Experimental Investigation of the Effect of Magneto-Rheological MR Damper on a Rotating Unbalance SDOF System

A. A. A. Zayed[1] , Samy F. M. Assal[3] , A. M. Khourshid[4]
[1, 3, 4] Production Engineering and Mechanical Design Department,
Faculty of Engineering,
Tanta University, Tanta,
Egypt

E. Saber[2]
[2] Mechanical Engineering Department,
Faculty of Engineering and Technology,
Arab Academy for Science and Technology,
Alexandria, Egypt

Abstract— In this paper the effect of using MR damper on the dynamic response of a single degree of freedom (SDOF) system with rotating unbalance masses is experimentally investigated. The Bouc-Wen model of MR damper is adopted to characterize its performance. The theoretical analysis of the passive un-damped system is introduced and simulations are conducted. A test rig of rotating unbalance system achieved by two meshed gears carrying unbalance masses is constructed. The experimental results of the un-damped and MR damped systems are presented for comparison. The results prove that the MR damper can effectively suppress vibration amplitude even at resonance.

Keywords—Vibration suppression; MR damper; rotating unbalance; MR fluids

I. INTRODUCTION

Rotating unbalance in machines leads to the presence of vibrations which cause several harms such as producing unpleasant stresses in machine parts, rapid wear in machine parts such as bearings and gears and decreasing accuracy of surface finish for cutting machines. It is necessary to develop suitable methods that decrease vibration amplitude and increase stability of machines. Vibration isolation systems can be classified as passive, semi-active and active suspension systems. Passive systems are based mainly on springs and dampers with constant damping coefficient. These systems have been used in wide applications because of their simplicity and easy operation. However, such systems exhibit an intrinsic limitation, that is, it is effective only in a certain frequency range [1]. Hydraulic actuator is an example of an active suspension system [2]. However, these systems require a high power demand. Semi-active systems are based on rheological dampers. These dampers are categorized as electro-rheological (ER) and magneto-rheological (MR) dampers. The ability to vary the damping coefficient has prompted many researchers to explore the possibility of improving the suspension system performance by using semi-active dampers [3]. As MR dampers need low power requirements than ER dampers and can give high values of damping force, MR damper is commonly used to represent a semi-active suspension system. MR damper resembles the conventional viscous damper but filled with MR fluid with base oil and different percentages of ferrous particles. When this fluid is exposed to magnetic field, the particles align in a colloidal form parallel to the lines of the magnetic flux. MR damper is an effective semi-active vibration actuator which can produce controllable damping force by supplying proper electrical current or voltage to MR fluid to change its rheological behavior from free flow – linear viscous to semi-solid [4]. This change is very quick, less than 25 milliseconds.

A lot of mathematical models are developed for modeling non-linear mechanical behavior of MR damper and can be generally grouped as parametric models and non-parametric models [5]. Parametric models such as Bingham model [6], Gamota and filisiko model, Bouc-Wen model, modified Bouc-Wen model [7], hyperbolic tangent model, Dahl model and many other models. These models are based on mechanical elements represented by springs, viscous, friction, etc. It requires only displacement and voltage as inputs. On contrary, non-parametric models such as neural-network models, fuzzy models [8], and neuro-fuzzy models [9] learn from experimental data and require more inputs/outputs data than parametric models.

Different studies on MR damper have been implemented in the literature to suppress vibration amplitudes. The Bouc-Wen model parameters were identified for MR fluid damper using adaptive changed system optimization in [10]. A genetic algorithm (GA) has been proposed for parameter identification of Bouc-Wen model of MR fluid damper in [11]. In [12] MR damper was used for turning tool holder. The analysis was made by ANSYS and the results showed that the use of MR damper reduces tool vibrations effectively. The effect of metal cutting input parameters such as spindle speed, feed rate and axial depth of cut on metal cutting forces was discussed in [13]. It was observed that the metal cutting forces are less with MR damper than without. The vibration of a seismically excited building structure equipped with an MR damper was reduced under a non-linear control based on a linear matrix inequality as presented in [14]. A semi-active skyhook controller was implemented in [15] for a half train. Simulation results showed that with the semi-active control the vibration of the suspension system is well controlled.

In this paper a semi-active suspension system employing MR damper experimentally implemented to investigate the effect of MR damper on the vibration amplitude of a SDOF system having rotating unbalance. The performance of MR damper in introduced in section 2, while the model of a SDOF system is introduced in section 3. The experimental test rig is constructed in section 4 and the results are discussed in section 5. Finally the results conclusions are extracted in section 6.

II. PERFORMANCE OF MR DAMPER

The modified Bouc-Wen model [7], is adopted to characterize the performance of the MR damper. The structure of this model is shown in Figure 1.

The mathematical model is given as follows:

$$F = \alpha z + c_0(\dot{x} - \dot{y}) + k_0(x - y) + k_1(x - x_0) \quad (1)$$

$$F = c_1 \dot{y} + k_1(x - x_0) \quad (2)$$

where

$$z = -\gamma|\dot{x} - \dot{y}||z||z|^{n-1} - \beta(\dot{x} - \dot{y})|z|^n + A(\dot{x} - \dot{y}) \quad (3)$$

$$\dot{y} = \frac{1}{c_0 + c_1}\left[\alpha z + c_0 \dot{x} + k_0(x - y)\right] \quad (4)$$

where F is the total force generated by this model, and the parameters γ, β, A, k1 and n are fixed parameters. By adjusting the hysteresis parameters γ, β and A the linearity in the unloading and the smoothness of the transition from the pre-yield to the post yield region can be controlled. The parameters α, c_0 and c_1 are assumed to be functions of the applied voltage.

$$\alpha = \alpha_0 + \alpha_b u \quad (5)$$

$$c_0 = c_{0a} + c_{0b} u \quad (6)$$

$$c_1 = c_{1a} + c_{1b} u \quad (7)$$

$$\dot{u} = -\eta(v - u) \quad (8)$$

where c_1 is the viscous damping at higher velocities , k_1 is the stiffness representing the accumulator, k_0 is the stiffness at higher velocities , x_o is the initial deflection of the accumulator gas spring , η is the time constant , α is the Bouc- Wen parameter describing the MR fluid yield stress , and υ is the command voltage sent to the current driver.

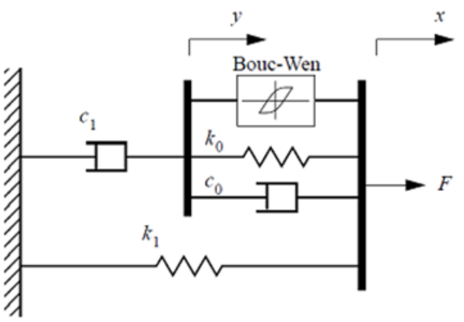


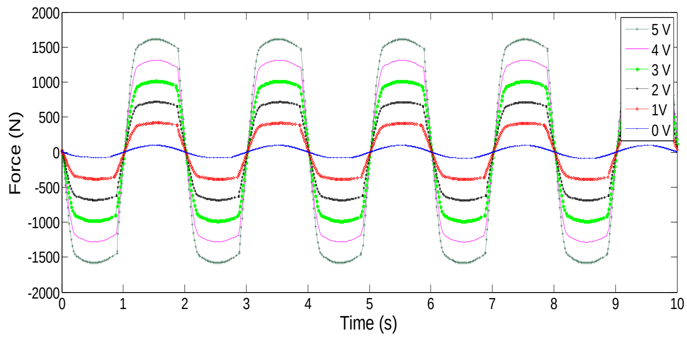
Fig. 1. Modified Bouc-Wen model [7]

The MR damper used in this study is the RD 8041-1 (long stroke) from Lord Corporation [16]. The parameters values of this damper are shown in table I[17].

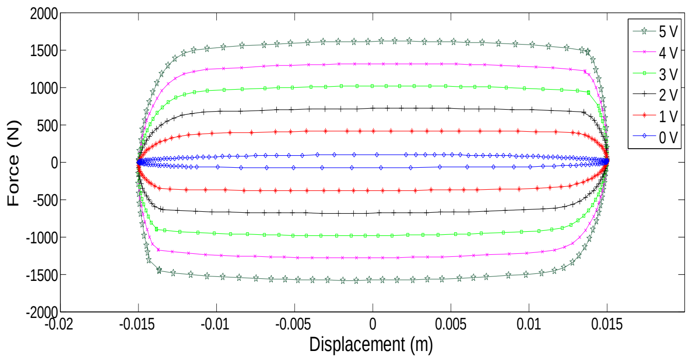
TABLE I Identified parameters for the modified Bouc-Wen model

parameter	Value	Unit
α_a	1921.141	N/m
α_b	5882.51	N/m-V
C_{0a}	651.4718	N-s/m
C_{0b}	1043.7559	N-s/m-V
C_{1a}	2089.263	N-s/m
C_{1b}	14384.918	N-s/m-V
K_0	1940.405	N/m
K_1	1.751268	N/m
γ	36332.07	m^{-2}
β	36332.07	m^{-2}
A	155.32	-
X_0	0.00	m
n	2	-
η	60	S^{-1}

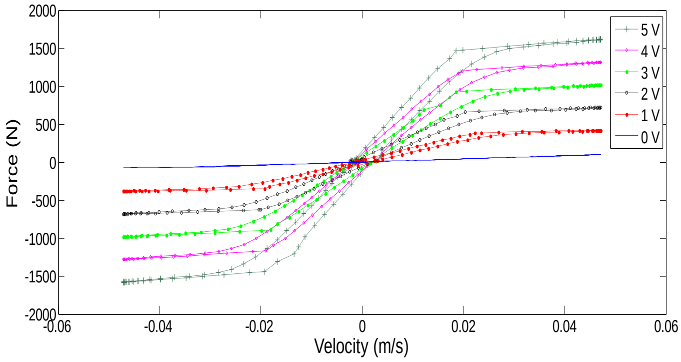
The MATLAB/SIMULINK software is used to simulate the damper model. The results in terms of force - time, force - displacement and force - velocity relationships under 0.5Hz sinusoidal excitation with amplitude of 0.015m for different voltages of 0, 1, 2, 3, 4 and 5 volts are shown in Figures 2a, 2b and 2c respectively.



(a) Force – time relationship



(b) Force – displacement relationship



(c) Force – velocity relationship

Fig. 2. RD-8041-1 MR damper characteristics

It is clear that with the increase of the applied volt, the damping force increases remarkably.

III. Model of SDOF System

The dynamic model of a SDOF passive suspension system having rotating unbalance as shown in Figure 3a is given as:

$$M\ddot{x}+C\dot{x}+Kx = f_o \sin\omega t \tag{9}$$

The steady state response is:

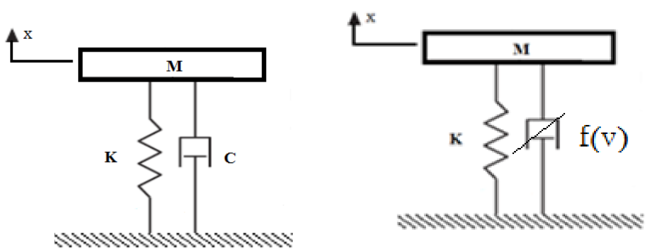
$$X = \frac{\frac{me}{M}r^2}{\sqrt{(1-r^2)^2+(2\xi r)^2}} \tag{10}$$

where $f_o = me\omega^2$

$$r = \left(\frac{\omega}{\omega_n}\right)$$

where r is the frequency ratio, ξ is the damping ratio, me unbalance in the system, f_o is the amplitude of the exciting force, ω_n is the natural frequency of the system, ω is the operating frequency, M, K and C are the system mass, spring stiffness and viscous damping coefficient respectively.

The equation of motion of the semi-active suspension system shown in Figure 3b is the same as (9) except that the term of the damping force $C\dot{x}$ is replaced by the MR damping force F calculated from (1).



(a) (b)

Fig. 3. SDOF suspension system (a) passive (b) semi-active

IV. Experimental Test Rig and Excitation Unit

In order to investigate the effectiveness of the MR damper on a SDOF system having rotating unbalance an experimental test rig is constructed as shown in Figure 4.

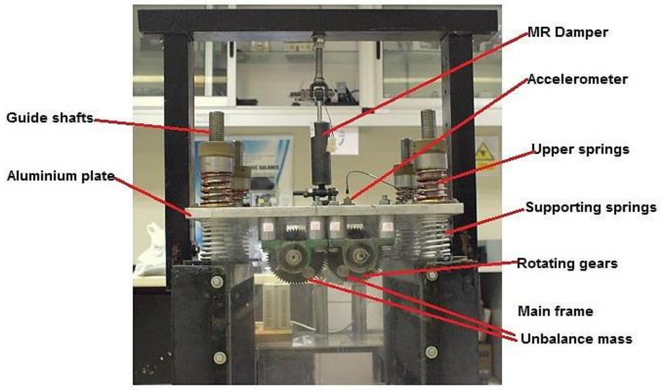


Fig. 4. Experimental test rig

The mass M is made of aluminum square plate with dimensions 500×500×20 mm. A set of four identical springs supports M and another set of four identical springs set above M locked with cupper sleeves are mounted on guiding shafts fixed to a heavy steel structure. The exciting force is induced from the rotation of two identical meshed spur gears. Each gear carries a steel rotating unbalance mass m of 214 g at a radius of 42 mm from the gear center. The two unbalance masses are mounted opposing each other on the two gears; in order to induce vibrations in the vertical direction only and cancel each other in the horizontal direction. The resultant vibration is in the form of ($X\sin\omega t$). The two gears are mounted on steel shafts attached to the lower surface of the aluminum plate via four ball bearings. The gear assembly is driven through a flexible shaft driven by a single phase electrical motor of power 0.3675 KW. A variable frequency inverter model SV008iC5 LG, is used to vary the motor revolutions per minute. The main concern range of frequency in this study is from 3 to 30 Hz.

The data is gathered using a high sensitivity tri-axial accelerometer type 4506B from BRUEL & KJAER [18]. This accelerometer is attached to BRUEL & KJAER hardware that processes the incoming signals and transfers it to the software on a PC via Local Area Network link (LAN). A power supply is used to provide the MR damper with the input voltage .The block diagram of the system is shown in Figure 5.

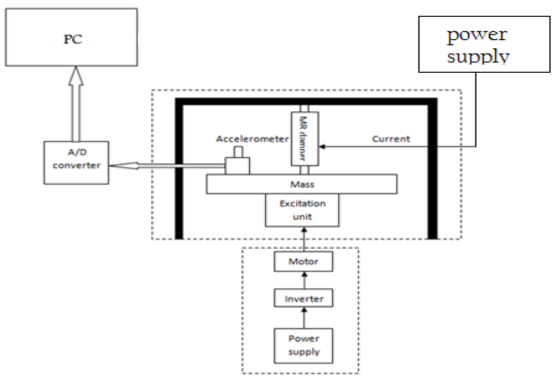


Fig. 5. Block diagram of a SDOF semi-active system

The values of the parameters used in this work are K=55666 N/m, M= 30.26, 34.26 and 38.26 Kg, m=214g and e=42 mm.

V. RESULTS AND DISCUSSION

Experiments with and without the MR damper were conducted to identify the effect of the MR damper on the system. Figure 6 shows the steady state response of the un-damped system calculated according to (10) compared to the amplitude of the experimentally induced vibration for different operating frequencies. It is clear that the simulation and experimental results are in close agreement and the experimental amplitude does not reach infinity due to the damping present in the system. The experimental results for the same system of different masses of 30.26, 34.26 and 38.26 Kg. are shown in Figure 7.

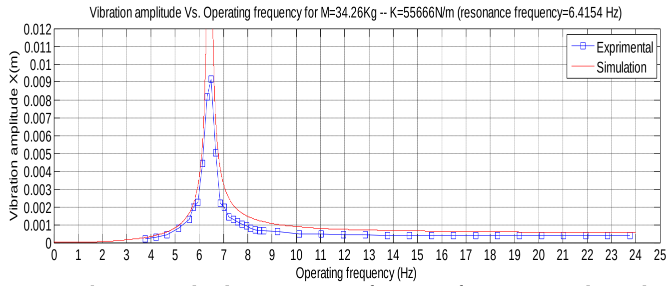


Fig. 6 Vibration amplitudes vs. operating frequency for passive un-damped system

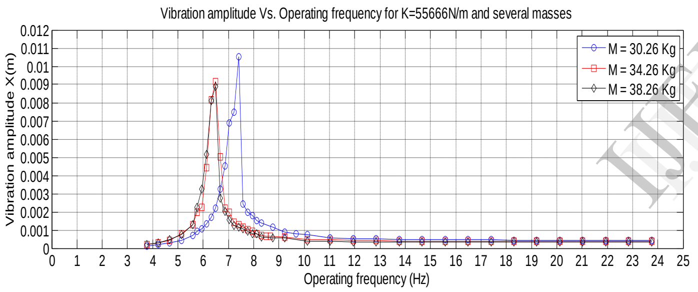


Fig. 7. Vibration amplitudes vs. operating frequency for un-damped system for different masses

Also, it shows that the three cases reach their maximum amplitude at a frequency that is very close the resonance frequency of the simulation results especially for M=34.26 Kg. This is a good indication that the designed system performs well.

Figures 8, 9 and 10 show the vibration amplitude as a function of the operating frequency for passive and semi-active systems for several volts for M= 30.26, 34.26 and 38.26 Kg. respectively. It can be noticed that for all values of voltages supplied to the MR damper, the damper can effectively reduce the amplitude of vibration at resonance.

Also it is shown that as the voltage increases, the amplitude of vibration decreases. The effect of increasing the mass of the system is clear Figure 8, 9 and 10. As the mass increases, the amplitude of vibration decreases and the maximum amplitude occurs at higher frequencies. This increases the band width of the system. Table II shows the average values of the operating frequency difference Δf between the un-damped and MR damped systems and its relation with increasing the mass of the system.

TABLE II Average values of the operating frequency difference Δf for each system mass

System mass (Kg)	Δf (Hz)
30.26	7.72
34.26	9.31
38.26	10.22

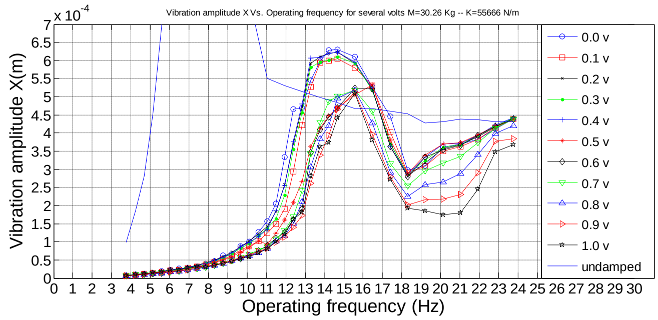


Fig. 8. Vibration amplitudes vs. operating frequency for passive and MR-damped system for different volts M=34.26 Kg.

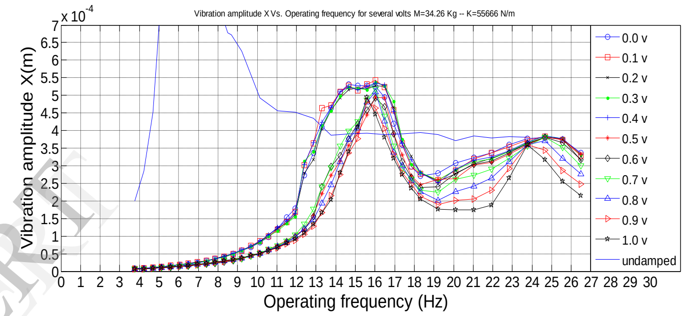


Fig. 9. Vibration amplitudes vs. operating frequency for passive and MR-damped system for different volts M=34.26 Kg.

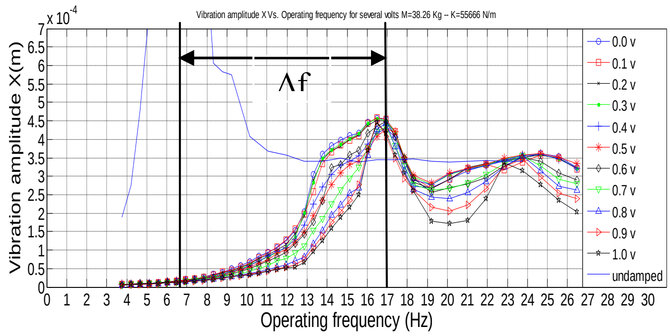


Fig. 10. Vibration amplitudes vs. operating frequency for passive and MR-damped system for different volts M=38.26 Kg.

In addition to the mentioned notes it is clear that, with increasing the operating frequency the semi-active system implies high amplitude that is higher than the un-damped system followed by a drop in the amplitude value then the amplitude increases again and finally decreases. Figures 11, 12 and 13 show a three dimensional (3D) plot of the vibration amplitude versus the operating frequency for several volts for M= 30.26, 34.26 and 38.26 Kg. respectively.

From these surfaces any amplitude for a specific operating frequency and supplied voltage can be calculated.

VI. Conclusion

From this study we can conclude that the MR damper is very effective for a SDOF system having rotating unbalance. The comparison between the experimental results of the passive undamped system and the semi-active system showed that the MR damper improves the stability of the system. Moreover, the increasing of the system mass with the use of MR damper decreases the system vibration amplitude and the maximum vibration amplitude occurs at higher frequencies resulting in increasing the operating frequency range that can be used.

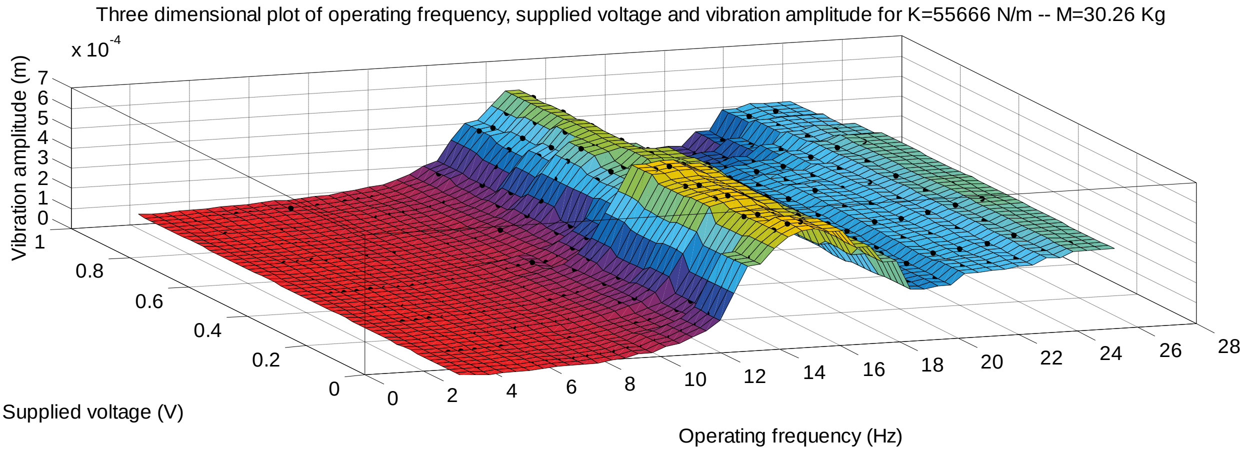


Fig. 11. 3D plot of vibration amplitude vs. operating frequency for several volts M=30.26 Kg.

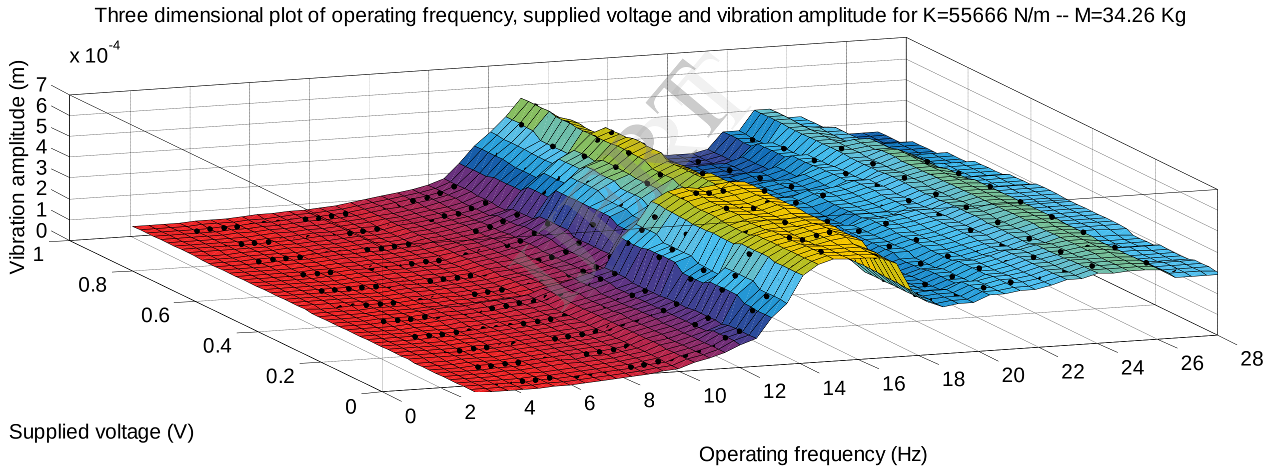


Fig. 12. 3D plot of vibration amplitude vs. operating frequency for several volts M=34.26 Kg.

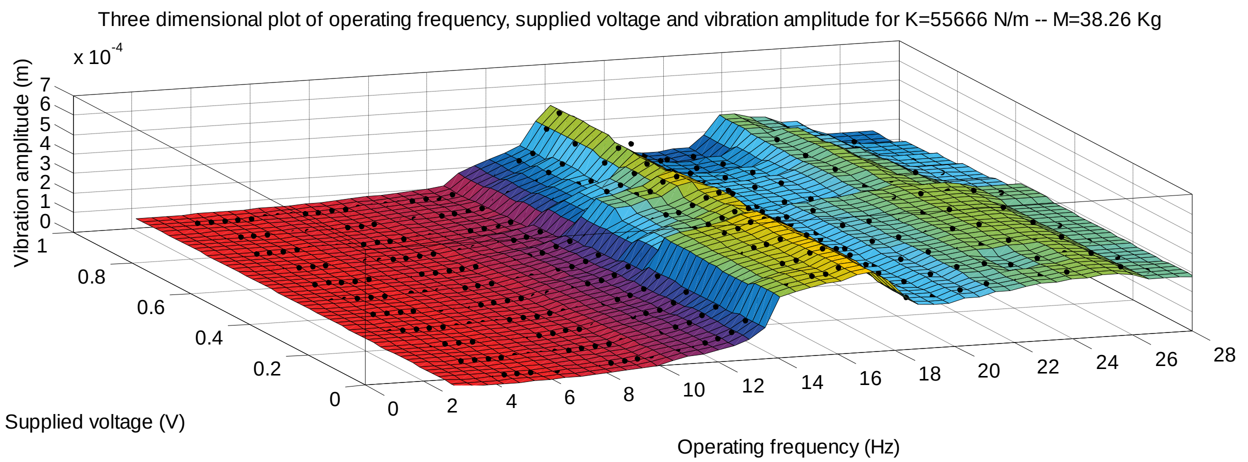


Fig. 13. 3D plot of vibration amplitude vs. operating frequency for several volts M=38.26 Kg.

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