Manufacturing And Analysis of Vibration Transmission Test Rig

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Abstract- *Spring- mass- damper system is a predominant system in the Mechanical Vibrations domain. It is used to predict, study and analyse the displacement transmissibility from unsprung mass to sprung mass. There are theoretically derived formulae for displacement transmissibility. However there is no simplified practical setup available which adheres to the assumptions and components taken into consideration during theoretical derivations which can be used to feasibly verify, compare and observe the vibrational behavior.*

This project aims to design, manufacture and analyze a simplified vibration/displacement transmission test rig which will allow students to compare theoretically calculated values of displacement with the practical values. The design is simple as opposed to the actual road vehicle systems and closely resembles the theoretical model.

Keywords- Motor, Spring Mass, Frequency, Amplitude.

I. INTRODUCTION

1.1 INTRODUCTION

When a vehicle travels on the road it is subjected to numerous external forces.These forces cause displacement of single or multiple connecting bodies.One of this displacement is caused by irregularities in the road surfaces. For simplicity of analysis this irregularities are considered as rambling strips.

During the motion of the vehicle wheel acts as the base or support for the system and it moves vertically up and down.

At the same time there is relative motion between wheels and chassis. So chassis is having relative motion to the wheels and wheels are having relative motion to road surface.
So displacement is transmitted to wheels from the road surface. Some part of this displacement is absorbed by spring and dampers and rest is transmitted to mass. This phenomenon is called *Displacement Transmissibility.*

Displacement transmissibility can be predicted by theoretical derived formula as shown below:

$$\frac{A}{B} = \frac{\sqrt{1+(2\in r)^2}}{\sqrt{(1-r^2)^2+(2\in r)^2}}$$

Where A = amplitude of vibration of mass
B = amplitude of bumps
r = frequency ratio
Ɛ = damping ratio

A special purpose setup is built and the correlation between theoretical values and accrual values is done in this project.

1.2 IMPORTANT TERMS IN VIBRATION

1) **Frequency** : No of cycles per unit time
2) ***Amplitude*** : Maximum displacement of vibratory body from mean position
3) ***Natural frequency*** : when no external force acts on a system after giving it an initial displacement the body vibrates . These vibrations are called free vibrations and their frequency is called natural frequency.[6]
4) ***Damping*** : It is the resistance to the motion of vibrating body.

1.3 PARTS OF VIBRATORY SYSTEM

A vibratory system consists of basically three elements namely mass , spring and dampers. They exchange energy among themselves.

Energy is stored by mass in the form of kinetic energy , in spring in the form of potential energy and dissipates in damper in the form of heat energy which opposes the motion of system. Energy enters by external force called external excitation . The excitation is given to mass and entire system begins to vibrate.

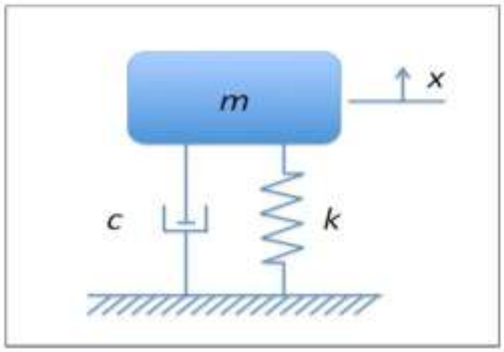

Fig1.1 vibrating system

The equation of motion of such a system can be written as

$mx'' + cx' + kx = 0$

where x''= acceleration, x'= velocity and x= displacement

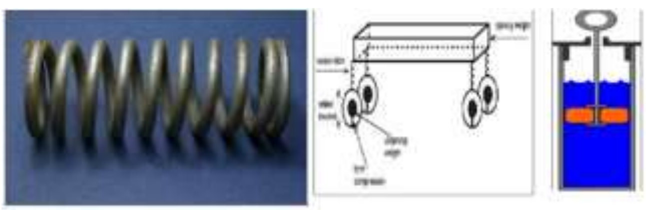

FIG: Spring , mass and damper

II. LITERATURE REVIEW:

Mr. Y.M Halde, Dr. J.T Pattiwar

In this paper the authors have developed a unique research and testing servo hydraulic complex designed to study various vehicle suspension units to determine their elastic damping characteristics and vibration isolation properties. The test program set the operating modes of suspension units at the leinematiz and force excitation using servo hydraulic equipment that is controlled by special software . This testing results in reduction of material cost as well as time and improvement of accuracy of the results

Prof. Fischer and Prof. S. Pellegrino

In this paper author has developed a new method for modelling the interaction between deployable structures and gravity compensation systems. It has been shown that the effects of adjustments to the suspension systems can be accurately predicted and that some improvements in the distribution of forces applied by the systems to the structure can be achieved by means of single adjustment of the suspension systems

Prof. Jordi Brunet

In this conference proceeding, the procedure to verify the dynamic behaviour of the suspension system on vehicle inspection was discovered .They found limit damping coefficient as a validation criteria below which the dynamic behaviour of vehicle demonstrates outstanding performance. Due to this computer model results have been confirmed by experimented test with enough accuracy**.**

Prof. Nikos. E. Mastorakis

In this paper the author presents the result on modeling and simulation of vehicle suspension testing .The results are in good concordance between experimental data and those provided by the mathematical model.

M. Gobbi , G Mastinu , M Pennati

In this paper the author presents the method for the indoor testing of road vehicle suspension systems. A car suspension is positioned on a rotating drum located in the laboratory for the safety of transport at the Politecnico di Milano and it is excited as the wheel passes over a cleat fixed on the drum. The wheel acceleration , displacement, and the forces acting at the suspension –chassis joints are measured in the frequency range of 0-120Hz . Five special six axis load cells have been designed and used. Transient wheel motions have been recorded

III. EXPERIMENTAL SET UP OF VIBRATION TRANSMISSION TEST RIG

3.1 IDEATION OF SETUP:

Before arriving at the finalized setup many different ideas were presented and analyzed. Many iterations of layout were carried out with thorough analysis of each stage including cost reduction possibilities and manufacturing feasibilities. Description of each stage is presented further in this chapter.

3.1.1 VERSION 1:

The setup consists of a conveyer belt placed at an elevated position by means of metallic structure. The belt is supported by two rollers, one powered and one non-powered. The powered roller will be given rotational motion by means of a motor.

The drawback of this setup is that the bumps are provided on conveyer and they should be flexible in structure in order to travel over rollers. However flexible bumps will not give effect of irregularities of road. So we discarded this version.

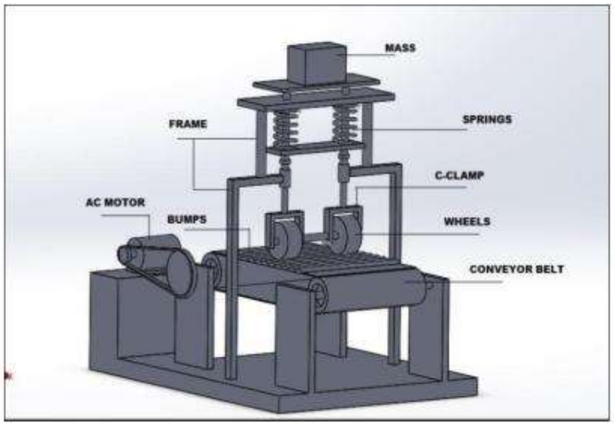

Fig 3.1 Setup version 1

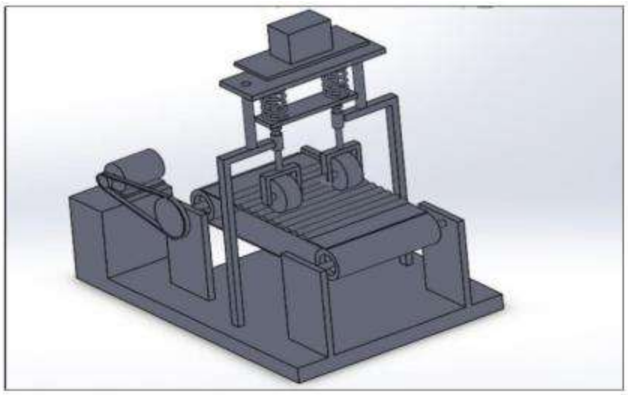

Fig 3.2 Setup Version 1 Oblique view

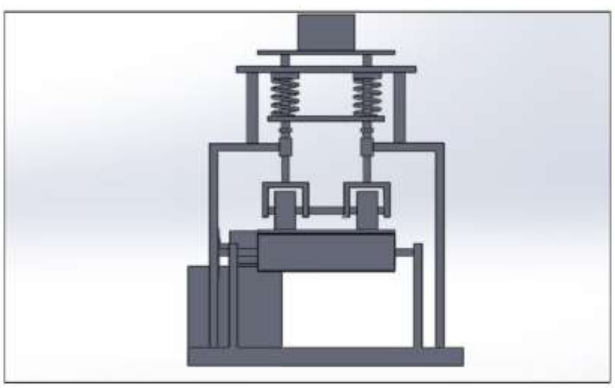

Fig 3.3 Setup Version 1 Front view

3.1.2 VERSION 2:

In order to overcome the drawbacks of first version the conveyer belt was replaced by a big cylindrical metallic drum. This idea was inspired by the paper Indoor testing of road vehicle suspensions by M. Gobbi , G Mastinu , M Pennati[5]

However the biggest drawback of this version is the rotating drum itself. It is very costly to roll a sheet of metal to required size which has to be done in professional rolling company. Hence, this layout was eliminated.

Fig 3.4 Wheel centre displacement measurement system [5]

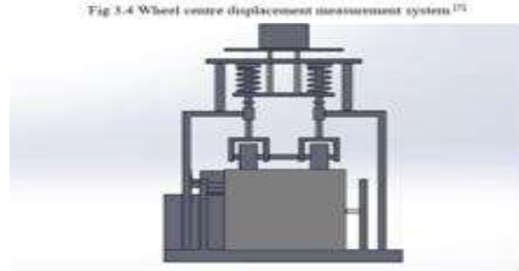

Fig 3.5 Setup Version 2 Front view

3.1.4 VERSION 3:

This version sees quite radical changes being made to the setup. The two wheels are totally separated as the setup gets divided into two separate columns of components working in conjunction. The big metallic drum is replaced by two flour mill pulleys. The flour mill pulleys are chosen for the fact that they have smooth surface at their circumference and have a pre-bored shaft hole and allen /hex bolt hole for attachment of shaft

The drawback of this setup is that it would be virtually impossible to practically synchronize the timing of bumps on both the pulleys. Also the setup will be large in size and will require large lab space.

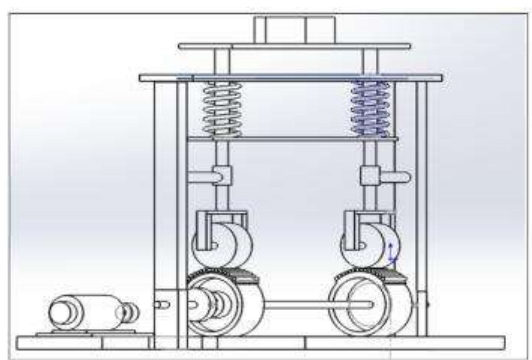

Fig 3.6 Setup Version 3 Oblique View

3.1.4 VERSION 4

This is the finalized version of the setup. This layout is virtually same as that of version 3 however for cutting costs and reducing dimensions and bulk, the version 3 setup is cut in half. This does not affect the stability of setup in any way.

Now the setup consists of a single floor mill pulley connected to motor via a speed reducer belt and pulley system. A single tyre of same or lesser diameter than that of pulley rests on top of it. Bumps are provided on the pulley. The tyre is supported by C clamp attached to a plate which is free to move inside the frame. This free moving plate carries a spring. Other end of this spring is attached to a plate fixed to the frame. Two long shafts starting at the free moving plate penetrate the fixed plate and end attached to a pan at the top. The pan will be used to carry mass. Displacement will be measured at the pan.

This setup avoids the problem of synchronizing the timing of bumps since there is only single set of components. Also the mass required at the top to compress the spring will be less.

FIG: Final setup

3.2 DESCRIPTION OF COMPONENTS

3.2.1 Flour Mill Pulley

The flour mill pulley was chosen because it has a very smooth surface without any undulations. It is made up of cast iron and hence is very stable under compressive load. The diameter of pulley is 12 inch or 304.8 mm. Also it had a pre-bored shaft hole of diameter 40 mm and allen/hex bolt hole for attachment of shaft.

FIG: Flour mill pulley

3.2.2 PULLEY SHAFT

A steel solid shaft of diameter 1.75 inch or 44.45 mm was chosen to be coupled to the flour mill pulley.

3.2.3 Water Pump

A Crompton Greaves single phase 1 HP 2780 RPM pump was chosen as a power source for rotational motion.

FIG: Water pump

3.2.3 MULTI-SLOT PULLEYS

Two multi slot pulleys were selected for speed reduction. The larger blue coloured pulley has constant diameter of 226.382mm (28 inch circumference) for all the three slots. The smaller silver coloured pulley has three steps of diameter- 84.893 mm, 60.638 mm and 37.394 mm. The different combinations of slots would theoretically give three different speeds of rotation- 1042.5 RPM, 744.643 RPM and 459.203 RPM.

FIG: Multislot pulley

3.2.4 V- BELT

A-57 Nylon V-belt was selected for transmission of power over the pulleys.

FIG: Belt

3.2.5 SPRING

Initially the shock absorber of Hero Splendor motorbike was chosen. However the stiffness of spring and damper setup was too high which meant that around 120-150 kg of mass would be required to compress the shock absorber which would be practically tough. Hence flour mill pulley of marginally lower stiffness (15.175 KN/m) was chosen

FIG : Spring

IV. EXPERIMENTAL PROCEDURE; OBSERVATIONS AND CALCULATIONS.

4.1 EXPERIMENTAL PROCEDURE

As shown above notedown the following things:

1) Note speed with which the pulley rotates with photo tachometer. From that we can calculate natural frequency.

2) "$A_{Practical}$ " amplitude of vibration of mass with the help of mechanical pointer

4.2 OBSERVATIONS AND CALAULATIONS

As per the procedure mentioned above experimentation was done. Six sets of readings were taken and the observations and calculations of the same are stated further in this sub-chapter.

Since the damping ratio for the setup in air is assumed to be zero, the formula for displacement transmissibility filters down to become

$$\frac{A}{B} = \frac{1}{\sqrt{(1-r^2)^2}}$$

The other formulae and constants are are

1). ω = $2\pi N$ /60 x 0.847
where
N= speed of rotation in RPM and 0.847 is the factor to compensate for the difference in diameter of pulley and tyre.
2). ωn =$\sqrt{k/m}$
Where
k= spring stiffness= 15.17 KN/m
3). r = ω /ωn
4.) . m= mass of plate + mass of pan + mass of weighs= 3.88+ mass of weighs

SETS OF READINGS:

1). N= 153.7
RPM ω= 13.632 rad/s
Apr= 17 mm
m= 12.23 kg
ωn= 35.2192 rad/s …(natural frequency)
r= 0.457
Ath= 17.64 mm

2). N= 672.7 RPM
ω= 59.669 rad/s
Apr= 6 mm
m= 12.23 kg
ωn= 35.2192 rad/s
r= 1.694
Ath= 8.02 mm

3). N= 659.2 RPM
ω= 58.4755 rad/s
Apr= 5.5 mm
m= 12.23 kg
ωn= 35.2192 rad/s
r= 1.66
Ath= 8.5 mm

4). N= 649.1 RPM
ω= 57.575 rad/s
Apr= 6 mm
m= 20.58 kg
ωn= 27.15 rad/s
r= 2.121
Ath= 4.3 mm

5). N= 659 RPM
ω= 58.455 rad/s
Apr= 3 mm
m= 20.58 kg
ωn= 27.15 rad/s
r= 2.153
Ath= 4.13 mm

6). N= 673.2 RPM
ω= 59.711 rad/s
Apr= 2 mm
m= 20.58 kg
ωn= 27.15 rad/s
r= 2.1993
Ath= 3.8 mm

4.3 RESULT

SR.NO	1	2	3	4	5	6
A/B	1.13	0.4	0.36	0.4	0.2	0.13
ω/ω_N	0.45	1.69	1.66	2.121	2.153	2.19

If we compare our results with standard graph then we can say that we have performed successful experimentation with some minute percentage of errors.

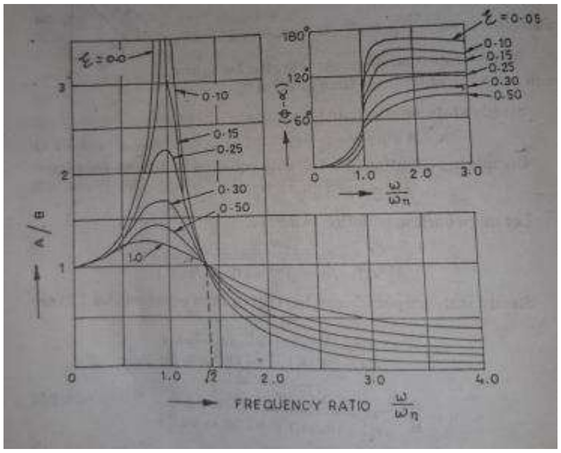

FIG : Standard Graph [6]

V. CONCLUSION AND FUTURE SCOPE

5.1 CONCLUSION

The objective of this project was to design and manufacture test rig for vibration transmission which adheres to the theoretical analysis. The corresponding literature survey was done and useful inputs such as components required for the test rig and method of studying were gained. Thus after multiple iterations and analysis, a final test rig design was finalised, the components required and their tentative dimensions were fixed and the rig was successfully fabricated. Following the fabrication, fine improvements of the setup were done and checking of theoretical formula was completed. Comparing practical and theoretical values of displacement transmitted it was found that there is a variation of 2- 3 mm between the two values in each run. The possible reasons for this are the errors induced due to friction between the two moving flat plates and the frame, the errors induced because of the measurement system (pencil attached to the top pan) and the errors due to transverse movement of plates in the frame.

5.2 FUTURE SCOPE

As stated in the previous sub- chapter there is a difference between practical and theoretical value. This error has the tendency to be minimized or even completely eliminated by adopting following methods in future-

1. Machining the flat plates with highest precision so that the transverse vibrations are virtually eliminated.
2. Adding rubber strips between the plates and the frame to minimize transverse movement and reduce noise during working
3. Displacement sensors can be used instead of mechanical measurement system for precise measurement of displacement.
4. Accelerometers can also be used to measure the displacement by measuring the acceleration and then differentiating it twice to achieve displacement.
5. The speed of motor in current setup is not under full control. A customized dimmerstat can be designed to perfectly suit the power of motor so that motor can be maintained at whatever speed that is desired.

While concluding this report, we fulfil in having completed the project assignment well on time. We had enormous practical experience on fulfilment of the manufacturing schedule of the working project model.

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