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Determining the Mass and Angular Position of the Unbalanced Load in Horizontal Washing Machines

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Abstract— **Horizontal washing machines use significantly less water and energy as opposed to conventional machines. Unbalanced load in washing machines limit the performance, and therefore accurate estimation of unbalanced loads, including their magnitudes and locations in washing drum, is required. In this paper, we propose an approach to evaluate the angular position and mass of the unbalanced load. A simulation model of the system was developed and various experiments were performed. The proposed algorithms and the developed experimental system can estimate the angular position of the unbalance load with maximum %3 error. The research on unbalanced load increase the performance of the washing machine and in addition to this, this research can be used to eliminate unbalanced load in future studies.**

I. Introduction

HORIZONTAL washing machines (horizontal-axis, front-loading) use significantly less water and energy to do the same job as conventional machines. These advantages make the horizontal washing machines increasingly popular.

Washing machine users are aware or concerned about the capacity of the machine, its spinning speed and its energy consumption. However, capacity of the machine and spinning speed affects the unbalanced load and unbalanced forces on the washing machine. There are three main programs. They are washing, rinsing and spinning in washing machines. The main difference among these three programs is the speed of the drum. In washing and rinsing programs, the speed of the drum is between 0 and 80 rpm. At these speeds, there is no mechanical and electrical problem in washing machine system. However in the spinning program, the angular speed of the drum is between 0 and 2000 rpm. In the spinning program, after 80 rpm angular speed, washing machine starts to make unwanted and risky vibrations; which may lead to machine break down.

As a result of the problem during the spinning program, after 80 rpm angular speed, clothes stick to surface of the drum due to the centrifugal force. Clothes disperse to the surface of the drum in a balanced or an unbalanced form. Sometimes clothes cannot balance each other or disperse unbalance form, and as a result unbalanced load is formed. Washing machines remain big and heavy, weighing usually over fifty kilograms. This is due to the unbalanced rotation of the laundry mass during spinning. The rotating clothes are not evenly dispersed in the drum, resulting in significant centrifugal imbalance forces, which tend to destabilize the washer. After 80 rpm angular speed, unbalanced load starts to cause vibrations on the washing machine. The magnitude of the vibrations depends on the amount of unbalanced load and the speed of the drum. However at the resonance speed, which is 180 rpm angular speed, the vibration of the washing machine can reach unacceptable vibrations such as walking, turnover and may break the drum.

Estimation of the unbalanced load for horizontal washing machines has also been studied by some researchers. Unbalanced load estimation using multiple sensors and utilizing sensor fusion has been presented in [1, 2]. Also, a simplified 3D model of a horizontal-axis portable washing machine and stabilization methods were presented in [3]. To improve the washing machine systems, designers try to decrease or eliminate the unbalanced loads and their forces. Designers improve several ways to decrease and eliminate the unbalanced load and force. Some extra equipment is designed for controlling vibrations such as "G-Fall Balancer" was presented in [4] and "Hydraulic Balancer" was presented in [5]. In addition to this control algorithm are used to decrease and eliminate the unbalanced load and force, such as active control strategy [6].

In this paper, we try to find out angular position of the unbalanced load and its weight in order to design of better equipments and working conditions, and new control systems and algorithms to reduce or eliminate the unbalanced load and its forces. Using multiple sensors and sensor fusion systems might provide better results, but these systems are more expensive and more complex, leading to serious increase in machine's production cost. The approach presented here is to solve this problem using the cheapest and simplest systems, so we use minimum number of sensor or sensors system.

This paper organized as follows. In section 2 the mechanical system of the washing machine is analyzed and equations of motion were generated. A simulation model including the variables affecting the unbalanced load and simulation results are presented in Section 3. In Section 4, we present the approach to determine the unbalanced load and angular position. Finally, the vibration piezoelectric sensor, experimental setup, and various experiments performed to determine the magnitude of the unbalanced load are explained in Section 5.

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II. MODELLING

In this part, two dimensional basic dynamical model of the washing machine with unbalanced load is analyzed. We use the model of the washing machine to analyze the movement of the drum according to unbalanced load.

After 80 rpm drum speed, loads are attached to the inner surface of the drum; this situation causes the unbalanced load. Vibrations and the displacements are formed by the unbalanced load force. Spring and damper mechanism develop counter force to the unbalanced load forces. To analyze the dynamic mechanism, free body diagram is used. A free body diagram of the drum is shown in Figure 1.

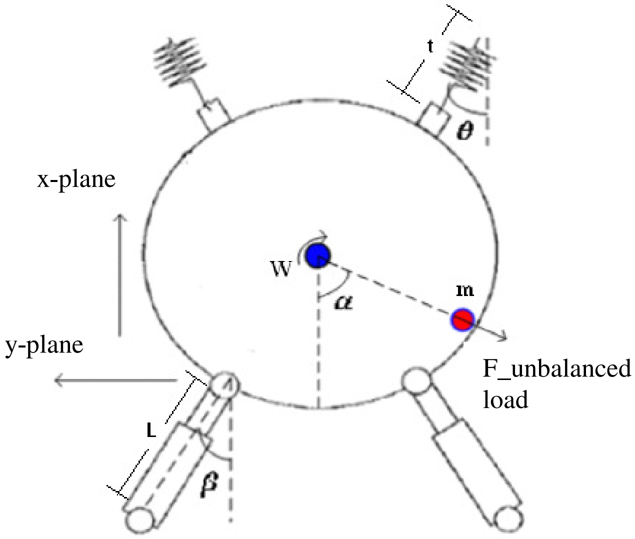

Fig. 1. Model of the unbalanced load in a horizontal load washing machine.

The parameters used in the analysis have been defined in Table I.

TABLE I
PARAMETERS

Symbol	Definition
$m_{unbalanced\_load}$	Mass of the unbalanced load
m_{load}	Mass of the total load
x	Distance and direction on the x-plane
y	Distance and direction on the y-plane
k	Spring constant
t	Distance and direction in which the spring is deformed
c	Damper constant
L	Distance and direction in which the damper is deformed
θ	Angle between spring and x-plane
β	Angle between damper and x-plane
α	Unbalanced load angular position according to x-plane
g	Gravitational acceleration
r	Radius of the drum
ω	Angular velocity

The main formula (1) comes from the Newton's second law. Second law says that the net force on a particle is proportional to the time rate of change of its linear momentum.

$$F_{total\_mass} = (m_{unbalanced\_load} + m_{load})\ddot{x} \tag{1}$$

At the first step, we calculate spring force at the x dimension and y dimension. Compression on the spring is showed by t and the angle of the spring with the x plane is showed by θ. According the force of the spring formula (2), we derive (3) and (4).

$$F_{spring} = kt \tag{2}$$

$$F_{spring\_x} = kt\sin\theta \tag{3}$$

$$F_{spring\_y} = kt\cos\theta \tag{4}$$

At the second step, damper force is calculated. Displacement changing on the damper is showed by sign l. Dampers are placed with an angle, so it includes both forces at x and y dimensions. When we decompose the damper forces, we get the forces at x dimension by (6) and y dimension by (7).

$$F_{damper} = c\dot{L} = \mu SP_n sign(\dot{L}) \tag{5}$$

$$F_{damper\_x} = c\dot{L}\sin\beta = \mu SP_n sign(\dot{x}) \tag{6}$$

$$F_{damper\_y} = c\dot{L}\cos\beta = \mu SP_n sign(\dot{y}) \tag{7}$$

Lastly, we calculate the force which is formed by the unbalanced load. This force is called by centrifugal force. It changes by the mass of the unbalance load, speed of the rotating system and radius of the rotating system. Centrifugal formula is represented by (8).

$$F_{unbalanced\_load} = m_{unbalanced\_load} r\omega^2 \tag{8}$$

The equations of motion of the system for the x-plane and y-plane are given in (9) and (10), respectively.

$$\begin{aligned}&(m_{unbalanced\_load} + m_{load})\ddot{x} + 2kx\sin\theta + \mu SP_n sign(\dot{x}) - \\ &(m_{unbalanced\_load} + m_{load})g = m_{unbalanced\_load} r\omega^2 \sin\alpha\end{aligned} \tag{9}$$

$$\begin{aligned}&(m_{unbalanced\_load} + m_{load})\ddot{y} + 2kx\cos\theta + \mu SP_n sign(\dot{y}) = \\ &m_{unbalanced\_load} r\omega^2 \cos\alpha\end{aligned} \tag{10}$$

III. SIMULATIONS

According to theoretical model presented in Section II, a MATLAB simulation of the unbalanced load and the measurement system was built up as shown in Figure 2.

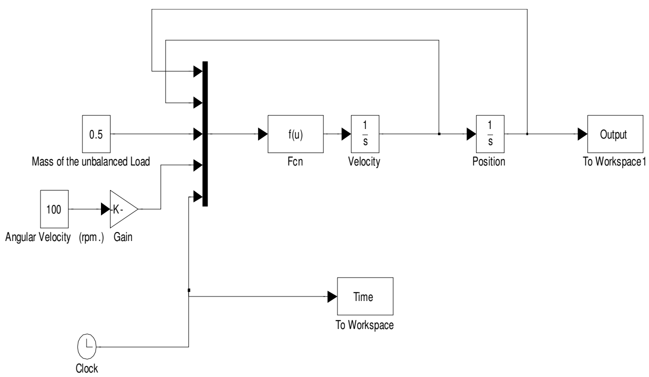

Fig. 2. MATLAB model of the unbalanced load and the measurement system.

In the simulation model, parameters are the angular velocity of the drum and the mass of the unbalanced load. The output is the motion of the drum along the x-axis and y-axis. In Figure 3, simulation result of the case where the drum spins at 100 rpm with a 0.5 kg unbalanced load is presented. In Figure 4, simulation result of the case where the drum spins at 100 rpm with a 1.0 kg unbalanced load is presented.

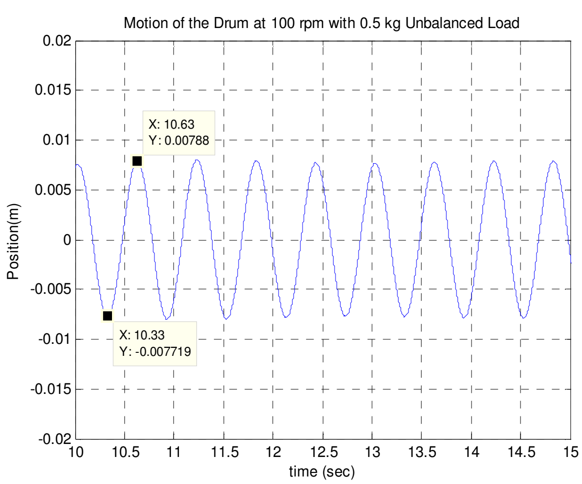

Fig.3. Motion of the drum on the y plane with 0.5 kg unbalanced load at 100 rpm.

By comparing Figure 3 and Figure 4, we can conclude that any increase of the mass of the unbalanced load results increased motion along y-axis of the drum. In fact, doubling the unbalanced load from 0.5 kg to 1 kg increased the oscillation peak value from 0.00788 m to 0.01576 m.

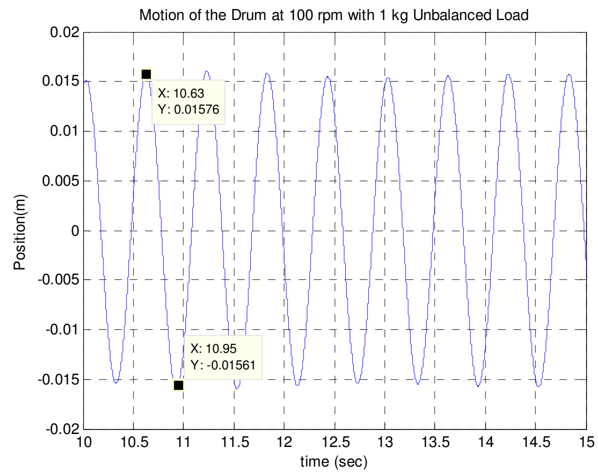

Fig.4. Motion of the drum on the y plane with 1 kg unbalanced load at 100 rpm.

IV. DETERMINATION OF THE ANGULAR POSITION AND MASS OF THE UNBALANCED LOAD

Determination of the angular position of the unbalanced load requires two signals, which are vibration signal and reference signal. Vibration signal is formed by the dynamic motion of the drum. Reference signal is formed by the spinning magnet and the stable hall sensor.

Firstly, we need to determine the zero point on the system. Zero position means that the reference position for the unbalanced load angular position. We put the magnet, hall sensor and the unbalanced load at the same position to get the reference values. Reference values only change by the speed of the drum. At the zero point, we measure the delay time between vibration signal maximum or minimum point and reference signal. Those delay times are our reference values. When the position of the unbalanced load changes, the vibration signal shifts. According to shifted signal, the delay between reference signal and the maximum or minimum point changes. In order to find the angular position of the unbalanced load according to zero point, the shifting time value is divided the period of the signal and then it multiplies with 360. Angular position of the unbalanced load according to maximum point will be obtained by (11). Angular position of the unbalanced load according to minimum point will be obtained by (12).

$$\alpha_1 = \frac{t_{\max} - t_{reference}}{t_{period}} 360^\circ \tag{11}$$

$$\alpha_2 = \frac{t_{\min} - t_{reference}}{t_{period}} 360^\circ \tag{12}$$

where, the time at maximum value of the vibration signal is given as $t_{\max}$. Similarly, $t_{\min}$ is the time at minimum value of the vibration signal. The period of the vibration signal is

t_{period} , and $t_{reference}$ is the reference value of the unbalanced load.

Figures 5 and 6 show the expected vibration signal at the zero and nonzero angular positions, respectively.

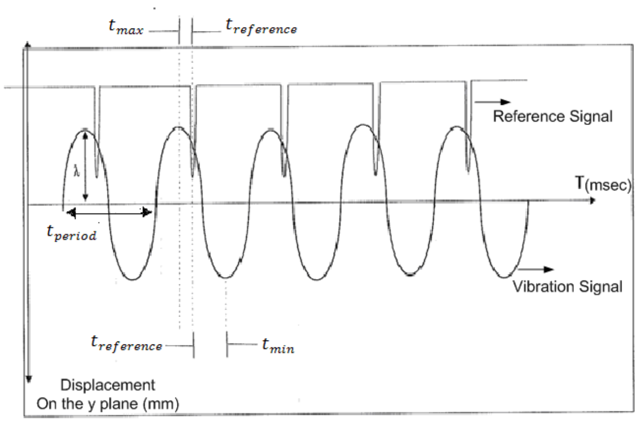

Fig. 5. The expected vibration signal at the zero angular position of the unbalanced load.

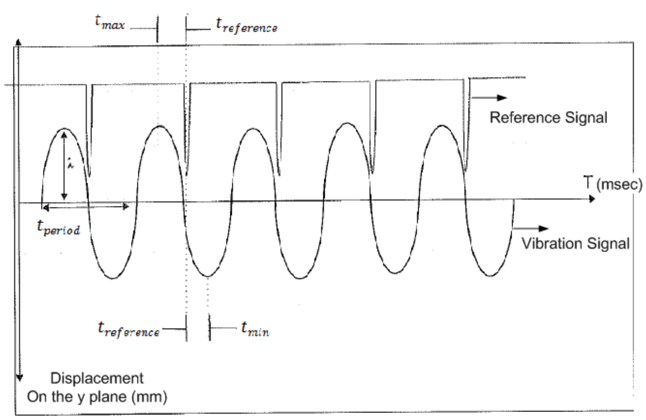

Fig. 6. The expected vibration signal at the nonzero angular position of the unbalanced load.

V. EXPERIMENTS

The system needs to measure the displacement of the drum to determine the unbalanced load angular position. To measure the displacement of the drum, vibration sensor (piezo electric sensor) is used. Secondly, system needs a reference point which means that zero degree for the angular position. Hall sensor and a magnet are used to form zero position which is shown in figure 7.

Piezo electric sensor behaves like a dynamic strain gage except that it generates signal and requires no external power source. The fundamental piezoelectric coefficients for charge or voltage predict, for small stress (or strain) levels, the charge density (charge per unit area) or voltage field (voltage per unit thickness) developed by the piezo polymer. The amplitude and frequency of the signal is directly proportional to the mechanical deformation of the piezoelectric material. The resulting deformation causes a change in the surface charge density of the material so that a voltage appears between the electrode surfaces. When the force is reversed, the output voltage is of opposite polarity. A reciprocating force thus results in an alternating output voltage [7].

Hall sensor changes the output voltage level high to low when it sees the south surface (S) of the magnet. Schematically, the implementation of unbalanced load measurement system is shown in Fig. 7. As shown in this figure, piezo electric sensor output goes to electronic signal conditioning circuit. Both hall sensor output and electronic signal conditioning circuit output go to microprocessor. Microprocessor runs to the algorithm to determine the angular position of the system.

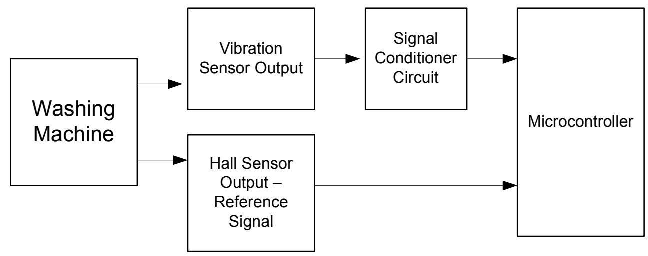

Fig. 7. Implementation of the unbalanced load measurement system.

In the signal conditioning circuit shown in figure 8, we have three main goals. First one is to amplify our signal to sense the low vibrations. Secondly, we aim at filtering the 50 Hz. noise signal with low pass filter. Our sensor behaves like antenna, so it collects the noise too much. Lastly, we shift our signal to 2.5V to get sinusoidal signal which oscillates between 0V and 5V. Microcontroller analog to digital converter can understand the analog signals between 0V and 5V.

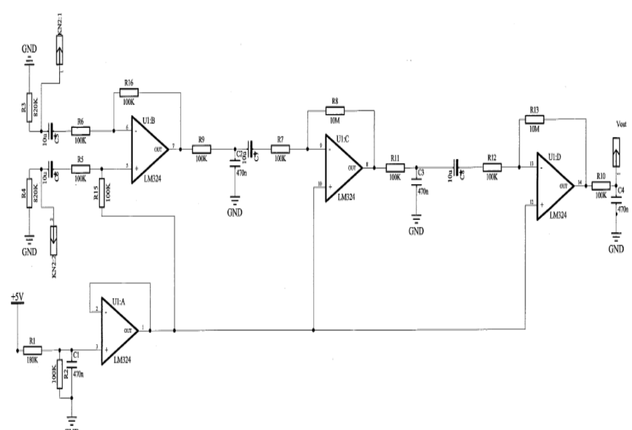

Fig. 8. Electronic circuit of the piezo electric sensor: The circuit amplifies, filters and conditions the signal.

An experimental setup as shown in Figure 9 was built using a horizontal washing machine. The magnet, hall sensor and the vibration sensor are placed to collect data from the washing machine.

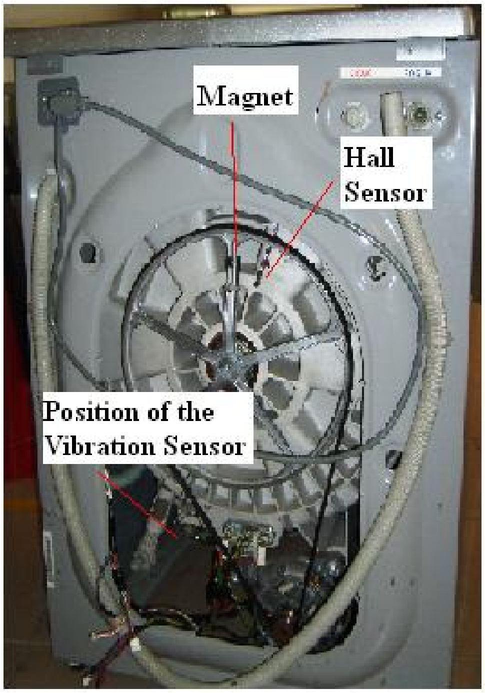

Fig. 9. The experimental setup showing the washing machine and the sensors.

Various experiments were performed to determine the unbalanced load and its angular position. In the first experiment, the drum spins at constant 150 rpm and the algorithm estimates the angular position as well as the mass of the unbalanced load. The Figure 10 shows the sensor signals obtained during the experiment. We repeated the same experiment by placing unbalanced load at different locations on the drum as presented in Table II, also by varying the unbalanced load mass as presented in Table III.

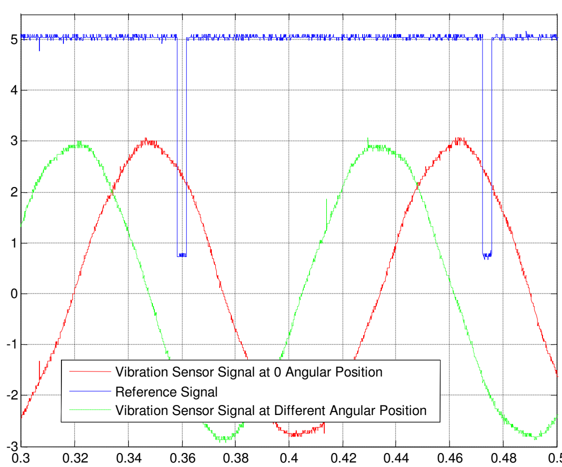

Fig. 10. Signals obtained during the experiment; the vibration sensor signal at zero (reference) position, the sensor signal obtained at a different angular position, and the reference signal.

TABLE II

ANGULAR POSITION DETERMINATION AT 150 RPM

Angular Position (Real)	Phase Shift	Period	Angular Position (Calculated)	Absolute Error
0º (Reference Point)	60ms	398ms	0º	0º
60 º	134ms	398ms	67º	7º
120º	192ms	398ms	118º	2º
180º	261ms	402ms	174º	6º
270º	360ms	400ms	271º	1º

TABLE III

LOAD DETERMINATION AT 150 RPM

Mass of the unbalanced load at 150 rpm.	Vibration Sensor output voltage amplitude
200 gr	320mV
400 gr	612mV
800 gr	1.31V
1000gr	1.58V

The effect of the mass of the unbalanced load on the vibration sensor output is shown in Figure 11 (Table III). The relation between the unbalanced load and the vibration sensor output is almost linear. By determining the slope, one can easily determine unknown unbalanced load directly from the sensor voltage readings.

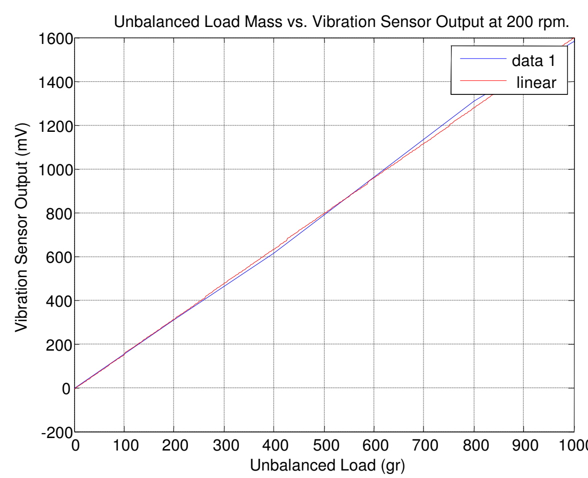

Fig. 11. Analyze of the calculated vibration sensor output with the different mass of the unbalanced load at 200 rpm.

In order to analyze the effects of the increased drum speed, the same experiment were repeated at 250 rpm. Table IV presents the angular position determination experiments, and Table V presents the load determination as the load is varied while keeping the drum velocity constant, at 250 rpm.

TABLE IV

ANGULAR POSITION DETERMINATION AT 250 RPM

Angular Position (Real)	Phase Shift	Period	Angular Position (Calculated)	Absolute Error
0º (Reference Point)	38ms	240ms	0º	0º
60 º	86ms	242ms	72º	12º
120º	121ms	240ms	124º	4º
180º	158ms	242ms	180º	0º
270º	216ms	240ms	268º	2º

TABLE V

LOAD DETERMINATION AT 250 RPM

Mass of the unbalanced load at 250 rpm.	Vibration Sensor output voltage amplitude
100 gr	350mV
200 gr	705mV
400 gr	1.34V
600gr	2.1V
800 gr	2.9V

In Figure 12 we can see that the effect of the mass of the unbalanced load on the vibration sensor output as shown in Table V. The relation between the unbalanced load and the vibration sensor output is almost linear again. Comparing Figures 11 and 12, we see that any increase of the spin speed results a larger slope. A 2D loop-up table can be formed with different rpm and unbalanced load values to estimate the unbalanced load from sensor readings directly.

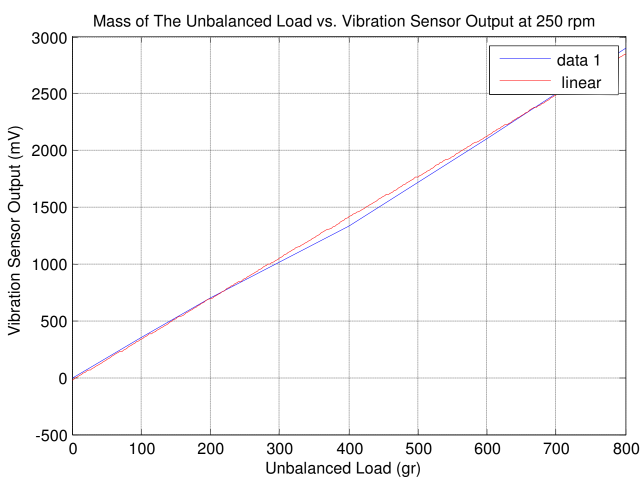

Fig.12. Analyze of the calculated vibration sensor output with the different mass of the unbalanced load at 250 rpm.

According to our experimental results, analyzing the y-plane vibration by using piezoelectric sensor is enough for the estimation (the vibration on the x-plane has not been considered). This approach has been determined as the cheapest and simplest way to find the unbalanced load magnitude and location with acceptable accuracy.

VI. ACKNOWLEDGMENTS

We would like to thank Arçelik A.Ş. for funding and supporting this project.

VII. CONCLUSION

In this paper, we proposed an approach to evaluate the angular position and mass of the unbalanced load for washing machine applications. Unbalanced load in washing machines limit the performance, and therefore accurate estimation of unbalanced loads, including their magnitudes and locations in washing drum, is required. There is a trade-off between using multiple sensors and cost. In this study, a simple and cheap system with acceptable estimate accuracy has been proposed. A simulation model of the system was developed and various experiments were performed. The proposed algorithms and the developed experimental system can estimate the unbalanced load as well as the angular position of the unbalance load with maximum ± %3 error. In other words, position is calculated with a maximum 12º error at 360º. Developed algorithms and the determination system work properly and also theoretical and experimental results are matched. The determination and elimination of the unbalanced load increase the performance of the washing machines; therefore, this research can be used on future studies on unbalanced load control and elimination to develop more effective and secure washing machines.

REFERENCES

[1] Y. Yuan, A. Buendia, R. Martin and F. Ashrafzadeh, "Unbalanced Load Estimation Algorithm Using Multiple Mechanical Measurements for Horizontal Washing Machines", IEEE Sensors 2007 Conference, pp 1033-1306.

[2] Y. Yuan, "Sensor Fusion based Testing Station for Unbalanced Load Estimation in Horizontal Washing Machines", Res. & Eng. Center, Whirlpool Corp., Benton Harbor, MI; 2008 IEEE Instrumentation and Measurement Technology Conference Proceedings 2008, pp. 1424-1428, 12-15 May 2008.

[3] E. Papadopoulos, I. Papadimitriou, "Modeling, Design and Control of a Portable Washing Machine during the Spinning Cycle", IEEE/ASME International Conference on Advanced Intelligent Mechatronics, AIM 01/01/2001, Vol.2, p.899-904.

[4] Y. Sonoda, H. Yamamoto, Y. Yokoi, "Development of the Vibration Control System "G-Fall Balancer" for a Drum type Washer/Dryer", in proceedings of 2003 IEEE/ASME International Conference on Advanced Intelligent Mechatronics, (0-7803-7759-1), 20-24 July 2003. Vol.2;p.1140.

[5] J. Chung, "Dynamic Analysis of an Automatic Dynamic Balancer For Rotating Mechanisms", Journal of Sound and Vibration, December 16 1999. Vol.228, Iss.5; p.1035-1056.

[6] Z. Yanqing, W. Zhiyan, Z. Yiqi, "An Active Control Strategy of Vibration Attenuation System Based on Nonlinear Decoupling", Industrial Electronics, 2003. ISIE '03. 2003 IEEE International Symposium on (0-7803-7912-8), 9-11 June 2003. Vol.1; p.448.

[7] Techman Catalog, Measurement Specialist, [Online]. Available: http://apps.meas-spec.com/myMeas/download/pdf/english/piezo/techman.pdf