Theoretical and Experimental Study of the Vibration of a Drum Type Washing Machine at Different Speeds

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Abstract

In the present work, theoretical and experimental Study of vibration of a drum type of Horizontal Washing Machine. The effect of the Isolators stiffness, damping coefficient and the drum mass for specific laundry capacity also has been studied. The work in this research has been carried out analytically by using MATLAB, and Study experimentally the effect of different speed and unbalance force during the spinning cycle of the washing machine at four sides of it. This analysis aims to reducing the excited vibration. This was achieved theoretically by investigate the effect of various parameters in order to assign property values to increase the isolation efficiency to reach optimum design. The results is show that drum vibration amplitude reduced to 42 % at spinning speed 1000 rpm and 41% at 1200, 1400 rpm when the applied selected parameters.

Keywords: Horizontal washing machine; spinning cycle; vibration drum.

Nomenclature

A,B	Constants	-
c	Damping coefficient	N.s/m
F_o	The inertia force as an x-component	N
F_{tx}	The force transmitted with x axis	N
F_{ty}	The force transmitted with y axis	N
k	Spring coefficient	N/m
M	Total mass of the drum	kg
m	Unbalanced mass	kg
N	Laundry number of revolutions	rev/min
R	The radius of the drum	m
r	Ratio between the exciting and natural frequencies	-
v	Velocity	m/s
ω	Angular speed	rad/s
ω_n	The natural frequency	rad/s
X	The peak amplitude of the drum vibration	m
x	Displacement from spring's neutral position	m
$\dot{x}$	Vibration velocity	m/s
$\ddot{x}$	Vibration acceleration	m/s^2
δ_s	The deflection in the spring	m
δ_d	The deflection of the damper piston	m
γ	Angle inclination spring with the vertical direction	deg.
β	Angle inclination damper with the vertical direction	deg.
ξ	Damping ratio	-
φ	phase angle between the output motion and the exciting force	deg.

1. Introduction

Unbalance in a rotary machine is a public basis of vibration excitation that one sees in daily life such as vibration of the washing machine. Horizontal washing machines are undergoing to squeaky vibration through the spinning process. By analysis and selection of the washing machine parameters can diminution vibration and noise excited by the rotating unbalance through the spinning cycle, as well increase the isolate performance.

Galal Ali Hassaan [1], [2015] investigated theoretically the vibration analysis of a horizontal washing machine during spinning unbalanced. He stated the effect of the various parameters of the Isolators stiffness, damping coefficient and the drum mass of the washing machine on the vibration amplitude and velocity of the machine drum in terms of the spinning speed with arrange from 200 to 1200 rev/min . Galal Ali Hassaan [2], [2015] studied the Comparison between isolation efficiency and the vibration velocity to assign optimal parameters in order to increase the efficiency of the Isolators and reduce the vibration velocity of the drum. Using MATLAB toolbox which taking the spinning speed as input variable, this leads an increasing the isolation efficiency to more than 97 % and reduce the drum vibration velocity to less than 14.8 mm/s RMS.Galal Ali Hassaan [3], [2015] discussed the modeling of the dynamic system and the dynamic absorber which is assumed to have a known mass and damping coefficient of a horizontal washing machine. The absorber stiffness is tuned using a MATLAB optimization toolbox. As a result of this paper the drum vibration velocity reduced to less than 0.7 mm/s RMS and increases the isolation efficiency to greater than 99.7 % for spinning speeds ≥ 400 rev/min. Papadopoulos and

Papadimitriou [4], [2001] simplified three dimensional dynamic model of a horizontal-axis portable washing machine. This used to predict the verge of walking instability during the spinning cycle. Two methods of stabilization are done: design-based and control-based methods to eliminate instability and vibrations. These methods satisfy the current trend towards portable, lightweight full-feature washing machines. They concluded that rotational slip can be a major problem if the washer center of mass is not on the plane of rotation of the laundry mass.Koizumi, et al. [5], [2005] presented a modeling of a vibration analysis model for a drum type washing machine. The modeling of a vibration analysis model for a drum type washing machine was presented. This model can be analyzed the vibrations of the outer tub, the frame, and the force to the floor which is considered as a rigid body in the spin-drying stage under various unbalance conditions varied from 100 rpm to 300 rpm and Unbalanced masses of 300 g and 150 g . Spelta, et al. [6], [2008] analyzed and designed a control system for the reduction of the mechanical vibration and acoustic noise in a washing machine. An experimental protocol was proposed and tested on a sensored machine to highlight the system dynamical behavior and it was shown that the damping control ensures a low level of vibration at every working condition. Nygards and Berbyuk [7], [2010] concentrated on the few parts of vibration elements in clothes washers, the limit expansion through the investigation of tub development, the vibration yield from the machine to the environment, and the strolling, propensity of the framework. Kolhar and Patel [8], [2013] proposed an idea for the optimization of a washing machine in terms of reduction in drum vibration, power consumption and water consumption. They formulated a mathematical model for reducing the drum vibration and an improved drum design to further reduce the vibrations.

2. Experimental Work

Washing machine used in this research of the analysis is the Front load washing machine of the type (LG direct drive) with wash capacity 8 Kg and dry capacity 5 Kg, dimension 600 mm width, 560 mm depth and 850 mm height [9].

A manufacturing device have been used to measure spring stiffness, because not to provide a device can withstand high weights to carry out the experiment where the requested measure spring stiffness Force of up to 160 N Also Examination device of the damping coefficient of the user to measure the damping coefficient of Shock Absorber where springs each of a stiffness K = 5.88 KN/m and inclined with the vertical direction is $\theta = 10^c$ degree, and Three dampers each of a damping coefficient c =74.8 N.s/m and inclination with the vertical direction is $\beta = 30^c$ degree. With drum total mass is M = 30 Kg and The laundry mass is m (0.1, 0.15 and 0.2 Kg) at rotor speed is N (1000, 1200 and 1400 rpm).

Procedure of Measuring Spring Coefficient (K)

The aim of experiment is to measure the spring coefficient (K) for spring type of LG washing machine, as shown in Figure (1) [10].

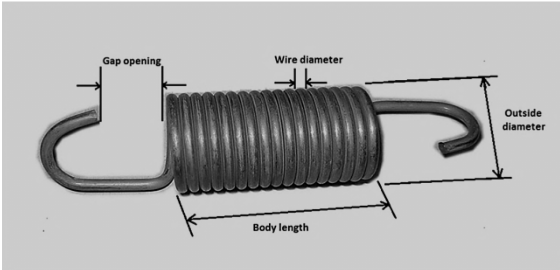

Figure 1: LG Electronics Washing Machine Hinge Spring

The spring was hanged on the installer column and after increasing masses on the spring, the extension of spring was measured after the addition of each mass, this device is shown in figure (2).

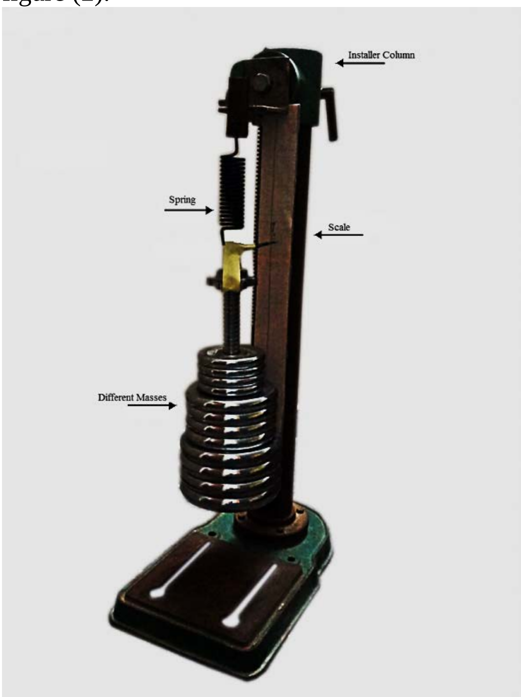

Figure 2: Examination device of the spring coefficient

The Graph:

Figure (3) shows a relation between the displacement magnitudes of hinge spring with the affected forces for each mass added, the tendency is a straight line, and the constant stiffness (k), is calculated:

$$K = \Delta y / \Delta x = 5880 \text{ N/m}$$

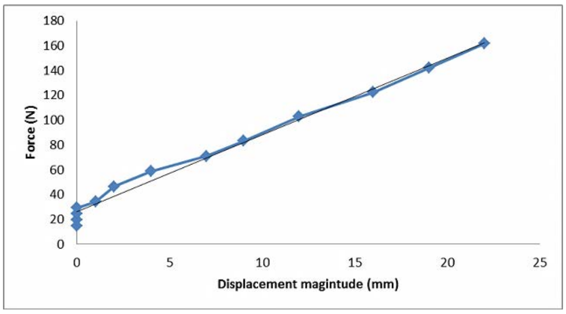

Figure 3: Force – displacement curve for Hinge Spring of washing machine

Procedure of Measuring damping coefficient(c)

The aim of experiment is to measure the damping coefficient (c) for Shock Absorber type of LG washing machine, as shown in Figure (4).[11]

Figure 4: Washing Machine Shock Absorber

Method of Measurement:

A. Suspened the damper on the installation column.
B. Determination a certain distance from the tip end of the damper until the surface level.
C. Suspened the blocks on the free end of the damper.
D. Adjustment the stopwatch to calculate the time it takes to reach the mass of the surface level at every addition of the block.

Examination device of the damping coefficient of the user to measure the damping coefficient of Shock Absorber is shown in figure (5).

The Graph:

Figure (6) shows a relation between the velocities of shock absorber with affected forces for of each mass addition, the tendency is a straight line, the constant coefficient (c) that is calculated:

$$c = \Delta F / \Delta V = 74.8 \text{ N.s/m}$$

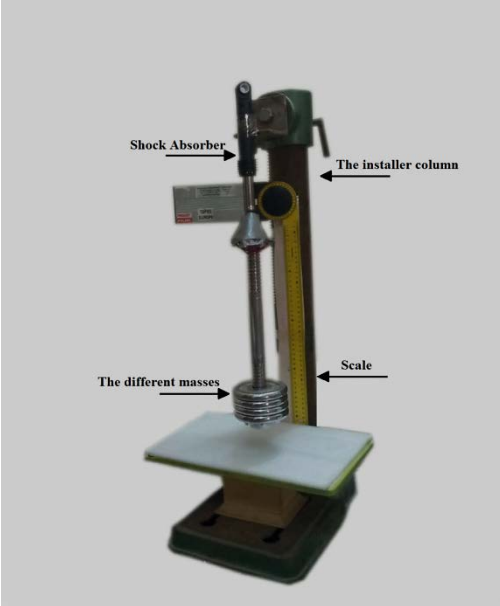

Figure 5: Examination device of the damping coefficient

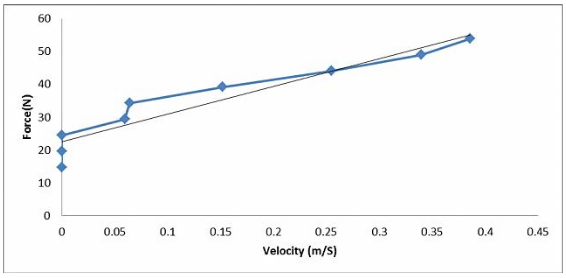

Figure 6: Force – velocity curve for shock absorber of washing machine

To measure the Velocity and Displacement, A device it has been used (VIBRATION METERS) Where Vibration sensors are installed sites in the middle of each panel in the right and left and upper and above the washing machine base, as shown in figure (7).

Figure 7: Installation locations of Vibration sensors

Vibration Meter Calibration

The calibration device (vibration meter) by comparing it with (Accelerometer Sensor) is shown in figure (8). The ADXL345 Digital Accelerometer features 4 sensitivity ranges from +/- 2G to +/- 16G. And, it supports output data rates ranging from 10 Hz to 3200 Hz. [12]

Accelerometer Sensor was calibration by determining the sensor output for each axis, supporting the sensor towards the earth into the X, Y and Z directions and considering (zero g) rate for each axis, as shown in figure (8).

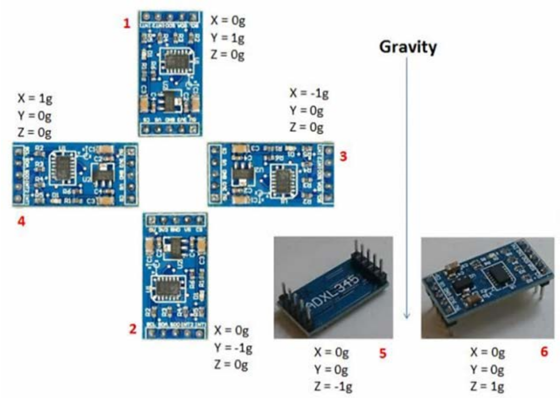

Figure 8: Accelerometer Sensor Calibration

Unbalanced masses were attached to the inside of the spin drum. Instead of attaching clothes to the inner wall of the spin drum in the spin-drying stage, Pieces of iron plates were used. Compose system test with Vibration measuring, as shown in figure (9).

Unbalanced masses were attached to the inside of the spin drum. Instead of attaching clothes to the inner wall of the spin drum in the spin-drying stage, Pieces of iron plates were used. Pieces of iron installed by magnets of the type Super Strong Round Disc Magnets, Figure (10) illustrates the picture of where the mass is installed in a drum of washing machine by Magnets.

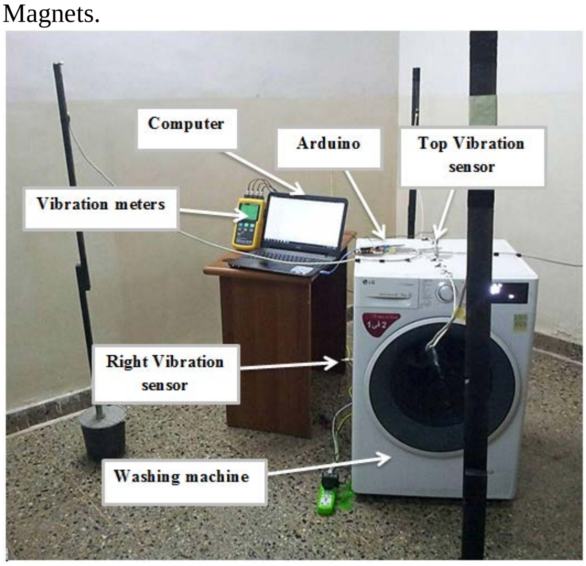

Figure 9: Building equipment for framework of washing machine

Figure 10: Unbalance mass site

3. Theoretical Part

For the purpose of investigation the system can be reduced in to simple physical diagram system as shown in figure (11) and (12), Balancing involves placing correction masses on to the drum, so that the centrifugal forces due to these masses cancel out those caused by the inherit unbalanced mass, thus canceling out vibration.

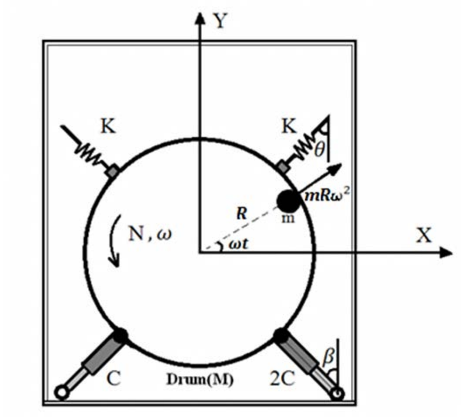

Figure 11: Physical Model of a Front load washing machine

Figure 12: Illustration of suspension parts of washing machine

3.1 Drum Vibration analysis during spinning

By the basics of force stability and using the Newton second law of motion the equation of motion can be written as follows:

$$\sum F_x = (M - m)\ddot{x} \quad (1)$$

M = Total mass of the drum

m = unbalanced mass

On more simplistic we get the equation given below:

$$M\ddot{x} + c_e\dot{x} + k_e x = F_x \qquad (2)$$

Where,

$$F_x = F_o \cos\omega t = mR\omega^2 \cos\omega t \qquad (3)$$

This force will have an orientation ωt with the x-axis; the inertia force F_o will have an x-component.

The dynamic model of the system can be derived in terms of the dynamic motion x and y in the two directions (x and y). However, it is possible to perform simpler analysis by considering only the motion in one direction (say x). The drum dynamic motion in the horizontal direction is x [1]. The deflection in the spring δ_s depends on tits inclination angle θ as shown in Figure (13):

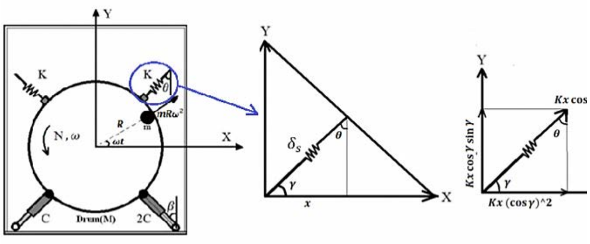

Figure 13: Representation deflection in the spring

$$\delta_s = x\cos\gamma = x\sin\theta \qquad (4)$$

In the same way, the deflection of the damper piston δ_d is:

$$\delta_d = x\cos\gamma = x\sin\beta \qquad (5)$$

The differential equation of the drum is obtained as:

$$M\ddot{x} + 3c(\sin\beta)^2\dot{x} + 2k(\sin\theta)^2 x = mR\omega^2\cos\omega t \qquad (6)$$

Where,

$\dot{x}$: is the vibration velocity & $\ddot{x}$: is the vibration acceleration

With: $c_e = 3c(\sin\beta)^2$, $k_e = 2k(\sin\theta)^2$

In particular, the response is harmonic and has the same frequency as the excitation frequency. Hence, a solution of Eq. (6) can be assume in the form

$$x = A\cos\omega t + B\sin\omega t \qquad (7)$$

In which A and B are constants yet to be determined, Inserting Eq. (7) into Eq. (6) give,

$$M(-A\omega^2\sin\omega t - B\omega^2\cos\omega t) + c_e(-A\omega\sin\omega t + B\omega\cos\omega t) + k_e(A\cos\omega t + B\sin\omega t) = F_o\cos\omega t \qquad (8)$$

Equation (8) can be satisfied provided the coefficients of $\sin\omega t$ on both sides of the equation are equal, and the same can be said about the coefficients of $\cos\omega t$, which yields two algebraic equations in A and B as follows:

$$(k_e - M\omega^2)A + c_e\omega B = 0 \qquad (9)$$

$$(k_e - M\omega^2)B + c_e\omega A = F_o \qquad (10)$$

Using Cramer's rule, the solution of Eq. (9) and (10) is:

$$A = \frac{-c_e\omega F_o}{(k_e - M\omega^2)^2 + (c_e\omega)^2} \qquad (11)$$

$$B = \frac{(k_e - M\omega^2)F_o}{(k_e - M\omega^2)^2 + (c_e\omega)^2} \qquad (12)$$

Introducing Equations (11) and (12) in Eq. (7), the steady-state solution can be obtained as follows:

$$x = \frac{F_o}{\sqrt{(k_e - M\omega^2)^2 + (c_e\omega)^2}}\sin(\omega t - \gamma) \qquad (13)$$

Otherwise $\omega_n = \sqrt{\frac{2K}{M}}$, $\xi = \frac{3c}{2M\omega_n}$ $F_o = mR\omega^2$

Therefore

$$x = \frac{\left(\frac{mR}{M}\right)r^2}{\sqrt{(1-r^2)^2 + (2\xi r)^2}} \qquad (14)$$

Where,

$$x = steady\ stat\ solution$$
$$\frac{\omega}{\omega_n} = r$$

4. Results and Discussion

The vibration amplitude and velocity vibration are obtained from the experimental work with respect to time is plotted and presented, at different unbalance masses and speed.as shown in the figures (14) and figure (15). It Can be seen from these Figures that:

1-The vibration amplitude and velocity in the right hand that contains two dampers (right panel) is higher than the other side that contains one damper (left panel).

2-Increasing vibration amplitude and velocity vibration by increase the proportion of unbalanced mass for each of the right panel and the left panel during the dry speed (1000, 1200 and 1400) rpm

1-The vibration amplitude and velocity vibration in the right hand that contains two dampers (right panel) are higher than the other side that contains one damper (left panel) because of the displacement.

2- Increasing vibration amplitude and velocity vibration by increase the proportion of unbalanced mass for each of the right panel and

the left panel during the dry speed (1000, 1200 and 1400) rpm.
3- Increasing vibration amplitude and velocity vibration by increase the rotational speed but they seem very nearly in magnitude at speed (1200 and 1400) rpm into sides of panel.
4- Vibration amplitude decreases with different percentages between the two sides (right and left) panel with increased unbalance mass during the dry speed (1000, 1200 and 1400) rpm.
5- Velocity vibration increases with different percentages between the two sides (right and left) panel with increased unbalance mass during the dry speed (1000, 1200 and 1400) rpm.
6- They illustrate the percentage of vibration amplitude and velocity vibration during the spinning cycle (1000, 1200 and 1400) rpm with unbalanced masses, as shown in tables (1) and (2).

4.1 Examination and study effect of the various parameters theoretically

This work presents theatrically the effect of the parameter such as total mass of the drum (M), spring coefficient (K) and damping coefficient (c) on the vibration amplitude at unbalance masses (100,150 and 200) g shown in figures (16) to (18),to Select the washing machine parameters in order to decrease the vibration amplitude and velocity. From these figures it can effortlessly to a strong conclusion that the selected parameters of the isolators and drum mass are: M = 50 kg, k = 5000 N/m and c = 300 N.s/m.

Comparison between the real parameters and selected parameters to drum vibration amplitude at the spinning speed (1000, 1200 and 1400) rev/min with unbalance masses (100,150 and 200) g, The minimum vibration amplitude was obtained at a mass of 150 g at 1400 rpm is shown in Table (3)

Table 1: Percentage of vibration amplitude during the spinning cycle (1000, 1200 and 1400) rpm with unbalanced masses

Dry Speed (rpm)	Unbalance Mass (g)	Decreasing of Vibration Amplitude (%)		Vibration Amplitude between Right and Left panel (%)
		Right panel	Left panel	
1000	100	49	42	30
	150	62	62	19
	200	63	66	15
1200	100	58	19	59
	150	67	37	58
	200	70	47	55
1400	100	57	23	59
	150	67	43	57
	200	69	48	55

Table 2: Percentage of velocity vibration during the spinning cycle (1000, 1200 and 1400) rpm with unbalanced masses

Dry Speed (rpm)	Unbalance Mass (g)	Increasing of Velocity Vibration (%)		Velocity Vibration between Right and Left panel (%)
		Right panel	Left panel	
1000	100	52	10	53
	150	61	17	58
	200	64	21	60
1200	100	55	21	54
	150	64	35	55
	200	68	42	55
1400	100	57	24	55
	150	65	43	51.5
	200	68	42	56

Table 3: Vibration Amplitude of real parameters and selected parameters

Spinning Speed (rpm)	Unbalance masses (g)	Real Vibration Amplitude (mm)	Selected Vibration Amplitude (mm)	Ratio Improvement (%)
1000	100	0.7998	0.4655	42
	150	1.2101	0.7043	42
	200	1.5927	0.9271	42
	100	0.7915	0.4651	41

1200	150	1.1976	0.7036	41
	200	1.5763	0.9262	41
1400	100	0.7866	0.4648	41
	150	1.1901	0.7032	41
	200	1.5665	0.9256	41

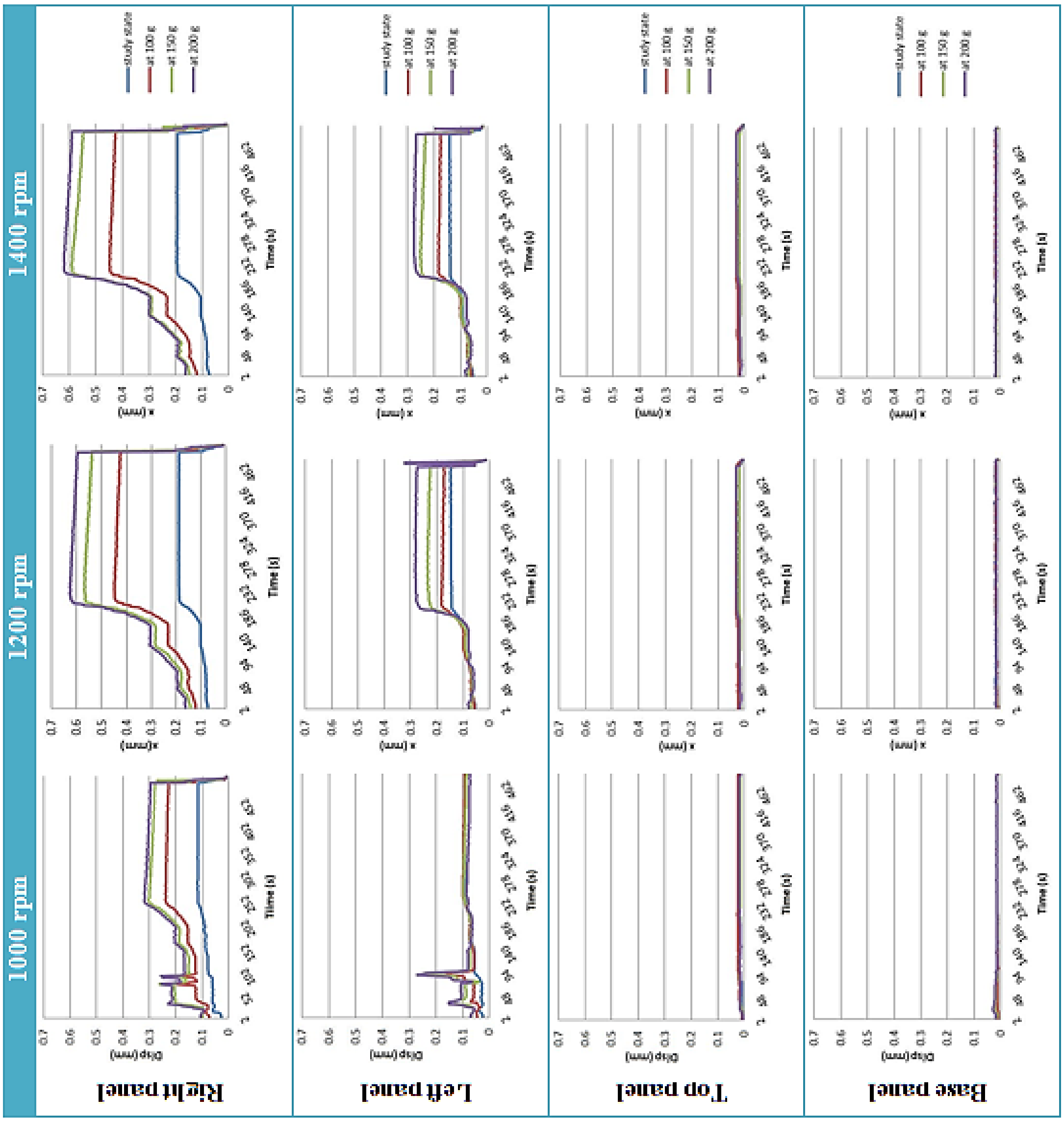

Figure 14: Amplitude of Vibration in Right, Left, Top and Base panel versus time at different unbalance forces (100,150 and 200) g and different speed (1000, 1200 and1400) rpm

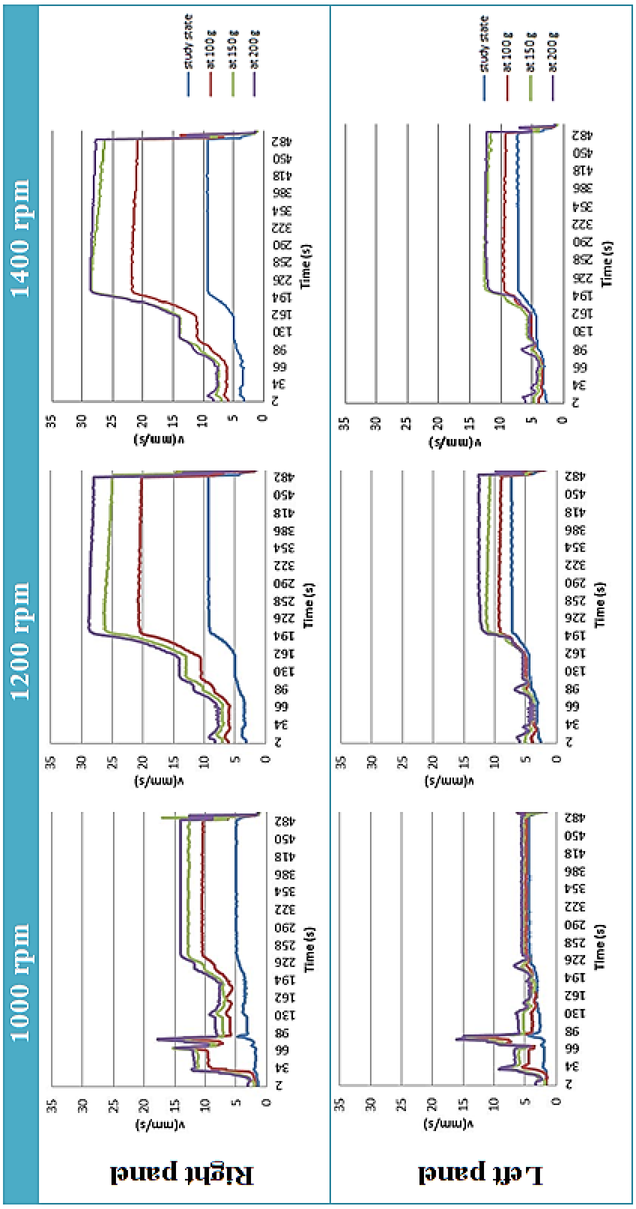

Figure 15: Velocity Vibration in Right and Left sides of panel at different unbalance forces and different speed (1000, 1200 and 1400) rpm

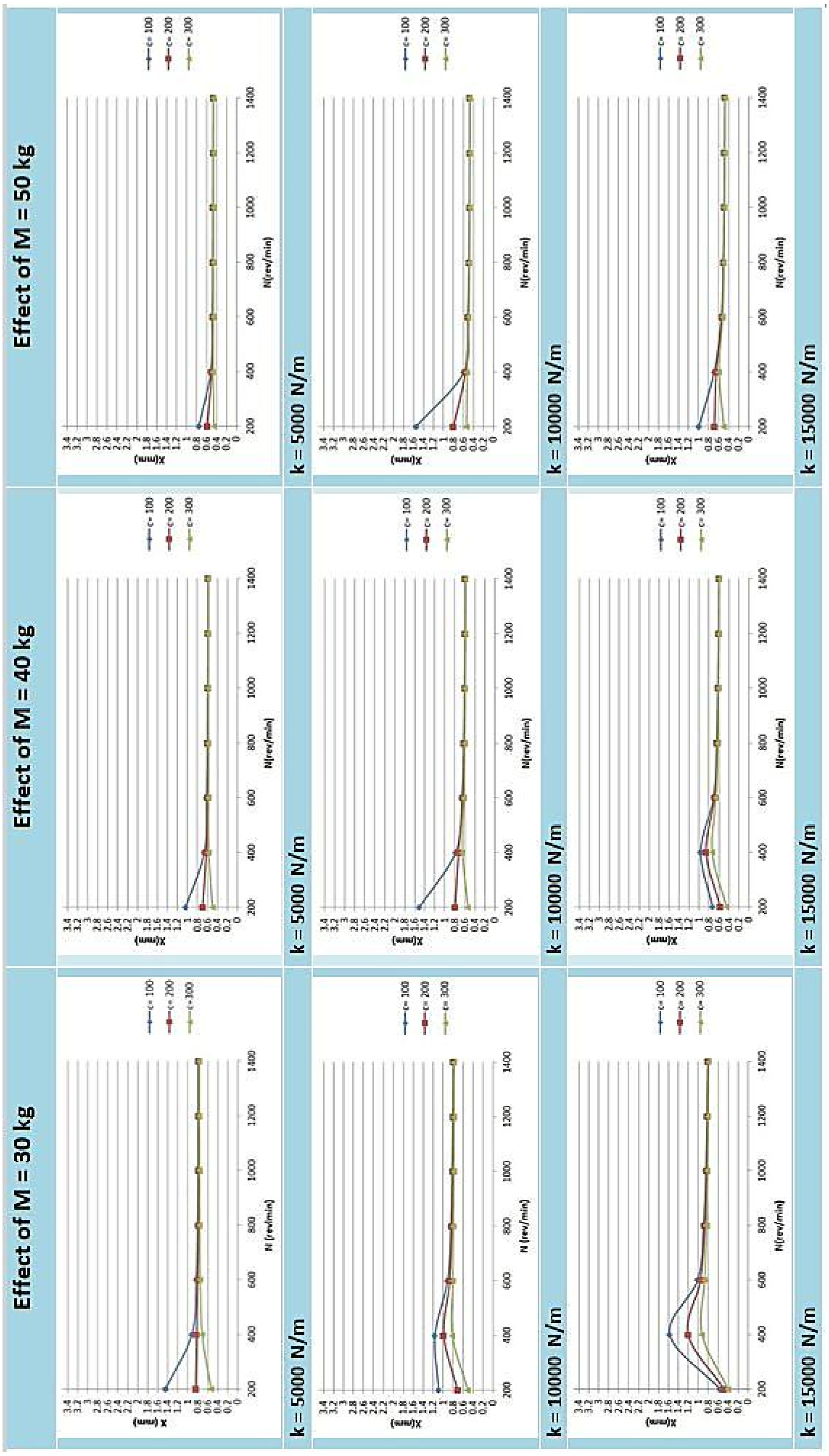

Figure 16: Amplitude of Vibration versus time at Mass of drum (30, 40 and 50) Kg and spring coefficient (5000, 10000 and 15000) N/m at speed 1000 rpm

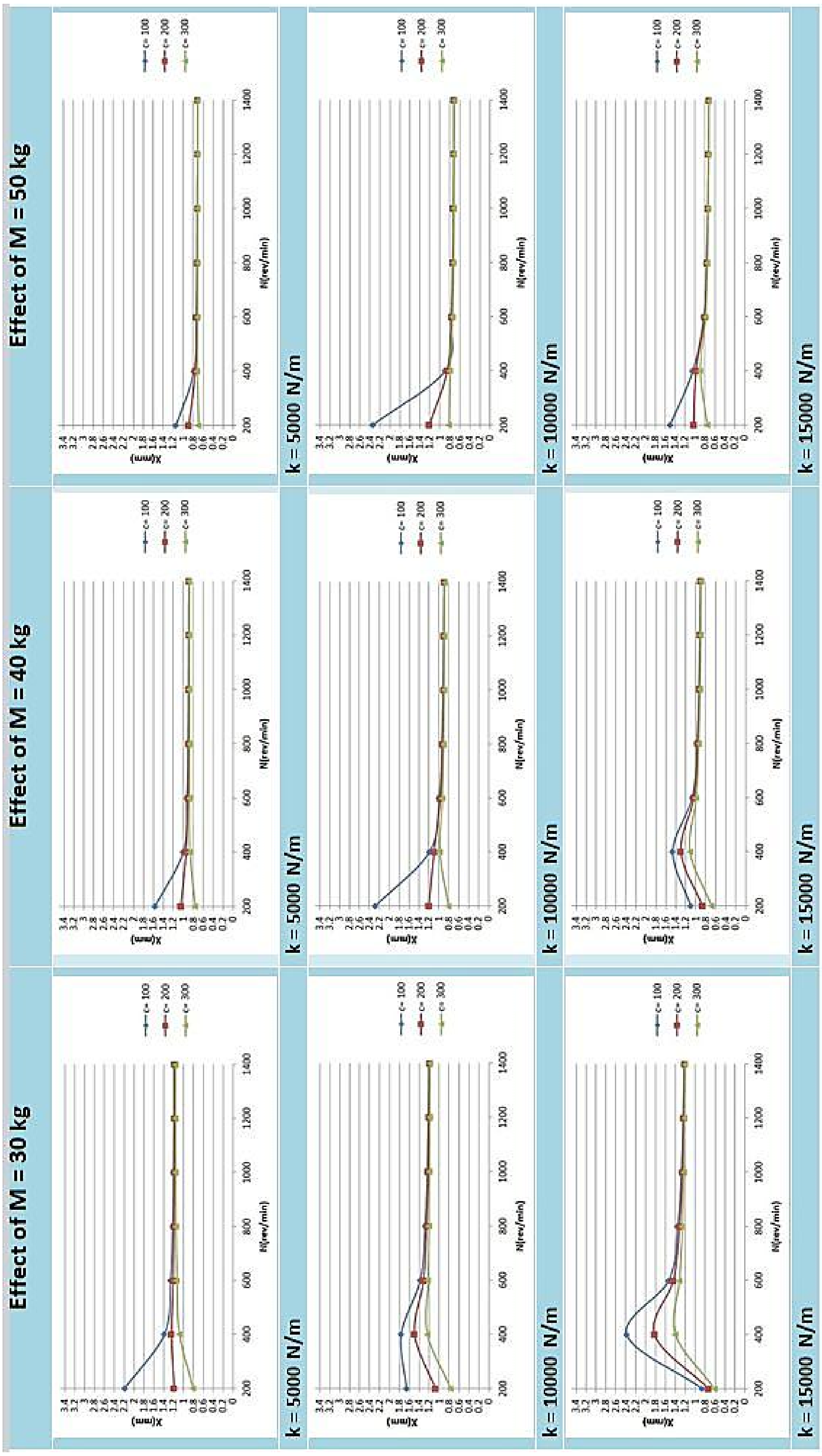

Figure 17: Amplitude of Vibration versus time at Mass of drum (30, 40 and 50) Kg and spring coefficient (5000, 10000 and 15000) N/m at speed 1200 rpm

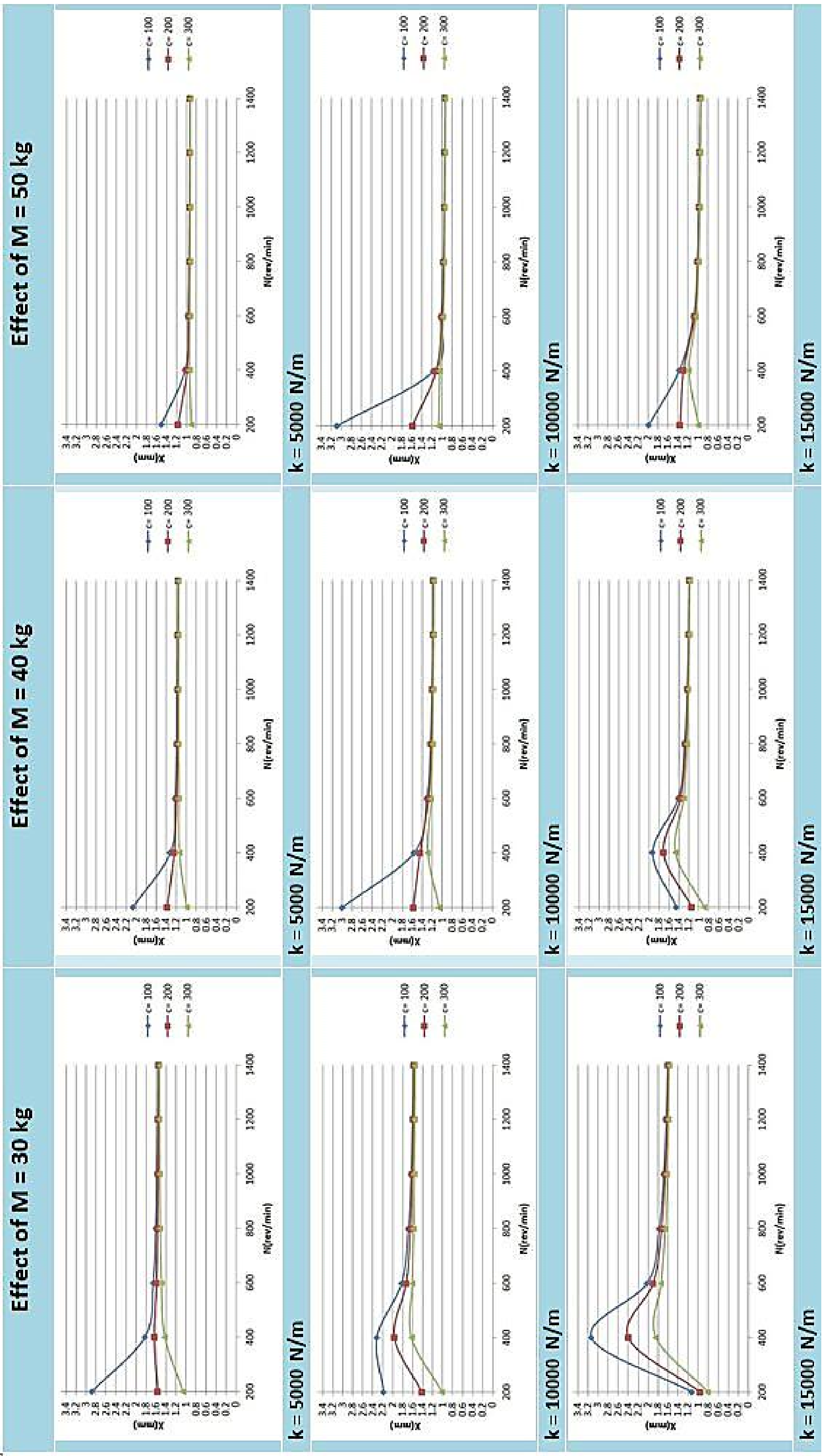

Figure 18: Amplitude of Vibration versus time at Mass of drum (30, 40 and 50) Kg and spring coefficient (5000, 10000 and 15000) N/m at speed 1400 rpm

5. Conclusions

1- It was possible by the proper selection of isolators stiffness, damping coefficient and the drum mass to come down with the vibration amplitude ratio Improvement to 41% at (1200 and 1400) rpm and 42% at 1000 rpm .

2- With a drum mass of 30 kg, isolator stiffness of 5 KN/m and a damping coefficient of 300 N.s/m, the drum vibration amplitude decreased from 0.79 to 0.46 mm with unbalance mass 100 g and from 1.21 to 0.7 mm with unbalance mass 150 g and from 1.59 to 0.92 mm with unbalance mass 200 g from 1000 to 1400 rpm.

3- The vibration amplitude and velocity in the side that contains two dampers is higher than the other side that contains one damper, where vibration amplitude between Right and Left panel is 30% with unbalanced mass 100 g, 19% with unbalanced mass 150 g and 15% with unbalanced mass 200 g during the spinning cycle 1000 rpm. Also Velocity Vibration between Right and Left panel is 53% with unbalanced mass 100 g, 58% with unbalanced mass 150 g and 60% with unbalanced mass 200 during the spinning cycle 1000 rpm.

4- Increasing vibration amplitude and velocity vibration by increase the proportion of unbalanced mass during the dry speed (1000, 1200 and 1400) rpm.

5- Vibration amplitude decreases with different percentage between two sides (right and left) panel with increased unbalance mass during the dry speed (1000, 1200 and 1400) rpm.

6- Velocity vibration increases with different percentage between two sides (right and left) panel with increased unbalance mass during the dry speed (1000, 1200 and 1400) rpm.

7- Selection of the washing machine parameters to a strong conclusion, selected parameters of the isolators and drum mass are: M = 50 kg, k = 5000 N/m and c = 300 N.s/m.

Recommendations

The following studies can be suggested for future work:

1-Studying the effect of many types damper and spring on the transmissibility and Isolation Efficiency.

2- Investigating experimentally the effect of Isolators stiffness, damping coefficient and the drum mass on the transmissibility and isolation efficiency at speed (200 to 1000) rpm.

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دراسة نظرية وتجريبية للأهتزاز لنوع من حوض آلة الغسيل بسرعات مختلفة

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الخلاصة

في هذا العمل، أجريت دراسة نظرية وتجريبية لاهتزاز حوض غسالة ذات فتحة امامية. لدراسة تأثير جساءة العزل و معامل التخميد وكتلة حوض الغسيل لسعة غسيل محدده. تم تنفيذ العمل التحليلي في هذا البحث باستخدام برنامج MATLAB، ودراسة تجريبية لتأثير مختلف السرع وقوى عدم الاتزان خلال دورة تجفيف الغسالة عند الجوانب الأربعة. يهدف هذا التحليل إلى تقليل الاهتزاز المثار بواسطة التحقق نظريا من تأثير مختلف المعاملات لتحديد قيم المعاملات (k ,M and c) لزيادة كفاءة العزل وتحسين التصميم. أظهرت النتائج أن سعة اهتزاز الحوض خفضت إلى 42٪ عند سرعة دوران 1000 دورة في الدقيقة و 41٪ عند السرع 1200،1400 دورة في الدقيقة عند تطبيق المعلمات المختارة.