not much effective. In any case amplitude is small in this region and hence much damping is not needed.The damper is very effective at resonance condition. It is more effective over a large range of clearance. Hence the higher mass ratio looks to be advisable if practically possible. Twelve impacts per revolution are observed in most of the cases but they are not equally spaced. They are nearly equal near resonance. Impact duration is observed to be from 4 to 10 milliseconds but is subject to experimental errors as some sort of chattering is observed in the oscilloscope readings.

5.   CONCLUSIONS

As seen from the graphs, at resonance 12 impacts occur in one cycle of impact damper. One cycle means the theoretical time duration at a particular rpm for which the impact occurs. Below resonance 16 impacts occur in one cycle of impact damper. At low rpm, 20 impacts occur in one cycle of impact damper. Above resonance, 9 impacts occur in one cycle of impact damper. At high speeds, 4 impacts occur in one cycle of impact damper.

Impact duration at resonance is not more than 20 milliseconds. Above and below resonance impact duration does not exceed 25 milliseconds. At very high and very low speeds impacts do not occur. As the clearance increases impact duration decreases.

The clearance at which minimum amplitude is observed is called as optimum clearance. As the damper mass decreases, optimum clearance goes on increasingand optimum clearance is found to be at 0.32 mm for 492 grams damper,0.4 mm for 324 grams damper, 0.48 mm for 214 grams damper and 0.72 mm for 99.3 grams damper.

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