Vibration Reduction of Top-Load Washing Machine Based on Suspension System

Churnika N. Narkhede[1], Dr. K.K. Dhande[2]

***Abstract* - In fully automatic washing machine, laundry acts as a unbalance mass which causes vibration problems because in spin drying stage drum rotates at relatively high speed. This paper focus on the study related to design of a suspension system for the reduction of vibrations in washing machine. In finite element analysis, the modal analysis and harmonic analysis are performed to extract the mode shapes and the resonant condition. A mass spring damper system has been implemented to reduce vibrations and analytical calculations have been done for the same. The experimental analysis is carried out to measure the vibrations occurring in existing washing machine. For validation the analytical calculations have been compared with the experimental results. This has been done to use the same experimental setup when the experiment is conducted with the newly designed suspension system.**

Keywords—Vertical axis washing machine; Vibration; Finite element analysis.

I. INTRODUCTION

Reduction of noise and vibrations in high speed washing machines is an immense challenge in the present market scenario. Not only that but demand for energy saving is increasing rapidly in all industries and industrial applications and even in home appliances such as microwaves, refrigerators, air conditioners, power tools, vacuum cleaners and washing machines. A number of different parts which are used in a number of different ways can be found inside a machine. But the main goal is to have each part working correctly, cohesively and safely. If a machine is off balance or is vibrating more than it should, not only could it damage the parts inside it but also the floor that it is sitting on. Reducing machine vibration reduces the damages that can be seen in the machines and the surrounding area while maximizing the working efficiency.

Washing machines are important category of domestic machines used for automating manual tasks and therefore helping humans over a number of years. In a fully automatic washing machine, an unbalanced mass of clothes in a spin drum can cause vibration problems. During spinning cycle of washing machine, laundry placed in it acts as a rotating unbalance. In a spin drying stage, the drum spins at a relatively high speed which causes the clothes to be pressed against the inner wall of the drum and become a large unbalanced mass

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until the end of the stage. This transmits very large forces to the cabinet and radiates noise. This frequently occurs in the water extraction process when the drum starts to rotate giving rise to significant centrifugal imbalance forces and an imbalanced laundry mass rotation which results in vibration and shaking. By removing such vibrations, more silent washing machines can be designed for higher wash loads within the same housing dimensions. Therefore, the suspension parameters of the drum must be properly designed to limit the vibration transmissibility and increase the isolation efficiency.

This work focuses on the suspension of vertical axis drum-based washing machine which links the drum to the machine cabinet. The aim of the suspension is to reduce the vibrations transmitted from drum to the chassis which leads to the acoustic noise and damage of machine parts.

II. LITERATURE REVIEW

The vibration of washing machine can be controlled using two approaches. The first approach is based on control of the tub balance and second approach is based on control of suspension system. One method following the first approach, which is used to reduce vibration, is by use of a hydraulic balancer which contains salt water and it is attached to the upper end of the tub. Based on this technique, S.Bea, et al. [1] performed dynamic analysis of a vertical axis automatic washing machine during spin drying stage. They derived mathematical model of a hydraulic balancer in steady state condition which is validated by experimental result of centrifugal force. The results of experiments which were performed on washing machine during spin drying stage were compared with the simulation result. Vibration affecting parameters were investigated by parameter study. From the parameter study, they observed that the vibration can be reduced by increasing mass and decreasing volumetric ratio.

Another method of reducing vibration based on tub balance approach is by using two balancing masses. Evangelos

Papadopoulos, et al. [2] in their work explained the active balancing of drum using one and two balancing masses. In this method, the two balancing masses were attached along the periphery of the drum. These balancing masses move along the rim of the drum for which two actuators are required. They observed that the passive and active methods of stabilization are not mutually exclusive and therefore, the washer's spinning response could be improved by using them in parallel. But, the drawback of this technique was that it leads to complicated structure, high cost of manufacturing and maintenance which are major obstacles for a wide application of this method.

NOMENCLATURE		
M	:	Mass of the machine
m_o	:	Unbalance mass of clothes
C	:	Damping coefficient
K	:	Spring stiffness
X	:	Amplitude of vibrations
F_o	:	Centrifugal force
ω	:	Excitation frequency
ω_n	:	Natural frequency
ξ	:	Damping ratio
C_c	:	Critical damping coefficient
e	:	Eccentricity of mass from centre axis
N	:	Rotational speed, rpm
S_{ut}	:	Ultimate tensile strength
S_{yt}	:	Yield tensile strength
E	:	Young's modulus
N	:	Poisson's ratio
ρ	:	Density

Based on second approach which is control of suspension system, cristino spelta, et al. [3] proposed to reduce vibration by replacing passive dampers with magneto rheological dampers. They analyzed the dynamic behavior in case of different mounting conditions of MR damper. Two adaptive strategies were proposed, designed and tested. The experiments were performed in an anechoic chamber in order to study the effect of vibration control on the acoustic noise. Their study concluded that the electronically controlled MR dampers are more effective than the standard passive dampers.

Feng Tyan, et al. [4] developed a multibody dynamic model for front load washing machine. The bearing model between the tub and drum was verified by constructing this model. An analysis of the suspension system composed by two springs and MR dampers between case and basket was also conducted. The multibody model of front load washing machine with MR dampers was generated in commercial "Recurdyn" package. They concluded that PI control strategy is the best for reducing vibration of the basket and case at the same time.

Sundeep Kolhar and Dhiren Patel, et al. [5] explained the idea of the optimization of a washing machine with respect to reduction in drum vibration, power consumption and water consumption. For reducing the drum vibration they formulated a mathematical model. A modified drum design was proposed to further reduce the vibrations. Based on the values obtained from the mathematical model, the Finite Element Analysis (using Solid-Works Cosmos software) of the old and the new model is performed by them. They observed that the new model reduced the drum displacement to a considerable extent.

In a study, Sichani, et al. [6] explained vibration responses of a horizontal washing machine which they observed during run-up and run-down. They carried out number of impulse tests to compare and validate the results. The modes of the washing machine with operational tests were identified using both the EFDD and SSI methods. The natural frequencies, damping ratios and shapes were identified for modes of the body between 0 to 55 Hz in both methods. A comparison of the results of OMA with the classical modal testing (impact test with an instrumental hammer) was also conducted. Also, research has been done to find out how and where stabilization diagrams and stochastic subspace identification could be used. The false peaks and closely coupled modes were easily identified. The final conclusion of their research was that run-up/run-down can be used to identify the modes of a vibrating system in all cases except when the modes are present close to the working frequency of the system's rotating parts.

Multiple researches have been conducted on the design and application of MR dampers to control the vibration of front load washing machine. Referring to the above study, the main objective of the present study is to reduce vibrations in vertical axis drum based washing machines based on the control of suspension system using springs and dampers. The study initially analyzes the causes of vibration and studies existing suspension system of washing machine. Based on that configuration of springs and dampers are decided.

III. WASHING MACHINE

A washing machine model (Whirlpool WhiteMagic 123 650s) having 6.5 kg capacity (dry clothes) is selected for experimental setup. The specifications of machine are obtained from the user's manual of the machine as follows as given in table 1,

TABLE 1
SPECIFICATIONS OF MACHINE

Parameter	Label	Value	Unit
Speed	N	1400	RPM
Torque	T	20.46	Nm

Power	P	3	kW

Fig.1Washing Machine (WhiteMagic 123 650s)

IV. DRUM VIBRATION

Most common top loaded washing machines have 4 springs for suspension positioning as shown in fig.2,

Fig.2 Suspension System of existing Washing Machine

In this study to reduce vibration, suspension system was modified based on literature survey. Modified suspension system is used which consists of 4 springs and 4 dampers whose positions are as per given in figure 3.

The weight of wet clothes in the washing machine which acts as the unbalance weight causes an angular force which results into vibration. The springs and damper minimizes these vibrations. The above system can be mathematically modeled as one degree of freedom consisting spring mass damper facing rotational imbalance.

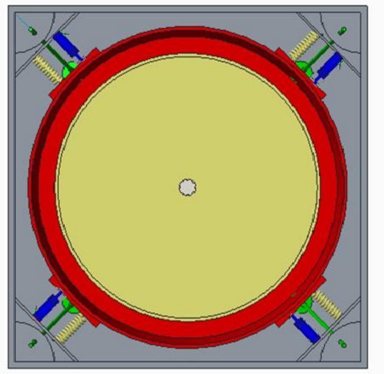

Fig. 3 Spring-damper arrangement for modified suspension system

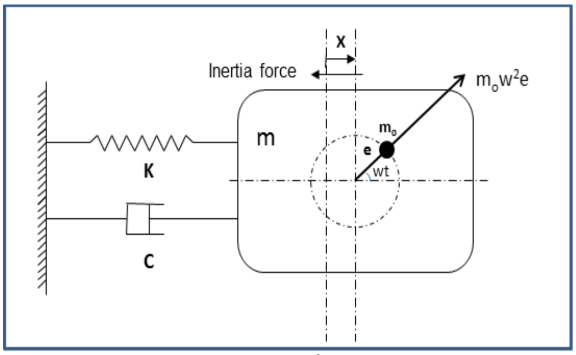

Fig.4 Free body diagram of washing machine drum

The amplitude of steady state vibration of above model is given as:

$$X = \frac{m_o e \omega^2 / K}{\sqrt{\left[1 - (\frac{\omega}{\omega_n})^2\right]^2 + [2\varepsilon \frac{\omega}{\omega_n}]^2}}$$

In order to obtain amplitude of vibration of washing machine having springs as suspension system, certain values are set as follows:

Mass of machine (M) = 20kg
Mass of wet clothing (m_o) = 9kg
Maximum rotational speed (N) = 750rpm
Spring stiffness, K = 1280N/m

After substituting above values, the amplitude of vibration that is obtained is 25.85mm.The amplitude of vibration can be reduced using spring – damper system by varying values of damping coefficient and spring stiffness. The values of damping coefficient and spring stiffness that were obtained are,

Spring stiffness, K= 2000 N/m
Damping coefficient, C = 35 Ns/m

After substituting above values, the amplitude of vibration that was obtained is 17mm. Based on the literature survey, these values were confirmed as the best suited values.

V. Finite Element Analysis

In FEA, modal analysis and harmonic analysis of the drum were performed. Different mode shapes of the drum were extracted using modal analysis. In harmonic analysis amplitude of vibration at resonance condition was carried out. The procedure of FEA is as follows:

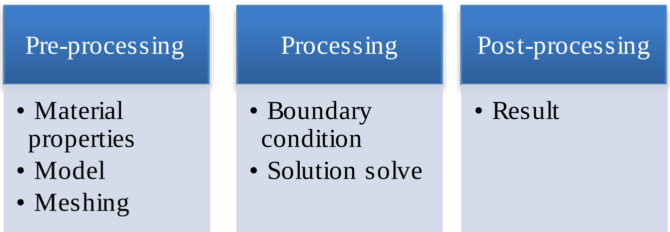

Fig.5 FEA procedure

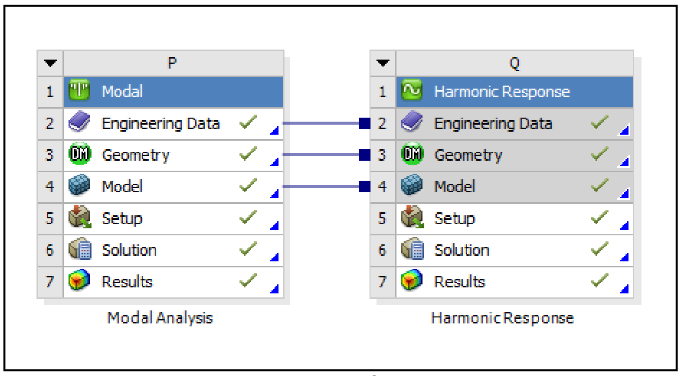

Fig.6 Project tree for analysis

The material of washing machine drum is stainless steel. Material properties are as shown in table 2.

TABLE 2
MECHANICAL PROPERTIES OF DRUM MATERIAL

Parameter	Label	Value	Unit
Material Grade		IS Standard 301	
Nominal Composition		Cr 18% Ni 8%	%
Young's modulus	E	204	GPa
Poisson's Ratio	υ	0.3	
Ultimate Tensile Strength	S_{ut}	610	MPa
Yield Tensile Strength	S_{yt}	240	MPa
Density	ρ	7800	Kg/m^3

The drum was modeled in ANSYS workbench. Input parameters for the drum are as per given in table 3.

TABLE 3
INPUT PARAMETERS FOR THE DRUM

Parameter	Label	Value	Unit
Mass of the drum	m_d	0.9761	kg
Volume of the drum	V	$9.76*10^6$	mm^3
Diameter of the drum	D_d	480	mm
Length of the drum	D_L	540	mm

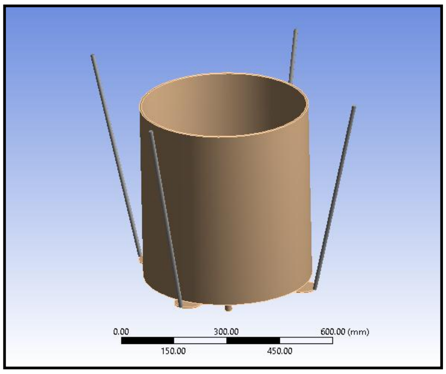

Fig.7 Washing machine Drum Assembly

Meshing of the drum is carried out using tetrahedral mesh with 20mm element size which gave us 17390 nodes and 8587 elements.

Boundary conditions used in FEA analysis are as follows.

Analysis settings for modal analysis are as follows:

No. of mode shapes to be extract = 10

Damping = No

For boundary condition, the four corners of washer's suspension support rod were fixed supported to the outer body.

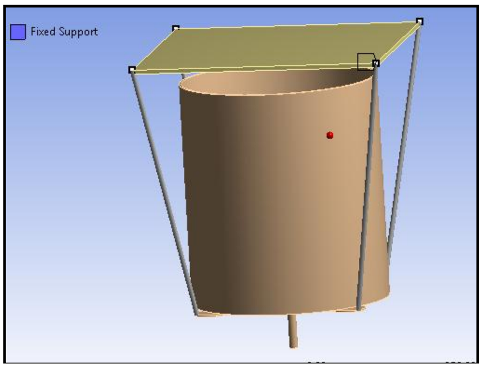

Fig.8 Boundary condition (Fixed Support)

A modal analysis is a technique used to determine the vibration characteristics of structures. In modal analysis we get the results in the form of natural frequencies and mode shapes. Natural frequency means the frequency at which the structure tend to oscillate without repeated external force and Mode shapes means a deformed body shape at each frequency.

The different natural frequencies obtained on different modes are,

TABLE 4
MODES SHAPE AND NATURAL FREQUENCY

Tabular Data

	Mode	Frequency [Hz]
1	1.	34.022
2	2.	34.022
3	3.	63.058
4	4.	605.66
5	5.	705.92
6	6.	769.39
7	7.	769.5
8	8.	777.72
9	9.	787.51
10	10.	1007.9

Modal analysis is performed to extract mode shapes for different natural frequencies. First frequency is the main natural frequencies. Total deformation for 1st mode shape for 34.002 Hz frequency is as shown in fig.9. Total deformation of 2nd mode shape and 3rd mode shape for 34.002 Hz and 63.058 Hz frequency is as shown in fig.10 and fig.11.

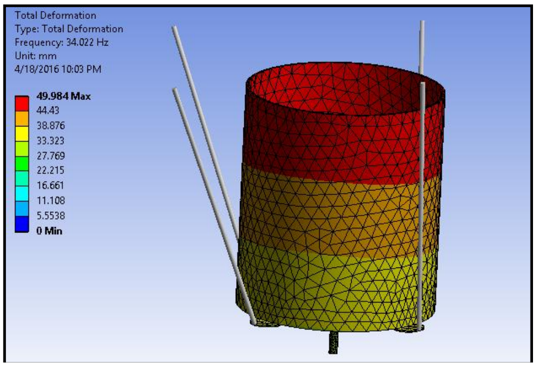

Fig.9 1st Mode shape

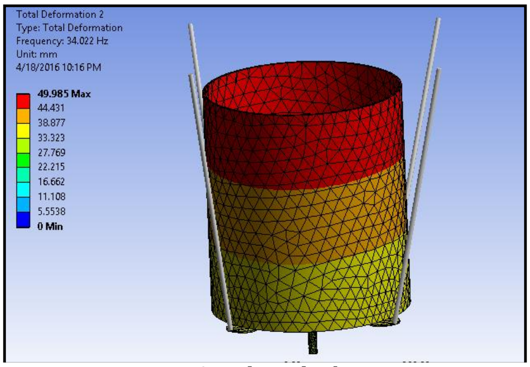

Fig.10 2nd Mode shape

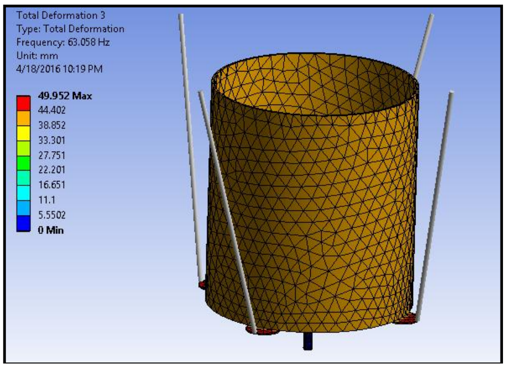

Fig.11 3rd Mode shape

Harmonic response is a technique to determine the steady state response of a structure to sinusoidal (harmonic) loads of known frequency. We have to give input as a harmonic load (forces, pressures, and imposed displacements) of known magnitude and frequency. Harmonic analysis is performed to detect resonant response and to avoid it if necessary.

In harmonic analysis, Analysis setting contains,

TABLE 5
ANALYSIS SETTING FOR HARMONIC ANALYSIS

Frequency Spacing	Linear
Range Minimum	0. Hz
Range Maximum	100. Hz
Solution Intervals	50

TABLE 6
BOUNDARY CONDITION FOR HARMONIC ANALYSIS

Type	Force
Define By	Components
Coordinate System	Global Coordinate System
X Component	0. N

Y Component	0. N
Z Component	1165N
X Phase Angle	0. °
Y Phase Angle	0. °
Z Phase Angle	0. °

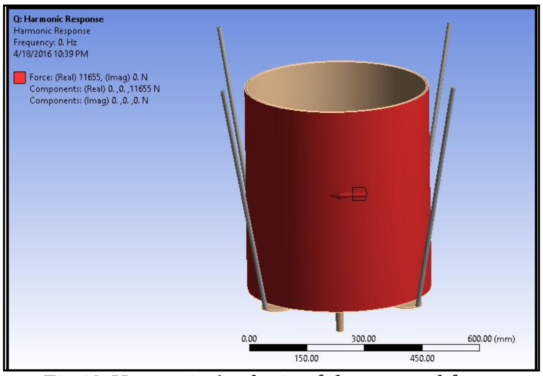
Fig.12 Harmonic Analysis of drum at real force

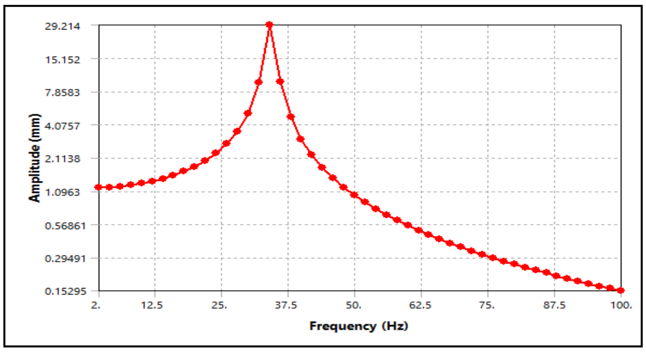

Fig.13 Frequency response plot of amplitude vs. frequency

From the harmonic analysis, it can be seen that resonance occurs at 37 Hz frequency for which amplitude of vibration is 29 mm.

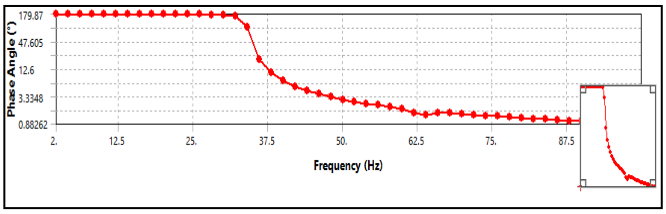

Fig.14 Frequency response plot of phase angle vs. Frequency

VI. Experimental Methodology

An operating test for the existing washing machine in spin drying stage was conducted in order to measure the vibration. The horizontal, cross and vertical displacement of the frame was measured.


Fig.15 FFT Analyzer


Fig.16 Accelerometer

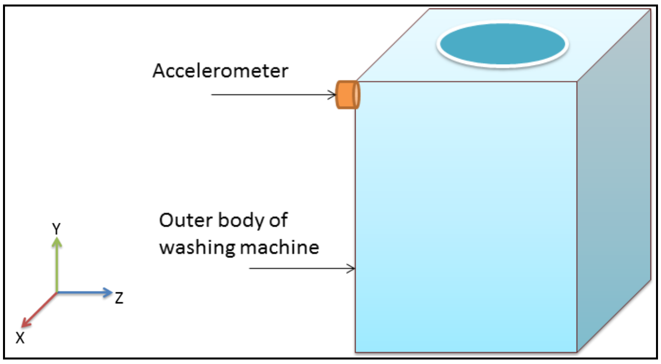

Fig.17 Schematic of vibration measurement setup

The measurements of the vibration were conducted when the rotation speed of the spin drum was from 60 rpm to 800 rpm. An accelerometer was attached to the right of the frame of washing machine. The measured displacement was displayed on a FFT analyzer.

The readings of the vibration of a machine were taken for load condition. The readings were taken for X, Y and Z direction. Where, X is lateral displacement, Y is vertical displacement and Z is horizontal displacement of steady state vibration.

VII. Result

The analytical and experimental investigation of drum of vertical axis washing machine has been conducted. Also, harmonic analysis of a drum has been performed in ANSYS software. The graphs are plotted in MS excel software based on the readings obtained.

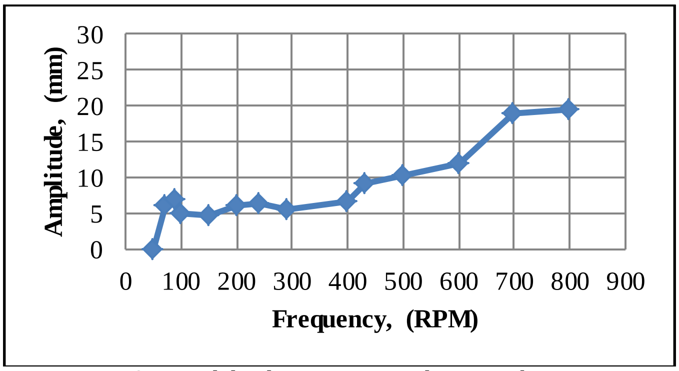

Fig.18 Lateral displacement in steady state vibration

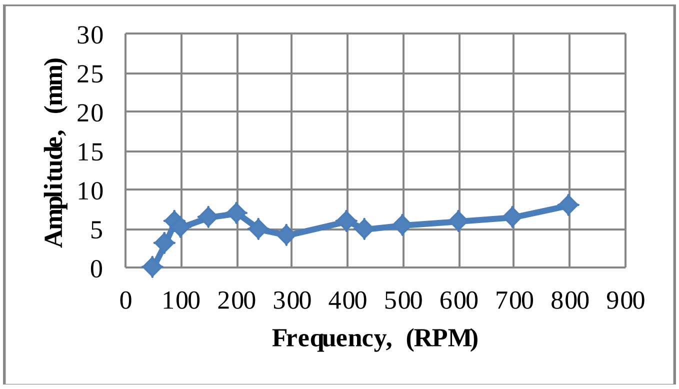

Fig.19 Vertical displacement in steady state vibration

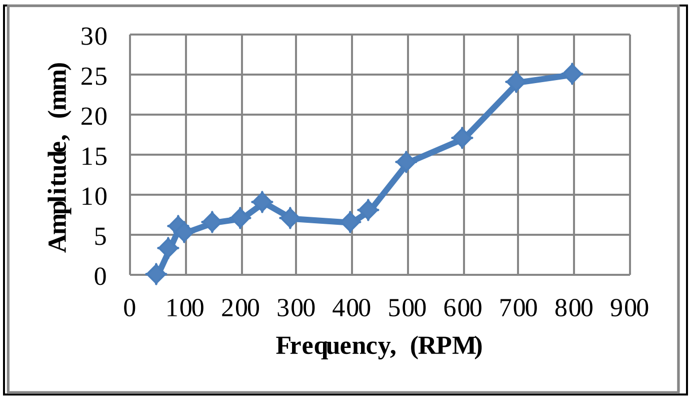

Fig.20 Horizontal displacement in steady state vibration

From the above graphs, the amplitude obtained in X direction is 20mm, in Y direction is 10mm and in Z direction is 25mm

Analytically obtained value of amplitude of vibration using spring – damper system is 17mm.

From the harmonic analysis, it is seen that the resonance will occur at 37Hz natural frequency with 29mm amplitude of vibration.

VIII. CONCLUSION

Based on the analytical and experimental results the following conclusions are drawn,

1. It is seen that, in existing washing machine the maximum amplitude occurs in Z direction, i.e. in horizontal direction which is 25mm.
2. The amplitude of vibration can be reduced using spring – damper system by varying values of spring stiffness and damping coefficient.
3. From analytical calculations it is seen that the amplitude of vibration can be reduced by 32%.

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