academicJournals

Vol. 8(29), pp. 1396-1406, 4 August, 2013
DOI 10.5897/SRE2013.5578
ISSN 1992-2248 © 2013 Academic Journals
http://www.academicjournals.org/SRE

Scientific Research and Essays

*Full Length Research Paper*

# A computational approach to analyze unbalancing in rotational systems

**Bruna T. V. Silva[1], Álvaro M. S. Soares[2,3]*, Mauro H. Mathias[3], Pedro P. L. Prado[4], João B. Gonçalves[4]**

[1]Engineering College of Resende (FER – AEDB) – Resende – Brazil.
[2]Department of Mechanical Engineering, University of Taubate, Taubate, Brazil.
[3]Department of Mechanics, UNESP – Univ Estadual Paulista, Guaratinguetá, Brazil.
[4]Department of Electrical Engineering, University of Taubate, Taubate, Brazil.

Accepted 25 June, 2013

**Rotational systems are widely used in industrial application and its monitoring can improve production and decrease stop times of machines, for example. In industry these kind of problem is, in general, solved using a vibration meter and a phase meter in order to obtain the necessary data to correct the rotor unbalancing. This work presents the development of a computational application, using LabVIEW, for improving the measure of unbalancing in a rotational system. The application developed in this work proposes that the phase and vibration be digitally measured. The idea is to use a piezoelectric accelerometer to measure vibration intensity and an inductive sensor to replace the phase meter. An algorithm to calculate phase angle is also described. With the application developed we can replace a vibrometer by a PC compatible computer that can calculate the unbalancing of a rotor too. In order to evaluate the results a test bench composed by a DC motor (1585 rpm and 0.19 HP) coupled to a shaft made of steel with a rotor on it, were assembled. In the test bench we simulated an unbalancing rotor and use the application to calculate the balancing mass and its phase angle. Preliminaries results shown a good agreement between the real mass and phase angle and the one calculated by the application.**

**Key words:** Rotational systems, unbalancing, LabVIEW, instrumentation.

## INTRODUCTION

Nowadays maintenance has been a very important subject into industry concerns. The idea of maintaining the machineries in a good state of preservation for a long period of time has received much attention once this idea brings improvement to the production, budget reduction, increase of profits and so on. When we talk about maintenance we should keep in mind that there are three major important kinds of ways to do it. The first one is corrective maintenance. This way should be avoided at any cost, once it implies that a fault problem happened. It is used when it's impossible to predict or prevent a failure. The second way is the preventive maintenance which proposes to repair the equipments before failure occurs, respecting the nominal lifetime of their components. In this kind of maintenance, components which are still in good conditions should be exchange because their useful life has been reached. The third major way is the predictive maintenance. This kind of maintenance is based on the actual conditions of a component which implies in monitoring components in order to keep them working until a predicted condition is reached. In the current literature we can note many researchers treating this kind of matter based on predictive maintenance of rotating machines. In Zhou and Shi (2001), the authors

*Corresponding author. E-mail: alvaro@unitau.br

present a review of the research work that was performed in real-time active balancing and active vibration control for rotating machinery, as well as the research work on dynamic modeling and analysis techniques of rotor systems. The basic methodology and a brief assessment of major difficulties and future research needs are also provided.

In Huang et al. (2009), the authors state that rotor vibrations caused by rotor mass unbalance distributions are a major source of maintenance problems in high speed rotating machinery. Minimizing this vibration by balancing under practical constraints is quite important to industry. So, the authors propose the balancing of two large industrial rotor systems by constrained least squares and min-max balancing methods. They also state that in current industrial practice, the weighted least squares method has been utilized to minimize rotor vibrations for many years, but the min-max balancing method can guarantee a maximum residual vibration value below an optimum value and they shown by simulation that it significantly outperform the weighted least squares method. In Swanson et al. (2005) the authors present a practical understanding of terminology and behavior based in visualizing how a shaft vibrates. With a simple text, they also examine issues that affect vibration in order to help the nonspecialist to have a better understand what is going on in the machinery. In Sinha et al. (2002) the authors developed a method which found to have the potential for fast and reliable rotor unbalance estimation. The method just uses measured pedestal vibration from a single run-down of a machine along with *a priori* models of the rotor and fluid bearings but the method accounts for the dynamics of the foundation. The method has been qualified on an experimental rotating rig and its sensitivity analysis was carried out and presented in the paper. In Gonçalves and Silva (2011) the authors address Predictive Maintenance techniques with oil analysis and vibration analysis. Those techniques, according to the authors, are the most important for monitoring some systems. The integration of these techniques has the potential to revolutionize industrial practices and provide a large economic gain for industries. To study the integration of both techniques the authors proposes the set up of a bench test to put to work to the extreme limit of use.

Tests were carried out with the lubricant recommended by the manufacturer of the equipment, using lubricants supplemented with various percentages of liquid contaminant and lubricants supplemented with several percentages of solid contaminant. The paper presents the results of the first test, that is, with the oil recommended by the manufacturer in extreme conditions. From the results the authors observed that if in a system an abnormal occurrence takes place, for example an extra load during a certain period of time, the lubricant analysis can be used together with the vibration analysis to complement it. In Siqueira et al. (2012), the authors explain that linear parameter varying (LPV) control is a model-based control technique that takes into account time-varying parameters of the plant. In the case of rotating systems supported by lubricated bearings, the dynamic characteristics of the bearings change in time as a function of the rotating speed. Hence, LPV control can tackle the problem of run-up and run-down operational conditions when dynamic characteristics of the rotating system change significantly in time due to the bearings and high vibration levels occur. In their work, the authors present the LPV control design for a flexible shaft supported by plain journal bearings. In this paper we address the problem of developing a computational tool to give support to the techniques described in literature. To do so we also mount a bench test composed by a DC motor (1585 rpm and 0.19 HP) coupled to a shaft made of steel with a rotor, supported by two bearings, on it. In the bench we can simulate unbalancing and use the software application developed to calculate the balancing mass and its phase angle. The test bench was completely automated and has an electrical/electronic circuit controlled by computer that turns the motor on and off according to the test that will be performed.

This paper is organized as follows. First is a brief description of the test bench designed to perform the tests and develop the application. Next is a summary of the calculation of unbalancing in rotors. This is followed by a description of the software environment, the results and conclusion as well as suggestions for some further work.

## THE TEST BENCH

Figure 1 shows the test bench mounted during the development of this work. The bench was assembled on a steel board (1) and is composed by a DC motor (6) which operates in 220 V and have a constant rotation of 1585 RPM (26.41 Hz) coupled to a steel shaft (3) by a flexible coupling (4), and a rotor (5) which is supported by two bearings (2). In order to have an automated system a contactor (7) triggered by a relay commanded by a data acquisition board is also part of the test bench. The contactor is used to turn the DC motor on and off according with the test that will be performed. The items numbered as 8 in the Figure 1, are a piezoelectric accelerometer (PCB Piezotronics model 353B18) and an inductive sensor used in tests of unbalancing machines. Also, three more elements were used to complete the experimental setup. A computer PC compatible, a data acquisition board (NI USB-6009 from National Instruments, 8 analog inputs (14-bit), 48 kS/s, 2 analog outputs (12 bit), 12 digital IO) and the software LabVIEW 8.5.

### The test bench operation

The test bench was designed to be modular. This means that we want to have an experiment that could be used in several kinds of tests, involving rotational systems. So, we have the freedom of changing the rotor, to add another rotor to work with unbalancing in two planes, change the bearings to simulate faults in bearing's spheres and so on. This application, specifically, works with unbalancing of rotors in one plane and was assembled with one rotor which has 6 (six) holes equally spaced on it that allow

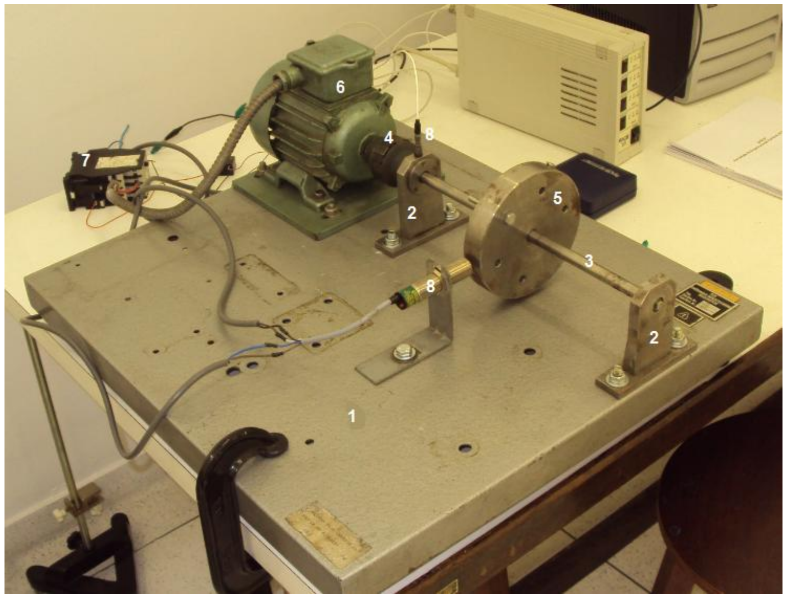



**Figure 1.** The experimental test bench.

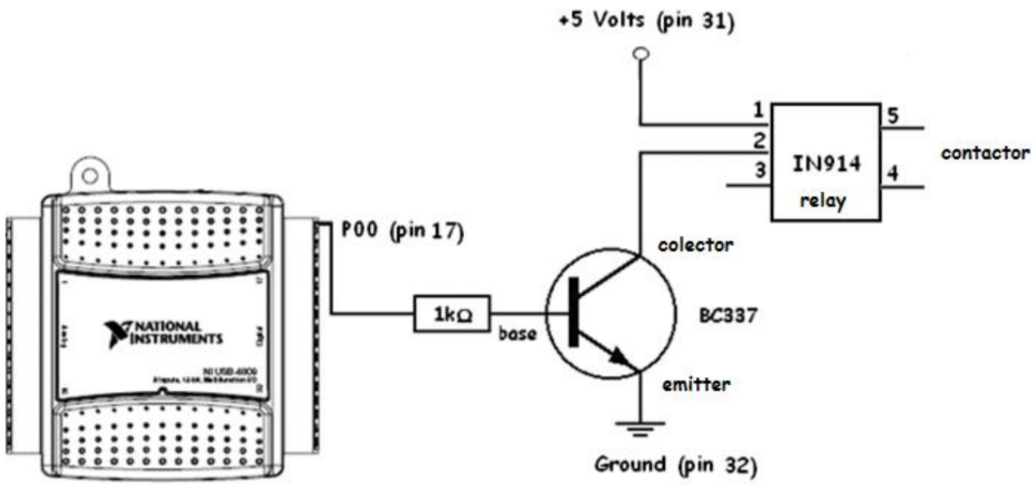



**Figure 2.** The electric/electronic circuit.

simulating unbalancing. One accelerometer to measure the vibration on the bearings support and an inductive sensor to measure the rotor angular's velocity. The electric/electronic circuit was idealized to give safety to the tests. Using the digital IO available in the data acquisition board we can, using a relay, to turn the motor on and off. With this feature the operator can design the test, to turn on the application, to give a step away from the test bench and wait for the test's finish. Figure 2 shows a scheme of the coupling between the data acquisition board, the relay and the contactor. The electric/electronic circuit is very simple and uses a transistor (BC337) a relay and a resistor to interface the data acquisition board with the test bench.

## CALCULATING THE UNBALANCING OF ROTORS

The static unbalance occurs when the principal axis of inertia of a rotor is displaced, however parallel to the axis of rotation. The distance between the center of gravity and the axis of rotation, the eccentricity causes the centrifugal force. Figure 3 represents this type of unbalance.

In this case the unbalanced mass and center of gravity are in the same plane, normal to the axis of rotation. A rotor with two equal unbalanced masses which are equidistant with respect to the center of mass also features a static unbalance, since the effect of both is equivalent to the effect of a mass located in the plane center of gravity. To eliminate the static unbalance we must move the center of gravity toward the axis of rotation of the rotor. Adding or removing rotor mass, the correction is performed, because the radial force caused by this change will be equal in magnitude to the force caused by the eccentricity of the center of gravity, but in the opposite direction. The procedure of balancing on a plane can be done using the vector method that consists in performing the following steps: (a) Measurement of the initial vibration; (b)

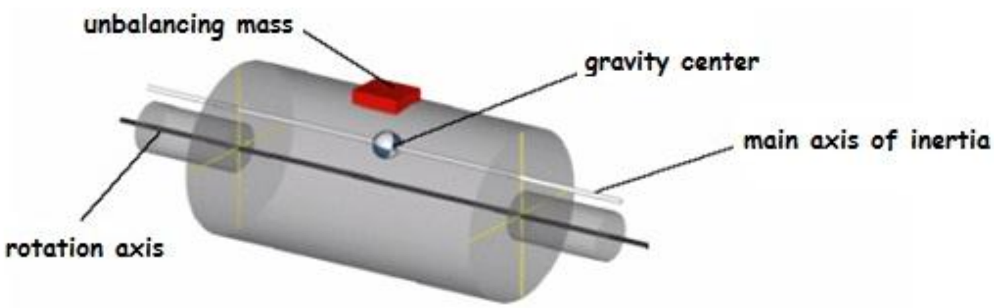



**Figure 3.** Static unbalancing.

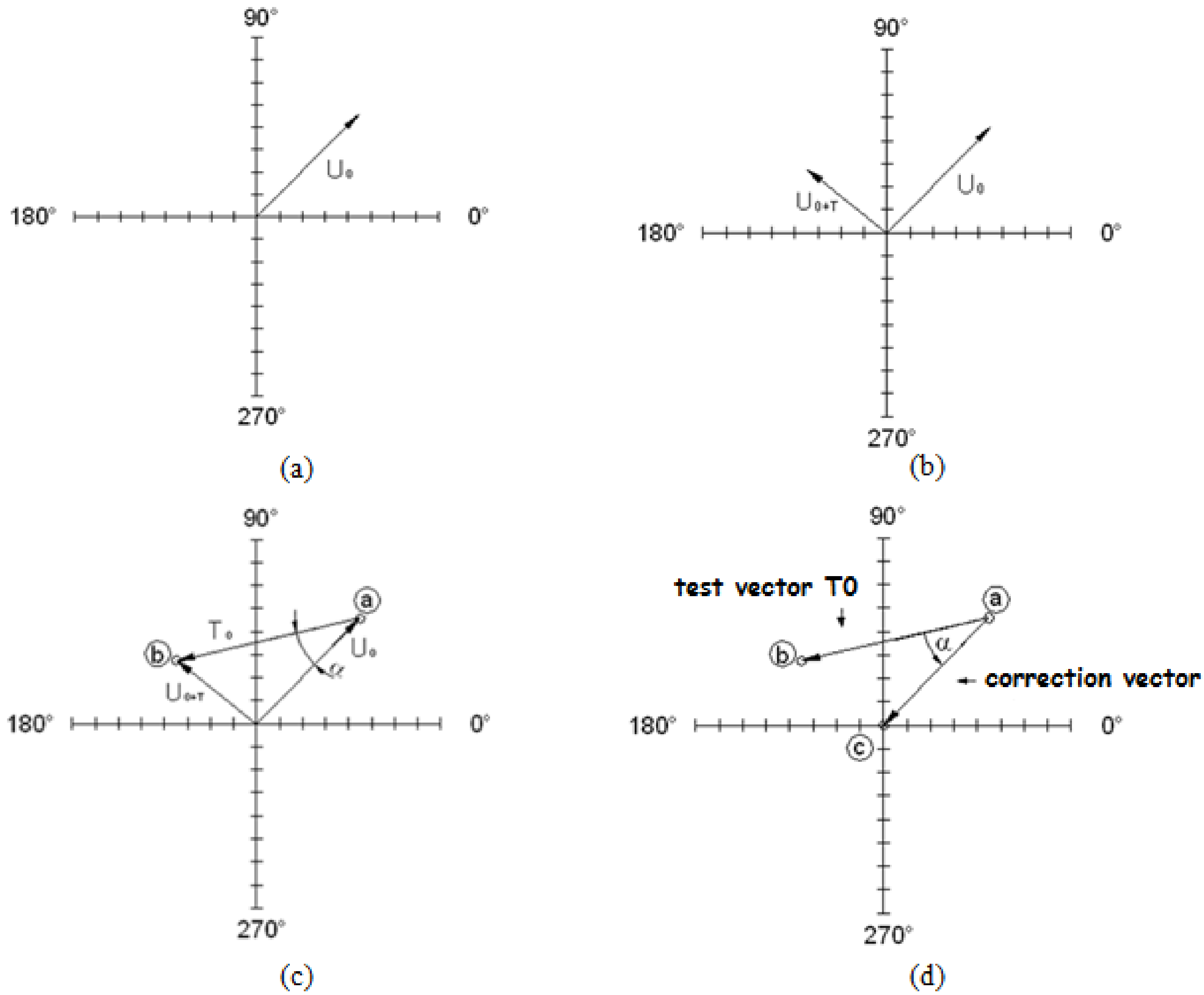



**Figure 4.** Vector method to balancing rotors.

Installation of a trial mass and measurement of the vibration again; (c) Definition of mass correction and angular position for mounting it; (d) Assembling of the correction mass and start a new vibration measurement. If the new vibration data is within an acceptable range, then the balancing job is complete. Those steps are depicted in Figure 4.

In the first step, by measuring the vibration, we determined initial unbalance and the phase angle of the vibration stages. From these values we plotted, on the vector diagram, the unbalanced vector $U_0$ which has module equal to the maximum amplitude of vibration signal, as shown in Figure 4(a). In the second step we assemble a test mass in the balancing plane of the rotor and the vibration, caused by this new unbalance, is measured again and so is its phase angle. We obtain then the vector $U_{0+T}$ in the diagram shown in Figure 4(b). From the obtained vectors, we draw the vector $T_0$, which starts at vector $U_0$ and finishes at $U_{0+T}$, since the change in the imbalance occurred in this direction. This vector is shown in Figure 4(c). From the measurements performed, we calculate the correction mass, shown in Equation 1 and we define the angular position found for this mass assembly.

$$Mc = \frac{U_0}{T_0} \times Mt \qquad (1)$$

In Equation (1), *Mc* is the correction mass; $U_0$ is the module of the vector of initial vibration; $T_0$ is the module of the vector test and *Mt* is the test mass. To complete the process of defining the angular position, we must move the test vector, so that it opposes the original unbalance vector, rotating it in the direction of the angle α, shown in Figure 4(d). The procedures described in this part of the work were implemented in the application using LabVIEW software. This implementation will be explained in the next part of this work.

## THE SOFTWARE ENVIRONMENT

The reason to choose LabVIEW as a software platform is because this software is quite good if you want to work with data acquisition and treatment. LabVIEW, developed by National Instruments, works with a graphical language and contains lots of components build in that minimize the programming work. We also used hardware from National Instruments (a data acquisition board) which has brought us more simplicity. Figure 5 shows a scheme of the experimental test bench integrated with the data acquisition board and LabVIEW.

When we translate the procedure of balancing on a plane to the software applications it should performs the following tasks:

1. Start the motor on, acquire and store data from the sensors;
2. Calculate phase angle and maximum level of vibration;
3. Add a trial mass to the system, start the motor on, acquire and store data from the sensor again;
4. Calculate phase angle and maximum level of vibration again;
5. Calculate the unbalance mass and its position, with the data acquired, according to the vector method;
6. Add the unbalanced mass to the system;
7. Start the motor on, acquire and store data from the sensors in order to check if there is no unbalancing.
8. If the new vibration data is within an acceptable range, then the balancing job is complete.

We tried to keep the application developed as simple as possible. When we work with LabVIEW there are two major screens that we should give attention: the Front Panel and the Block Diagram. The Front Panel is the user screen, where we can see the data output. The Block Diagram contains the code developed. In Figure 6 we show part of the Block Diagram developed.

The components in the first rectangle at left have two LabVIEW components. The component superior represents a link to a digital port of the data acquisition board and turns the motor on when the application starts. The component "Time Delay" holds the motor on for 2 s before we start the data acquisition to wait the system reach at a steady state. In the middle rectangle is where the data acquisition occurs. It is acquired and filtered with a band-pass digital filter adjusted with 15 Hz and 35 Hz frequencies, in order to improve the reading of the motor DC frequency (26.41 Hz) and eliminate frequencies that are not important. After that the data is stored. Figure 7 show a plot of the data acquired with the system without any unbalancing. The superior plot shows the inductive sensor's data. This sensor outputs 5 V each time the rotor turns 2π rd. The inferior plot is relative to the accelerometer and gives the vibration level of the test bench. The maximum vibration level, in this case, is around 0.52793 V.

The second part of the Block Diagram developed is show in Figure 8. This part is a subroutine which reads the files recorded in hard disk and calculates phase angle and the maximum vibration peak. Those results are used to calculate the unbalance mass and its position, in order to correct the unbalancing of the system.

The calculus of the maximum vibration peak is quite direct. We just need to choose the biggest element of a vector. In the other hand to calculate the phase angle, we initially locate the maximum vibration peak, let's say in position $\mathbf{p_3}$, from the accelerometer data Now we get the period of rotation from the inductive sensor data and sum and subtract this value from position $\mathbf{p_3}$. After that we should have a set of points located, as show in Figure 10.

The next step is to locate the two peaks ($\mathbf{p_1}$ and $\mathbf{p_2}$) of the inductive sensor before and after the maximum vibration peak. Initially we calculate the position of the maximum value of the vector that contains the inductive sensor (***max(si)***). Next, we calculate the maximum value of the following dummy vector (***max(y)***):

$$y_k = (k - \max(si))^2 \,.\, si_k ; \quad k = 1, 2, \cdots \# \quad of \quad samples \qquad (2)$$

So the points ***max(si)*** and ***max(y)*** are the point $\mathbf{p_1}$ and $\mathbf{p_2}$ show in Figure 9. A plot of the vector y is shown in Figure 10.

The phase angle denoted by φ, in rd, is calculated according with Equation (3).

$$\varphi = \frac{(p_3 - p_1)}{(p_2 - p_1)} \times 2\pi \qquad (3)$$

Once we can calculate magnitude and phase angle of a data set, the third part of the software applications calculates the correction mass and its position (Figure 11). With those values we can check if the new vibration data is within an acceptable range, if it's, then the balancing job is complete (step 8, 'the software environment' part of this work).

The block called pos&mass that appears in Figure 11 is shown in Figure 12. It implements the calculus of a sum of vector. The vectors that are summed are $U_O$ and $U_{O+T}$ from Figure 4.

The block called mag&phase was already shown in Figure 8.

## RESULTS AND DISCUSSION

After finish the software application, we start to carry on the steps depicted in calculating the unbalancing of rotors part of this work. The rotor used in the test bench have 6 (six) symmetrical holes (Figure 13 (a) and (b)) that allows us to simulate unbalancing. We expect that the test bench have a little structural unbalancing, which was neglected.

Figure 13 (a) shown a scheme of the rotor. It turns in anti-clockwise direction, from point (1) to (2). In point 1, indicated in Figure 13(a), we put a metallic part to mark the zero degree point. In order to disturb the system we simulate unbalancing in the rotor with a known trial mass. In the first test we used an unbalancing mass of 24.07 g positioned at 60° (positive, clockwise direction) in the

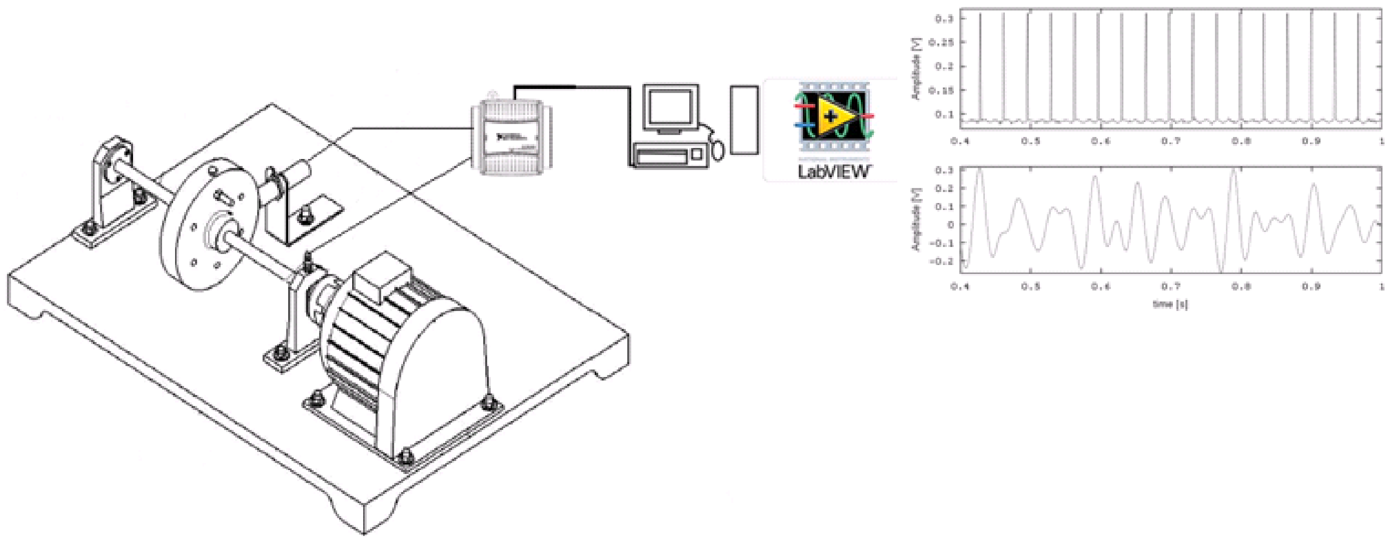


**Figure 5.** Scheme of the experimental test bench with LabVIEW.

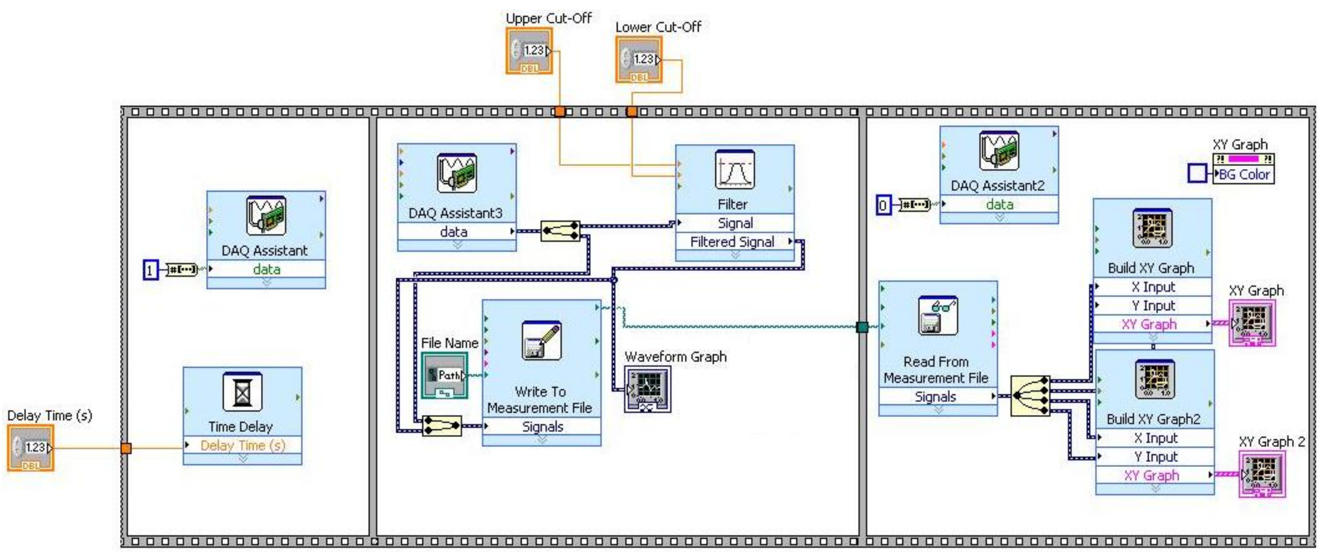



**Figure 6.** LabVIEW Block Diagram – Data acquisition and storage.

rotor (in point 2 in Figure 13(a)). The trial mass used had 14.79 g (in point 1 Figure 13(a)) and the software application calculates a correction mass of 21.4274 g positioned at -127.27°. These result shown that we should put a correction mass of 21.4274 g in an angular position of -127.27°. If we analyze this position, it will lead us to a point very close to point 5 in Figure 13(a). Point 5 is diametrical opposed to point 2, where the unbalancing mass, of 24.07 g was.

Figure 14 shown the comparison between the three states of the system, unbalanced (blue), unbalanced with trial mass (green), balanced after addition of the correction mass (red). Also, in Figure 14 the maximum vibration peaks are 0.82748, 0.88809 and 0.44356 respectively indicating that the vibration drops to a value almost close to half of the initial unbalancing. Table 1 illustrates two other tests performed with the test bench.

After analyze Table 1 we can conclude that the software application reach at quite good results. The maximum error in angular position was 4.4 and 10.98% for correction mass. Those error can be explained because the structural unbalancing of the test bench and because with this rotor (shown in Figure 13) we were able to put the unbalancing mass in any position. As an improvement of the project we intend to modify the rotor by design a proper grove along it circumference in

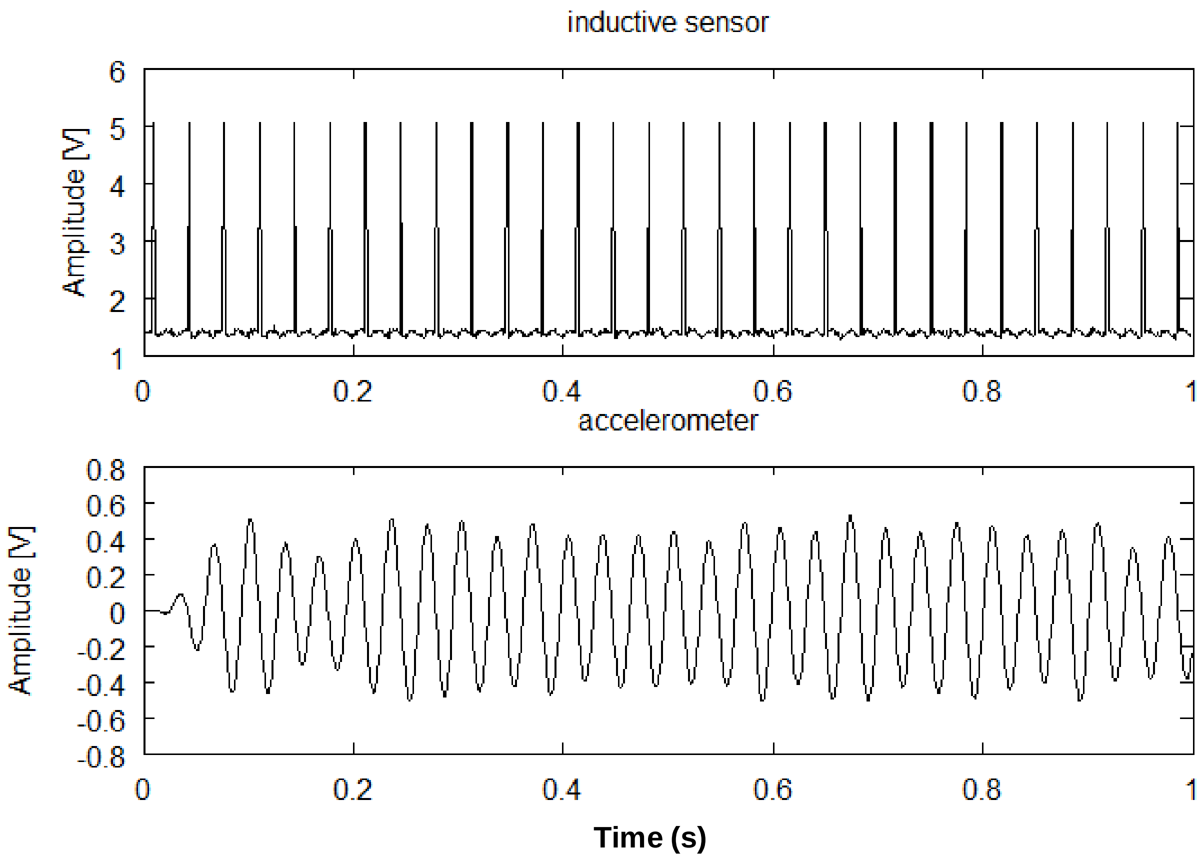



**Figure 7.** Data acquired.

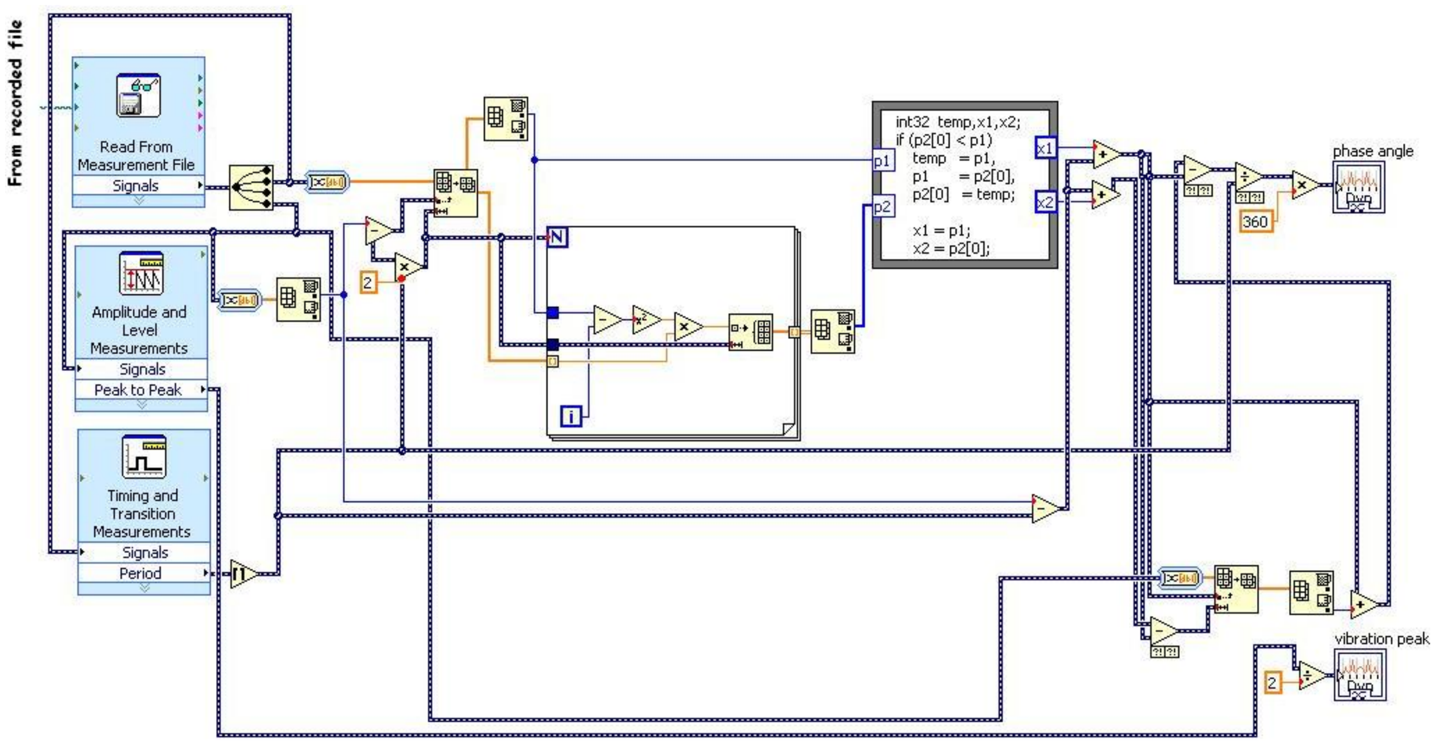



**Figure 8.** Subroutine to calculate phase angle and maximum vibration peak.

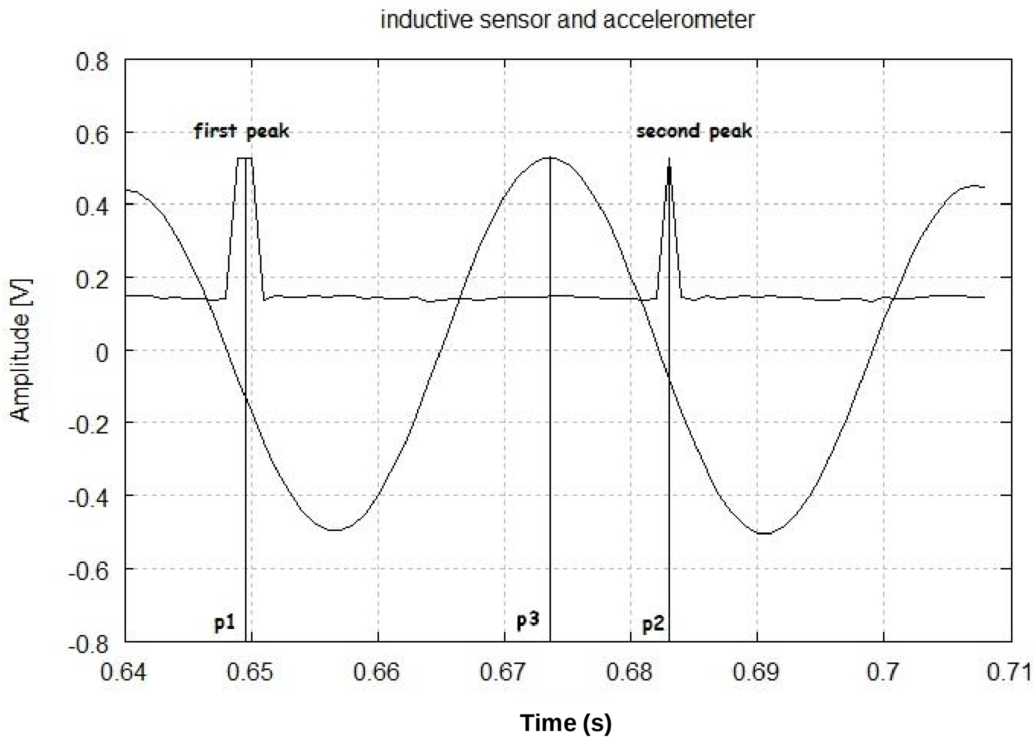



**Figure 9.** Calculating the phase angle.

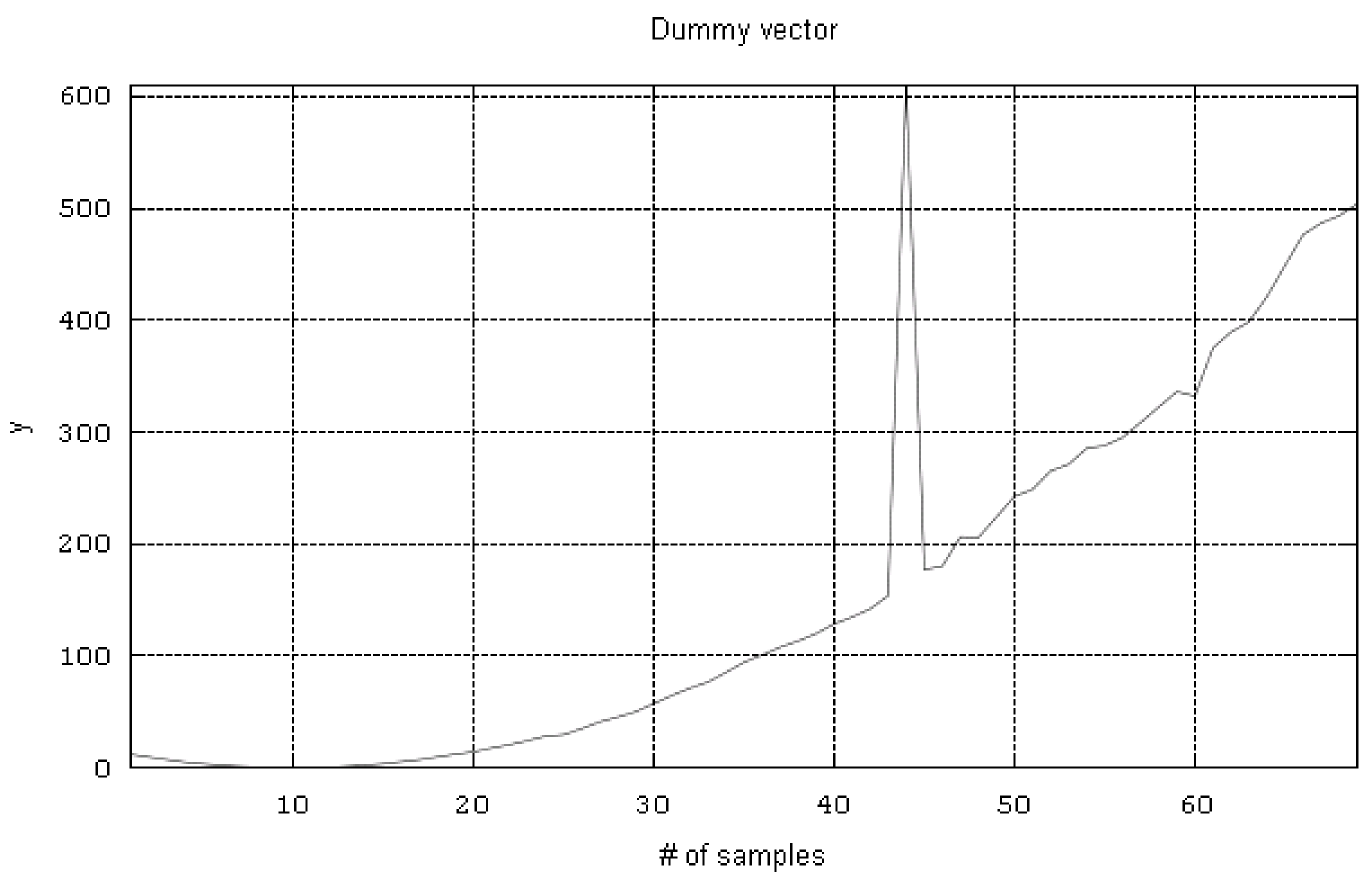



**Figure 10.** Dummy vector (y).

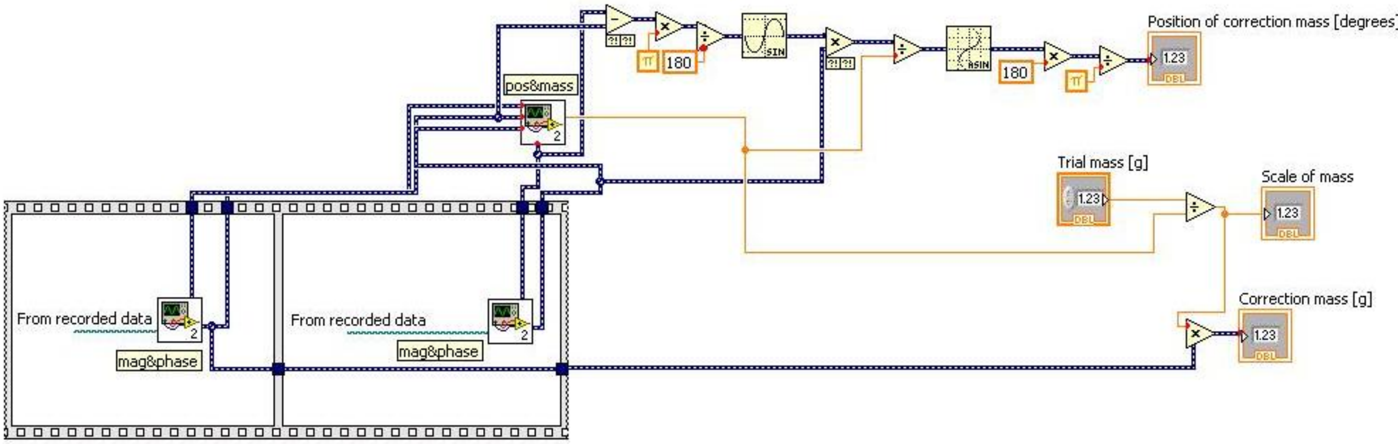



**Figure 11.** Calculus of correction mass and its position.

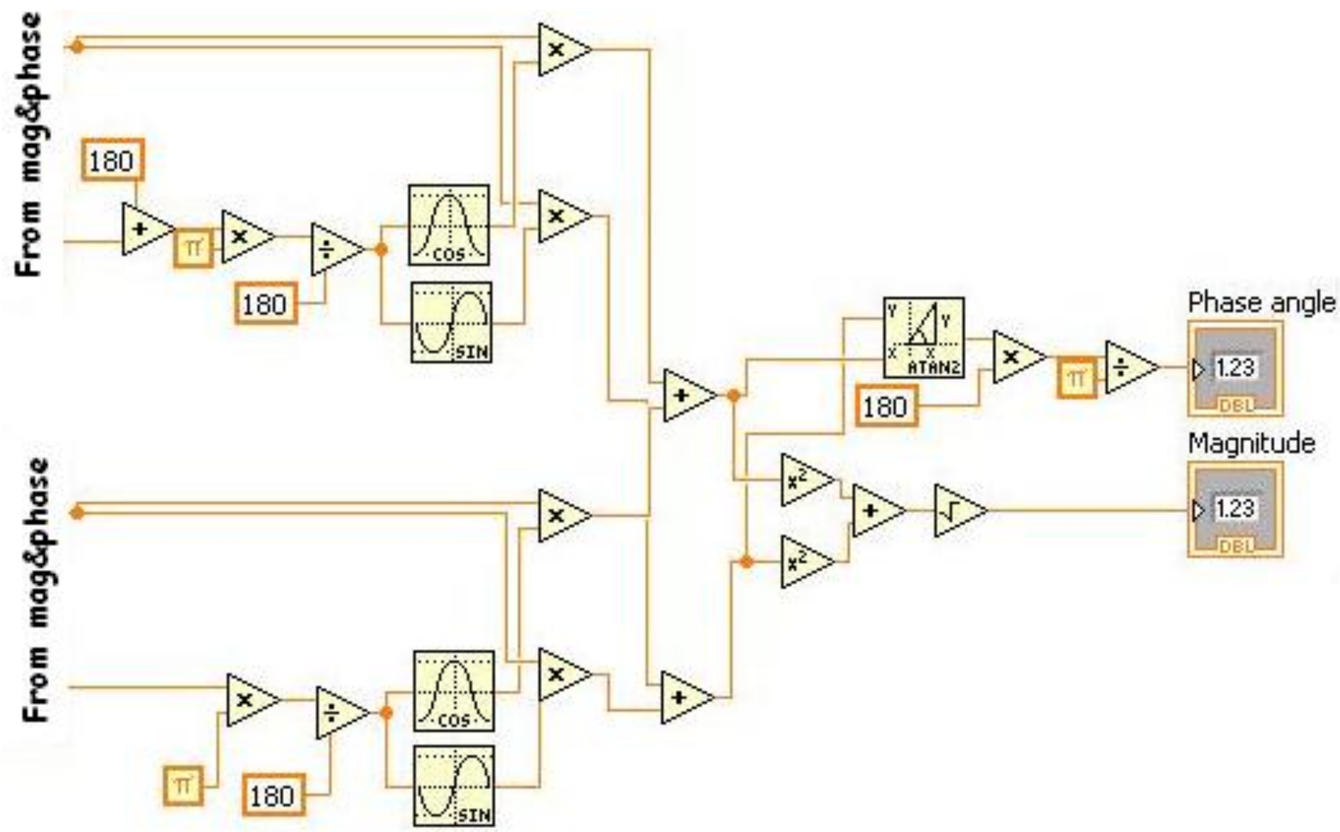
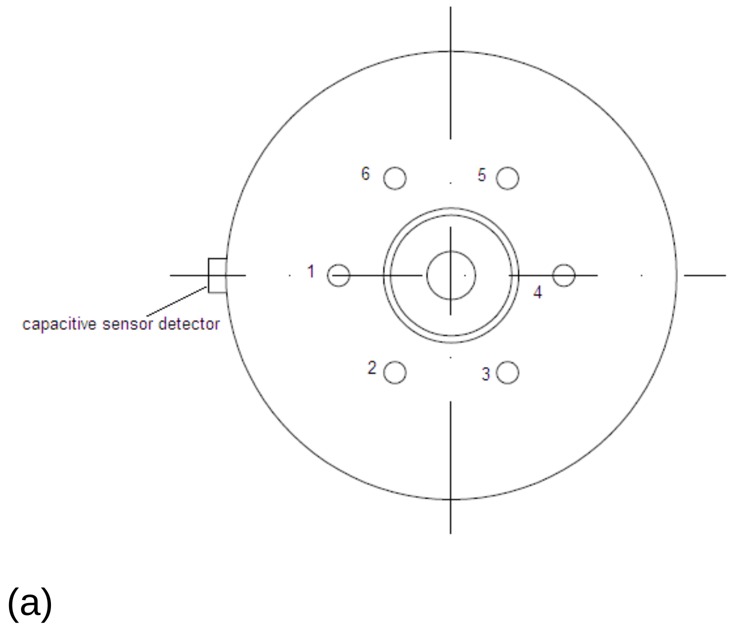



**Figure 12.** Calculus of magnitude and phase angle.

order to allow the unbalancing mass be put in any angular position.

## Conclusion

This work propose a software application, developed using the graphical language LabVIEW with the purpose of analyze faults in rotational systems. The work involved the development of an automated test bench full controlled by a PC computer which allows the simulation of faults in such systems. With the application running it can be used as a digital vibrometer. The fault treated here was the unbalancing of rotors. The software



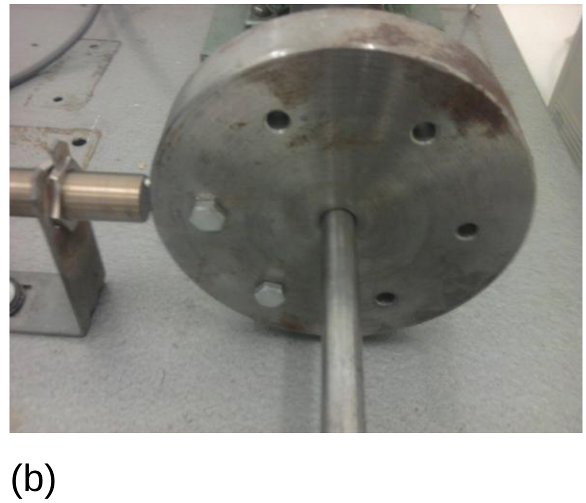




**Figure 13.** Holes in the rotor to simulate unbalancing.

**Table 1.** Experimental results.

| Case # | Simulated unbalancing at 60° | Trial mass (g) | Correction mass (g) calculated by the application | Angular position (º) calculated by the application |
|---|---|---|---|---|
| 1 | 24.07 g | 14.7 | 21.4279 | -127.27 |
| 2 | 14.79 g at 60° | 20.07 | 13.8208 | -128.00 |
| 3 | 14.79 g at 60° | 10.45 | 15.8958 | -117.89 |

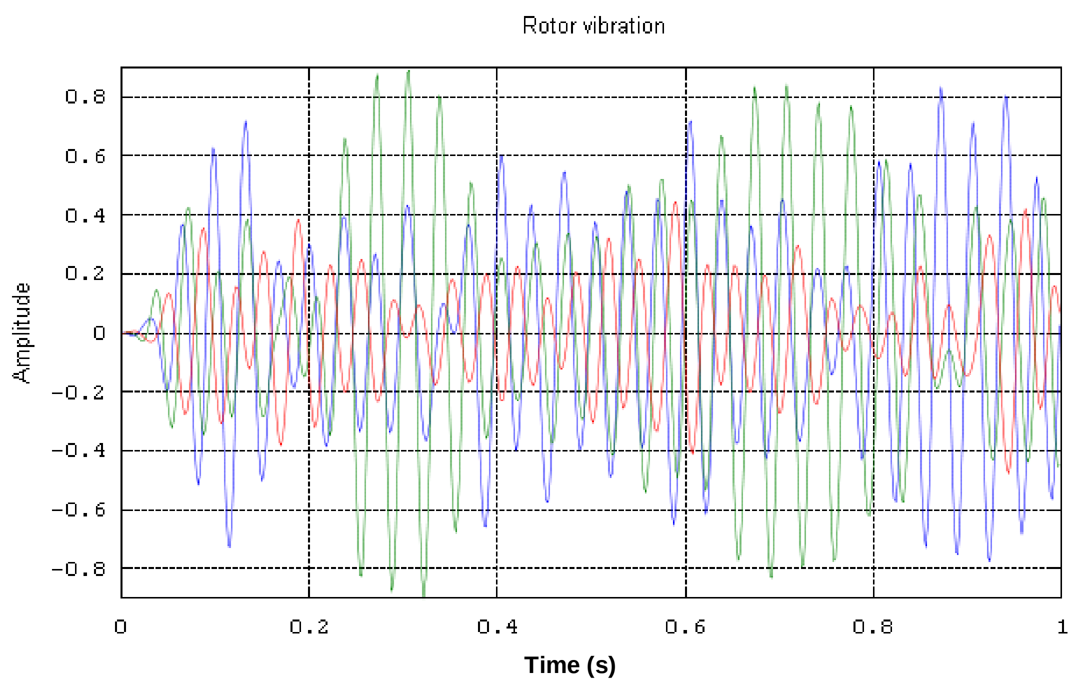



**Figure 14.** Comparison between the rotor's vibrations in three different states.

application worked quite well with reduced errors in unbalancing mass and positioning angle calculations, pointing to positions diametrical opposed to the simulated unbalancing mass. The work is in progress and for further works we intend to use the test bench to simulate fault in bearings and in bearings and rotors simultaneously. We also intend to improve the software to train a neural network in order to detect those faults.

## ACKNOWLEDGMENTS

The authors gratefully acknowledge the support of the School of Mechanical Engineering, University of Taubaté (UNITAU), Taubaté, São Paulo, Brazil and Mr. Lamarino Vendramini for his support and help during the development of the project.

## REFERENCES

Gonçalves AC, Silva JBC (2011). Predictive maintenance of a reducer with contaminated oil under an excentrical load through vibration and oil analysis. J. Braz. Soc. Mech. Sci. Eng. 33:1-7.

Huang B, Fujimura D, Allaire P, Lin Z, Li G (2009). Constrained balancing of two industrial rotor, systems: Least squares and min-max, approaches. Shock Vib. 16:1-12. DOI 10.3233/SAV-2009-0451.

Sinha JK, Friswell MI, Lees AW, Sinha RK (2002). Robust Method for the Rotor Unbalance Estimation. Proceedings of VETOMAC-2, 16-18 December, 2002, pp. 1-11.

Siqueira AAG, Nicoletti R, Norrick N, Cavalca KL, Castro HF, Bauer J, Dohnal F (2012). Linear parameter varying control design for rotating systems supported by journal bearings. J. Sound Vib. 331:2220-2232.

Swanson E, Powell CD, Weissman SA (2005). Practical Review of Rotating Machinery Critical Speeds and Modes. Sound Vib. 12(11):91-100. May

Zhou S, Shi J (2001). Active Balancing and Vibration Control of Rotating Machinery: A Survey. Shock Vib. Digest 33(4):361-371.