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Vibration Reduction of a Vertical Axis Drum Based Washing Machine

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***Abstract**: In a fully automatic washing machine, the laundry acts as an unbalanced mass which causes vibration problems because in the spin drying stage the drum rotates at a relatively higher speed. This paper focuses on the study related to design of a suspension system for the reduction of vibration in vertical axis drum based washing machine. A spring-damper system has been implemented to reduce vibrations and analytical calculations have been done for the same. In finite element analysis, the modal analysis and harmonic analysis are performed to extract the mode shapes and the resonant condition. The experimental analysis is carried out to measure the vibration occurring in washing machine with existing suspension system and in washing machine with newly designed suspension system.*

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

1. Introduction

The minimization of noise and vibrations in high speed washing machines is one of the biggest challenges that the industry is facing in the present market scenario. Not only that but energy saving has rapidly become a necessity 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. The main goal, however, is to have each part working correctly, cohesively and safely. If a machine is off balance or exhibits higher vibrations than it should, not only could it damage the parts inside it but also the floor that it is sitting on. This damage will in turn lead to a slowdown in production. Reducing machine vibration reduces the damages that can be seen in the machines and the surrounding area while maximizing the working efficiency.

Washing machines, the first of which was developed over 150 years ago, are an important category of domestic machines which are used for automating the manual effort that goes into washing laundry. Besides basic functions such as washing and spin drying, low vibration characteristic is becoming an important performance index of washing machines.

In a fully automatic washing machine, an unbalanced mass of clothes in a spinning drum can cause vibration problems. During the spinning cycle of a washing machine, the laundry placed in it acts as a rotating unbalance. In this spin drying stage, the drum spins at a relatively higher speed which causes the clothes to be pressed against the inner wall of the drum and become a large unbalanced mass until the end of the stage. The amount of the unbalance mass is dependent on both the weight of the laundry and the condition of the washing mode. This transmits very large forces to the cabinet and radiates noise. 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.

Current environmental awareness demands the improvement of washer efficiency. Talking about efficiency, the first thought that comes to mind is optimization. The basic factors that lead to optimization in a washing machine are its washing capacity, power consumption, cost and vibration. The industrial drive technology can be classified into two different groups. One group includes electrically driven machines requiring speed control systems for different applications for example, machine tools and measuring machines for which precision in movement is required. Second group includes consumer electrical systems, for example pumps, washing machines, food processors, vacuum cleaners and fans where precision torque or speed control systems are not needed. Common disadvantages of these systems are poor efficiency and distortion. High efficiency, reduced noise, extended lifetime, rapid time to market at optimum cost are the challenges faced by many industries.

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.

2. Literature Review

The vibration of a washing machine can be controlled by using one of two approaches - the first is based on control of the tub balance and the second is based on control of the suspension system. One method to reduce vibrations, which follows the first approach, uses a hydraulic balancer which contains salt water and is attached to the upper end of the tub.

S. Bae, et al. [1] (2002) performed dynamic analysis of a vertical axis automatic washing machine during the spin drying stage. They derived the 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.

Evangelos Papadopoulos, et al. [2] (2001) 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.

CristinoSpelta, et al. [3] (2008) 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] (2009) 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.

SundeepKolhar and DhirenPatel, et al. [5] (2013) 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.

Sichani, et al. [6] (2007) 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.

A.K.Ghorbani-Tanha et al. [7] (2009) described how the Operation of home appliances like washing machines can produce undesirable vibrations and noise and the purpose of their study was to analyze and develop a control system for vibration reduction of washing machines employing smart materials.

Shao Rui Ying et al. [8] (2014) studied and analyzed the dynamics characteristics of drum washing machine vibration isolation system based on the theory research and virtual prototype technology. According to the Lagrange method, the dynamics equations and motion differential equations of drum washing machine vibration isolation system was established. Through the establishment of rigid parameterized virtual prototype model of the vibration system, dynamics simulation analysis was accomplished based on ADAMS, the kinematics characteristics and mechanical characteristics were obtained.

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.

3. Drum Vibration

Most common top loaded washing machines have 4 springs for suspension positioning as shown in Fig. 1.

Figure 1 Suspension System of an existing Washing Machine

The vibration generated in washing machine reduces efficiency and increases structural damage of washing machine. This study has been performedin order to reducemore vibration occurring in washing machine due to unbalance mass. The literature study reveals that passive or active dampers can be used to reduce vibration in washing machine. Based on literature study, it is proposed to use hydraulic dampers and springs attached between outer tub and cabinet. The arrangement that reduces vibration effectively is as shown in figure 2.

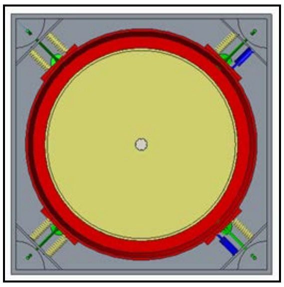

Figure.2 Spring-damper arrangement for modified suspension system

The above system can be mathematically modeled as one degree of freedom consisting spring mass damper facing rotational imbalance.

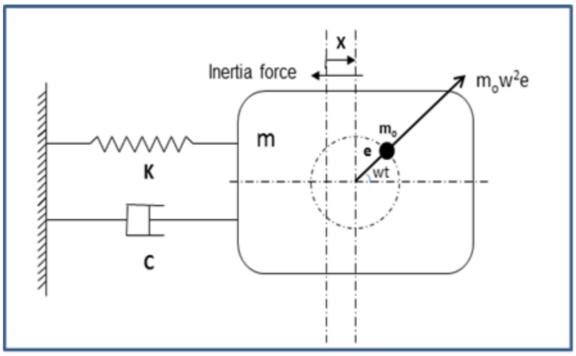

Figure. 3 Free body diagram of washing machine drum

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

$$X = \frac{m_0\, e\, \omega^2/k}{\sqrt{\left[1-\left(\frac{\omega}{\omega_n}\right)^2\right]^2+\left[2\xi\frac{\omega}{\omega_n}\right]^2}} \quad \text{------- (1)}$$

In order to obtain amplitude of vibration of washing machine with existing suspension system, certain values are set as given in table 1.

Table 1. Input parameters for amplitude of existing suspension system.

Parameter	Value	Unit
Excitation frequency (ω)	78.53	rad/s
Natural frequency (ωn)	24.49	rad/s
Damping ratio (ξ)	0	-
Stiffness of spring (K)	1.280	N/mm
Force (Fo)	2900	N

Hence, by formula (1)

$$X = \frac{m_0 e \omega^2/k}{\sqrt{\left[1-\left(\frac{\omega}{\omega_n}\right)^2\right]^2+\left[2\xi\frac{\omega}{\omega_n}\right]^2}}$$

$$X = \frac{2900/5.120}{\sqrt{\left[1-\left(\frac{78.53}{24.49}\right)^2\right]^2+\left[2*0*\frac{78.53}{24.49}\right]^2}}$$

= 25.85 mm

The obtained value of amplitude of vibration of existing suspension system is 25.85mm.The amplitude of vibration can be reduced using spring – damper system by varying values of damping coefficient and spring stiffness. In order to obtain amplitude of vibration of washing machine with new suspension system, certain values are set as given in table 2.

Table 2. Input parameters for amplitude of vibration.

Parameter	Value	Unit
Excitation frequency (ω)	78.53	rad/s
Natural frequency (ωn)	24.49	rad/s
Damping ratio (ξ)	0.173	-
Stiffness of spring (K)	2.000	N/mm
Force (Fo)	2000	N

Hence, by formula (1)

$$X = \frac{Fo/k}{\sqrt{\left[1-\left(\frac{\omega}{\omega n}\right)^2\right]^2+\left[2\xi\frac{\omega}{\omega n}\right]^2}}$$

$$X = \frac{2000/12000}{\sqrt{\left[1-\left(\frac{78.53}{24.49}\right)^2\right]^2+\left[2*0.07144\frac{78.53}{24.49}\right]^2}}$$

= 17.83 mm

The obtained value of amplitude of vibration is 17.83mm. Based on the literature survey, these values were confirmed as the best suited values.

4. 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:

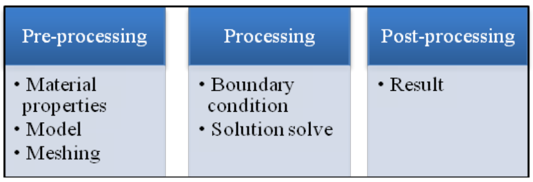

Figure 4. FEA procedure

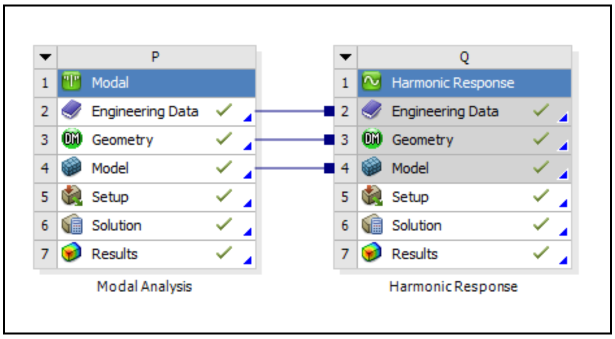

Figure 5. Project tree for analysis

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

Table 3. 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 4. 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*106	mm^3
Diameter of the drum	D_d	480	mm
Length of the drum	D_L	540	mm

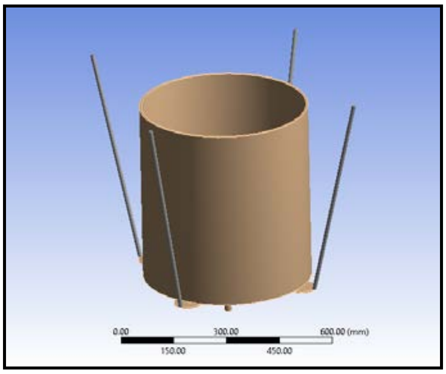

Figure 6. Washing Machine Drum Assembly

Another significant factor that will affect the possibilities to obtain acceptable results from the analysis is how the mesh is defined. A finer mesh will generate more accurate results, at the price of longer calculation time.

Table 5. Specification of meshing.

Mesh Used	Tetrahedral mesh
Element size	20 mm
Total nodes	17390
Total elements	8587

Boundary conditions used in FEA analysis are as follows.

Table 6. Analysis settings for modal analysis.

No. of mode shapes to be extracted	10	Linear
Damping	No	0. Hz

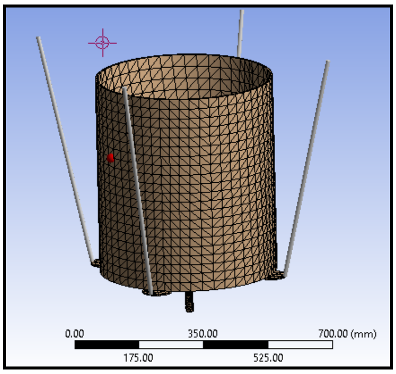

Figure 7. Meshing of washing machine drum

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

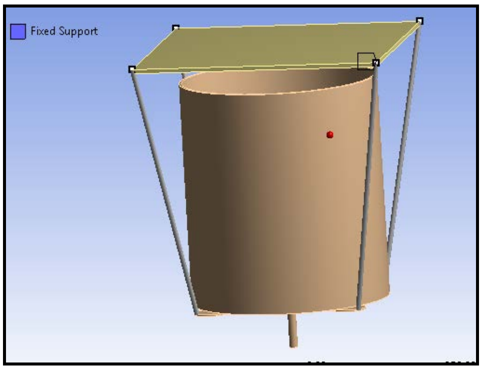

Figure 8. Boundary condition (Fixed Support)

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 7. Mode 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. 10. Total deformation of 2nd mode shape and 3rd mode shape for 34.002 Hz and 63.058 Hz frequency is as shown in fig.11 and fig.12.

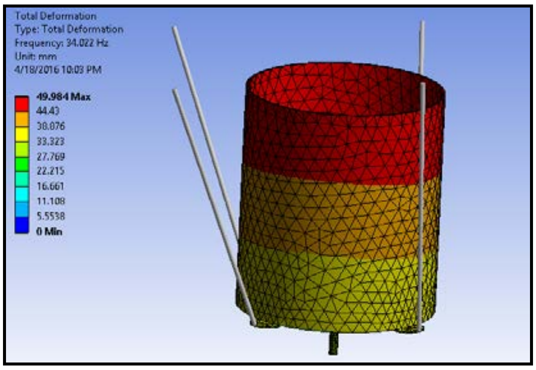

Figure 9. 1st Mode shape

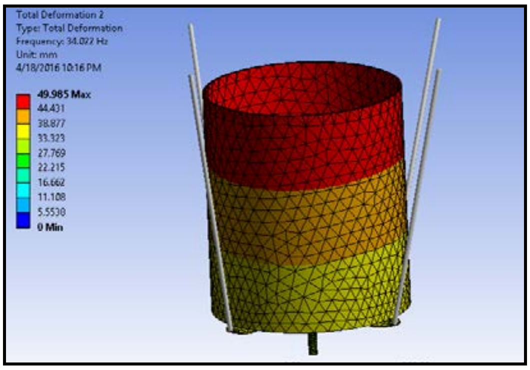

Figure 10. 2nd Mode shape

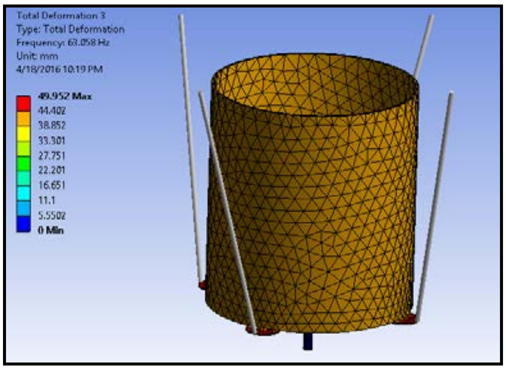

Figure 11. 3rd Mode shape

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 8. Analysis setting for harmonic analysis

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

Table 9. 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	11165N
X Phase Angle	0. °
Y Phase Angle	0. °
Z Phase Angle	0. °

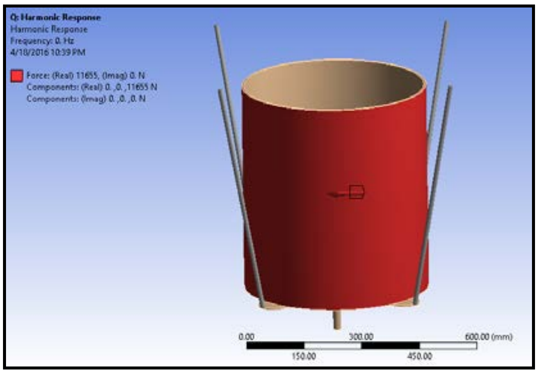

Figure 12. Harmonic Analysis of drum at real force

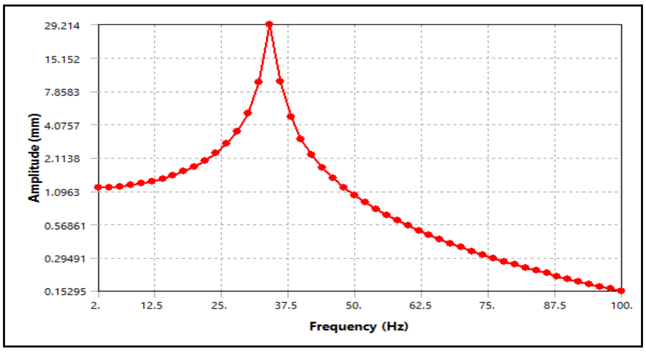

Figure 13. Frequency response plot of amplitude vs. frequency

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

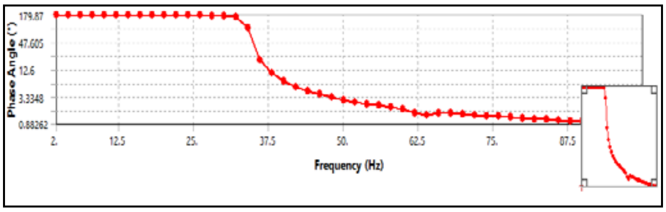

Figure 14. Frequency response plot of phase angle vs. Frequency

5. Experimental setup

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 10,

Table 10. Specifications of machine.

Parameter	Label	Value	Unit
Speed	N	1400	RPM
Torque	T	20.46	Nm
Power	P	3	kW

Figure 15. Washing Machine (White Magic 123 650s)

The dimension of this machine is 540*540*880 mm, weighs 33 kg along with digital display and control panel to set the settings. It has agitator with scrub pad wash system, 8 wash programs and 740 RPM is the spinning speed. The input power of the motor is 360 Watts.

An operating test for the existing washing machine and washing machine with modified suspension system in spin drying stage are conducted in order to measure the vibration. The measurement device used to measure vibration is vibration meter VM-6360.

Figure 16. Vibration Meter VM - 6360

The procedure for taking readings is as follows:
1) Attach an accelerometer to the right side of the frame of washing machine.
2) Another end of accelerometer is attached to the vibration meter VM-6360.
3) The measurements of the vibration are conducted when the rotation speed of the spin drum was from 60 rpm to 800 rpm.
4) Each time the Function key is depressed and released quickly, the meter will step to the next vibration measurement parameter with the corresponding unit showing on the display.
5) The measured value of displacement, acceleration, frequency was displayed on vibration meter.

Figure 17. Experimental Setup

6. Results

For the experimental results, the acceleration, displacement and frequency of the system have been measured using a vibration meter. These readings have been taken both before and after implementation of the newly designed suspension system. Post completion of the measurements, a thorough comparison has been conducted between two sets of readings. Using these readings, the frequency vs. acceleration and frequency vs. displacement graphs have been plotted for both the conditions (before and after the implementation of new suspension system) as shown in the below figure.

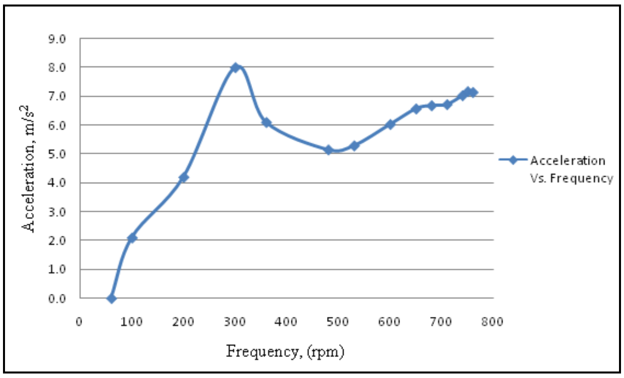

Figure 18. Frequency Vs Acceleration plot for machine with existing suspension system

Figure 18 shows the graph of frequency vs. acceleration of a machine with existing suspension system. From this graph it is concluded that the acceleration is high at 300rpm (when the spin cycle has just begun) but lasts only for two seconds and then varies between 5-8 m/s2for RPM ranging between 500rpm to 800rpm.

Figure 19 shows the graph of frequency vs. displacement. It shows that the displacement is 24mm when the drum rotating at 750 rpm which is the maximum speed that the machine attains.

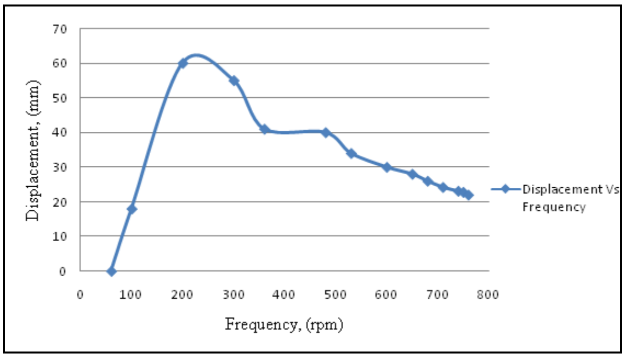

Figure 19. Frequency Vs Displacement plot for machine with existing suspension system

Figure 20 shows the graph of frequency vs. acceleration of a machine with new suspension system. From this graph we can conclude that the acceleration is high at 300rpm (when the spin cycle has just begun) but lasts only for two seconds and then varies between 5-6 m/s2 For RPM ranging between 500rpm to 800rpm.

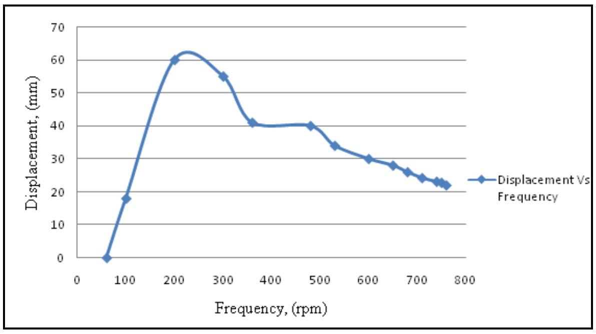

Figure 20. Frequency Vs Acceleration plot for machine with new suspension system

Figure 21 shows the graph of frequency vs. displacement. It shows that the displacement is 19mm when the drum rotating at 750 rpm which is the maximum speed that the machine attains

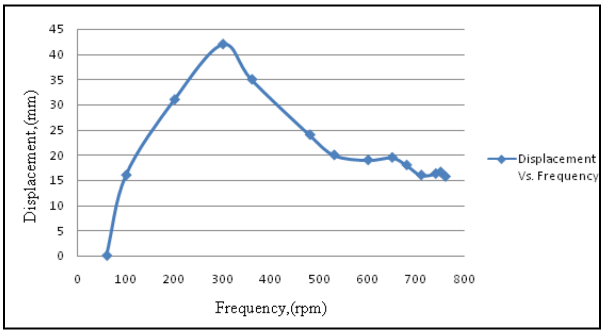

Figure 21. Frequency Vs Displacement plot for machine with new suspension system

7. Conclusion

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

1. Analytically, in the existing system the amplitude of vibrations (in a horizontal direction) is 25mm and 17mm in the newly designed suspension system. Hence an overall 32% reduction in vibration is predicted.
2. Through the FEA analysis, we have concluded that the resonance condition will never be satisfied since the natural frequency of the system is 37 Hz and external frequency measured is 12.5 Hz. Even with minor fluctuation in the external frequency, the difference is quiet large and hence resonance will never occur.
3. Experimentally we have measured a 25% overall reduction in vibration. In the existing system, the amplitude of vibration is gauged to be 22mm which reduces to 18mm after implementation of the new suspension system.

8. Acknowledgement

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