23rd ABCM International Congress of Mechanical Engineering
December 6-11, 2015, Rio de Janeiro, RJ, Brazil

DESIGN AND CONSTRUCTION OF A TEST BENCH FOR STUDY OF VIBRATION ANALYSIS TECHNIQUES APPLIED TO PREDICTIVE MAINTENANCE

Israel Antônio Macedo de Lima
Maria Alzira de Araújo Nunes
University of Brasilia, UnB Gama College, Automotive Engineering, Área Especial de Indústria Projeção A, Setor Leste, 72.444-240, Gama-DF, Brasil
israel.macedo1@gmail.com, maanunes@unb.br

Abstract. *In the current industry, modern technologies of Predictive Maintenance (PdM) are predominately being used to effectively monitor performance of equipment and plan maintenance interventions in a timely manner. As the majority of plant equipment is constituted for mechanical systems, vibration monitoring is generally the key component of most PdM programs. The constant development of the vibration analysis techniques has represented a spread in PdM mainly in rotating equipment. This subject motivated the authors to design and construct a PdM test bench for study and research purposes. This paper presents the initial conception, the theoretical concepts, analytical and numerical techniques used in the design stage and finally the entire practical steps for it construction. The proposed test bench is capable to simulate misalignment, unbalance of rotary components and bearing deterioration. Aiming a low cost of manufacture, as compared to commercial test benches, this is constituted of: electric motor, shaft, coupling, bearings and a perforated disc, most of them recycled components. The diagnoses of failures were obtained from practical vibration measurements performed in laboratory. The results show the viability to use this test bench in later advanced studies in PdM, as well didactic application.*

Keywords: *test bench, vibration, predictive maintenance, design, didactic.*

1. INTRODUCTION

Maintenance has during a long period of time mostly been associated with costs and stoppages and has, of this reason, acquired a connotation of being something necessary evil. Maintenance activities are playing an increasingly important role on ensuring structural reliability, safety, availability and maintainability as well as optimizing operating cost of modern engineering systems (Yuan, *et al.*, 2013). In the last decades, an increasingly number of companies replaces the current reactive maintenance strategy with proactive strategies such as predictive maintenance (Fredriksson and Larsson, 2012).

Predictive Maintenance (PdM) is an emerging technology that can be employed to detect potential failures that may not be evident through a preventative maintenance program. If the failure characteristics of the equipment are known, PdM can detect the failure well in advance and appropriate actions can be taken in a planned manner. This is done by monitoring the condition of the equipment continuously or periodically depending upon the need for the availability of the machine. The use of condition based maintenance has reduced non-value added maintenance by eliminating the need to unnecessarily shutdown equipment for maintenance checks. Some of the main technologies currently used in industry are: thermography, vibration monitoring, oil analysis, visual inspections and ultrasonic measurements (Liggan and Lyons, 2011).

Among the techniques for PdM one of the most used is the vibration analysis. Vibration monitoring is generally the key component of most condition based maintenance programs and is also the most widely used technique with 80% of the parameter measured are likely to be vibrations based. However, this technique is limited to monitoring the mechanical condition and should be used in conjunct with other monitoring and diagnostic techniques in order to provide all of the information that will be required for a successful condition based maintenance program (Mahmood, 2011).

The principle of the vibration analysis is based on the idea that the structures of the machines excited by the dynamic efforts give vibrational signs, whose frequency is equal to the frequency of the exciting agents. An imbalance in a machine component will cause increased vibration, once it causes an imbalance in the system and consequent increase in the power. Thus, observing the progression of the level of vibration, it is possible to obtain information on the state of the machine (Gonçalves and Silva, 2011).

However, the use of adequate sensors to gather information may be extremely valuable for data acquisition and analysis of vibration. It is highly recommended that personnel from both operations and maintenance department be trained, once this particular information will provide a deeper and broader understanding of the machine condition (Mahmood, 2011).

In this context, the use of test benches is a good alternative for professional training, teaching and researches about the theme. The studies of new scientific maintenance methodologies are specially needed to enhance equipment reliability and safety, reduce maintenance manpower, spares, and repair costs, eliminate scheduled inspections, and maximize lead time for maintenance and parts procurement (Yuan, *et al.*, 2013). So, these studies will be better conducted in laboratory with the use of test benches instead direct application in situ.

Some commercial test benches are available for these purposes. Although, the manufactures offer pre-defined models which works in specific operation conditions. The other point is the high cost for acquire this one, once most of them should be imported. For academic purpose, it is recommended a functional system which possibilities operational changes according the research interest. Other important requisite is the low cost, since the funds are limited meanly in Brazilian academics institutes.

In view of the importance of PdM for modern industries and engineering processes, the need of studies and researches about new maintenance methodologies, and the importance of introducing this theme to engineering students and professionals interested in the area, authors decided to design and manufacture their own test bench for PdM application. It will be part of the acoustic and vibration laboratory (Lab NVH) of UnB - Gama College (FGA). Prior it will be used in teaching, didactic and researches activities. The proposed test bench aims to attend beginners in experimental tests in rotating machinery, with capacity to simulate misalignment, unbalance of rotary components and bearing deterioration. It is basically constituted of: electric motor, shaft, coupling, bearings and a perforated disc, most of them recycled components.

Due to vibrations analysis technique applied to PdM be the most popular technologies used in recent days (Bhattacharya and Dan, 2014) and it be wide and reliably used technique in the industry (Mobley, 2002), this technique will be used is this work in order to demonstrate the applicability of the designed test bench.

The remainder of this paper is organized as follows: in section 2 are shown some commercial and academic test benches models which were references to our test bench proposed; section 3 describes the simplified numerical model using finite element method (FEM) in order to realize modal analysis; section 4 shows how the test bench was manufactured; section 5 show the experimental procedures in order to test the test bench and finally section 6 presents the main conclusions of the work.

2. TEST BENCH CONCEPTION

Before to start the test bench design, the authors made a bibliographic revision concerning to a similar works even realized in the same area. Many papers, articles, thesis and others were encountered but most of them doesn't describe in details the design of the test bench, neither discuss about it manufacture process (Pacholok, 2004; Meola, 2005; Abreu *et al*, 2007). In most of the bibliographic references the test bench is only a tool in order to test some new approach method for PdM. So, this was a motivation for authors to continue this work since it will be useful for many researches which needs manufacture one. Some works were selected after analysis aiming feasibility and low cost of the test bench manufacturing. Figure 1 shows some of the models which provide ideas to set our model.

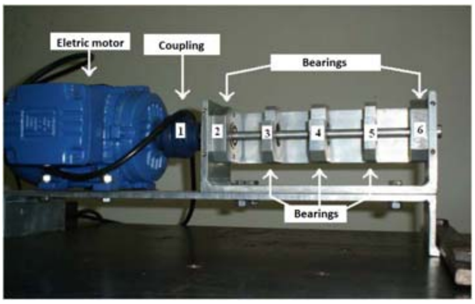

Figure 1a. Test bench manufactured by Meola (2005)

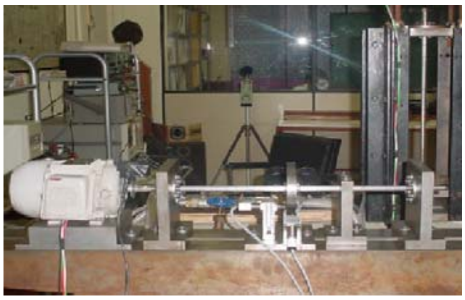

Figure 1b. Test bench used for Garcia (2005)

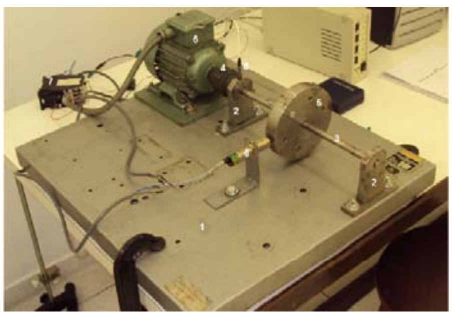

Figure 1c. Test bench used for Silva (2012)   Figure 1d. Commercial test bench at SENAI-Gama/DF

The advantages and disadvantages of each bench showed in Fig. 1 were analyzed resulting in a proposed model which meets our needs. CAD software was used to design it. Figure 2a shows the CAD model in details and the Fig. 2 is the isometric view. It is important to know that this final model suffered many modifications since its initial propose, mainly due to manufacture limitations and reduction of costs.

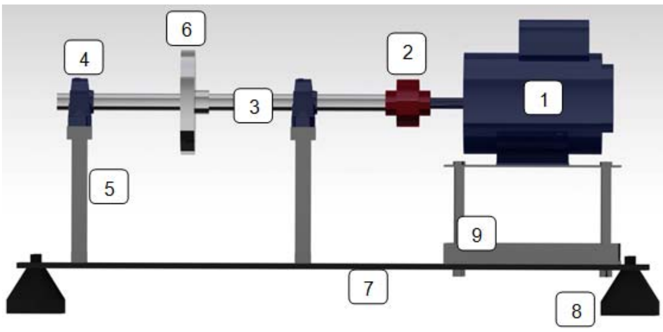

Figure 2a. CAD model of the proposed test bench.

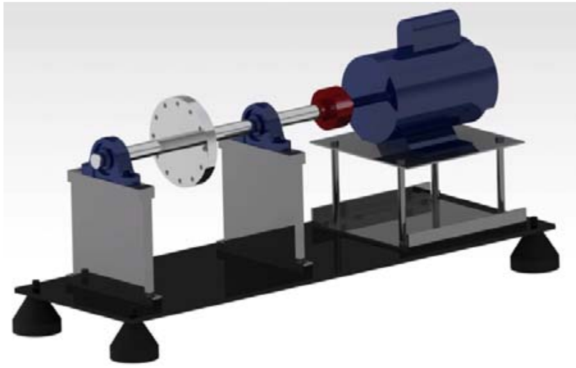

Figure 2b. Isometric view of the proposed test bench.

The components (numbering in Fig. 2a) of the test bench proposed are: 1-eletric motor; 2-coupling; 3-rotating shaft; 4-plummer block and bearing; 5-plummer block support; 6-perforated disc; 7-base plate; 8- rubber support; 9-adjustment device to impose misalignment in the system. The height of components 5 and 9 was established in order to possibility the alignment process of the system (laser shaft alignment). After conception of the test bench layout, numerical simulations were performed in order to verify the shaft resistance and natural frequencies of the system.

3. NUMERICAL MODELLING

This step was performed in this work in order to verify the mechanical resistance of the shaft and to estimate the natural frequencies of the main components of the equipment.

The first analysis realized in the shaft was static. After, centripetal force due to an unbalanced force was considered applied in the shaft (simulating the perforated disc with an eccentric mass). The shaft dimensions were chosen in function of the electric motor requisites and the commercial bearings dimensions. It was acquired commercially, with diameter 0.025 m and material: Steel ASTM A-36 with Young´s modulus: 165 MPa (Beer *et al*, 2006).

From analytical estimative, the maximum bending moment was computed as 1.2 MPa and the maximum shear force is 1.2 MPa. Both are much smaller than the Young’s modulus, so it is capable to support these applicants’ forces (Beer *et al*, 2006). The Von Mises stress was estimated by FEM using CATIA® software (*Generative Structural Analisys*). Figure 3a shows the static simulation result and Fig. 3b shows the simulation result due to the centripetal force (0.04 kg mass located in the perforated disc). The obtained results showed that the shaft is capable to support the applied forces.

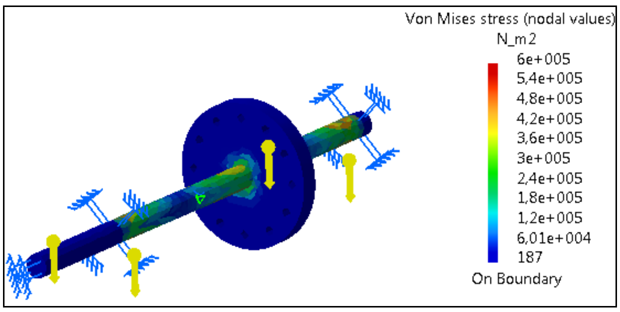

Figure 3a. Von Mises stress due to static forces.

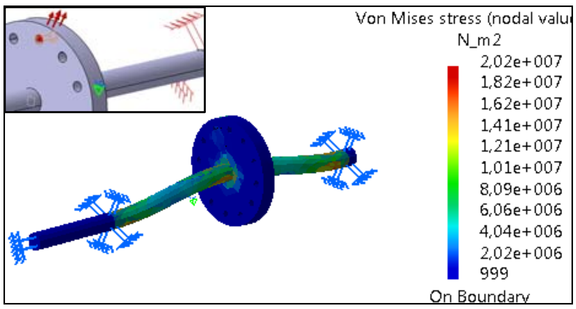

Figure 3b. Von Mises stress due to centripetal force.

The next numerical simulation is modal analysis. It is important to note that this step is only a preliminary verification of the modal behavior of the system, and no experimental validation will be perform later, since this is not the focus of this paper. The objective is just to verify how close the natural frequencies estimated numerically are to the rotation frequency of the electric motor.

A simplified finite element model was constructed using ANSYS® software. In order to obtain a simple model, all the components were created directly in ANSYS® workspace using areas and solid geometries. Each modeled component (numbered in Fig. 2a) has the same mass as in the CAD project, and consequently the same materials properties. Figure 4a shows the finite element mesh of the model. The elements are 3-D 8-nodes structural solids, size length 0.01 m, resulting in 83454 elements and 20842 nodes.

In order to perform modal analysis, boundary conditions were defined in the nodes located at bases plate restricting displacements in *x* and *y* axes. The first three natural frequencies simulated were: 100.8 hz, 136.1 hz and 183.84 hz. This was a satisfactory result since the electric motor rotation is 57.5 hz. So the natural frequencies simulated not coincide with the electric motor rotation and its harmonics. As said before, it is only a simplified investigation in order to avoid possible resonances and consequently damages or faults when the system is working.

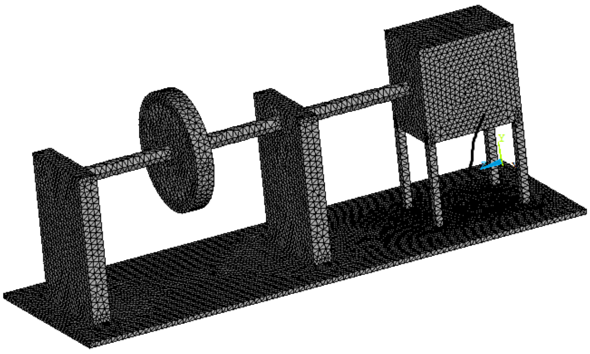

Figure 4a. Finite element mesh.

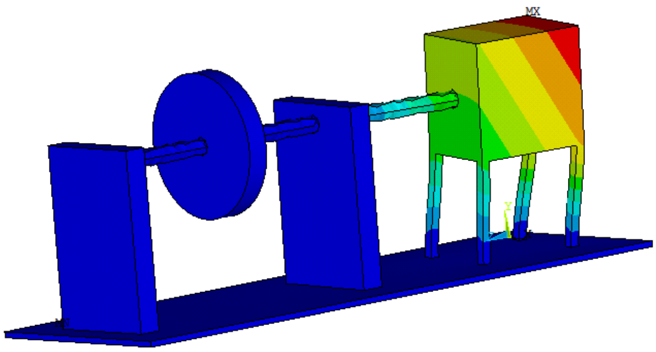

Figure 4b. Modal shape at 100.8 hz.

After analysis of the numerical results, the CAD model proposed was maintained and the next step is the test bench manufacturing. Next section will describe this stage.

4. MANUFACTURING OF THE PROPOSED TEST BENCH

Firstly, the following components were commercially acquired: electric motor (1), shaft (3), flexible coupling (2), rubber support (8), plummer block and bearings (4). The number into parenthesis is the correspondence with Fig. 2a. The perforated disc (6) and plummer block support (5) were machined from parts of pieces not used in the university/laboratory. They were machined in order to get the desirable form, geometry and final shape. All the manufacturing process was done at Processes of Fabrication Laboratory of the UnB-FGA.

The raw shaft had the following specifications: circular cross section of 0.0254 m diameter, 6 m length and ASTM-A36 steel. The shaft was machined, as illustrated in Fig. 5a. The final length is 0.575 m and 0.025 m diameter (in order to obtain h7 interference). The perforated disc was machined from a slab steel ASTM A-36 (Fig. 5b).

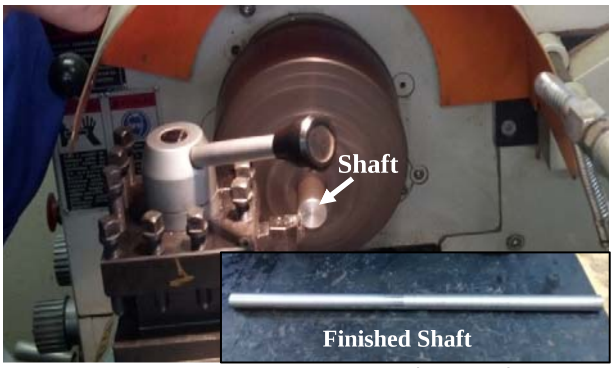

Figure 5a. Machining process of the shaft.

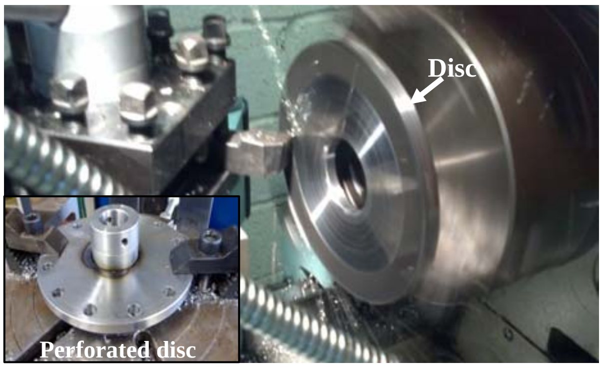

Figure 5b. Machining process of the disc.

The final shape of the disc is: 0.150 m external diameter, 0.020 m internal diameter and 0.020 m thickness. Twelve holes were made in the disc which will pin the unbalanced mass. The holes are separated from 30 degrees each other, and they have 0.008 m diameter.

The plummer block supports were manufactured using firstly a saw blade to approximate the raw board to the desired dimensions and after a milling machine was used to finalize the component as illustrate in Fig. 6a.

The final plummer block support was obtained connecting other part manufactured in order to support the final component. Finally the holes necessaries to pin this component to the plate base and the plummer block in it were done. Figure 7a to 7c shows the complete finalization of this component. It has 0.112 m height, 0.200 m length and 0.036 m thickness.

The bearing used is a Y type with only one ball filament. The specification is: SKF® Ball Bearing Insert YAR 205-2F. The Y-bearing plummer block unit has cast housing and grub screw locking. Its specification is SY505M SKF® with 0.025 m diameter.

Figure 6a. Cutting the support.

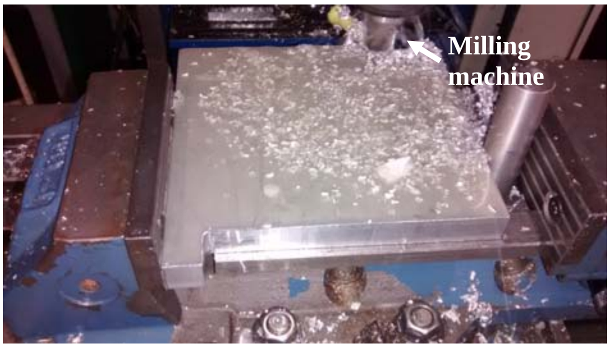

Figure 6b. Bushing support milling.

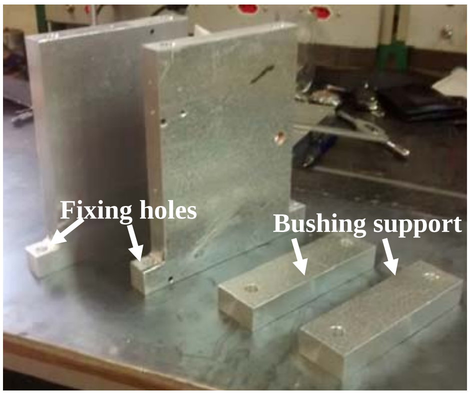

Figure 7a. Separated parts of support bushing.

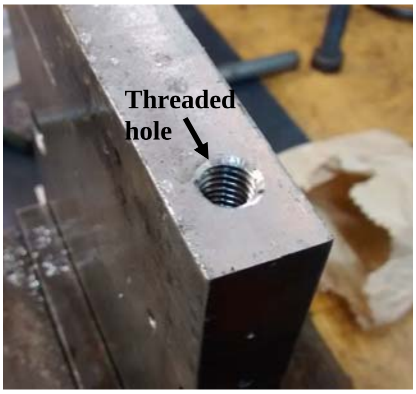

Figure 7b. Threaded hole in detail.

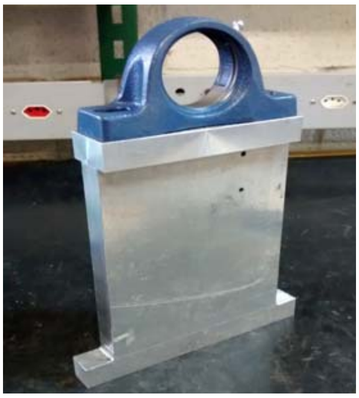

Figure 7c. Support bushing finalized.

The electric motor (not new but in perfect conservation state) has the following specifications: 3520 RPM speed, 0.5 CV rated output, 110/220 V input voltage, monophasic. Its speed rotation was checked experimentally using a tachometer (contact and laser). Ten experimental samples were measured and the mean speed (±standard deviation) estimated for the contact tachometer and laser tachometer model was respectively: 3,591.2 ± 1.03 and 3,592.5 ± 1.43. So the electric motor is working in accordance with fabricant specification.

The electric motor apparatus was designed in order to permit four alignments types. Figure 8a shows the electric motor fixed in this apparatus which have four threaded bars (1), top plate (2), lower plate (3) and four setscrews (4). This system allows misalignment in both vertical and horizontal planes. In the horizontal plane (the same plane as the top plate), the four threaded bars are capable to impose parallel or angular misalignment in the shaft, as illustrated in Fig. 8b. In order to impose misalignment in the vertical plane (orthogonal to top plate), it is necessary to adjust the screws identified in Fig. 8a as number 4. This adjustment allows a movement along the direction showed in Fig. 8c using the marked screw (red circle) in the same figure. It is important to note that was necessary to make oblong holes in the lower plate, in order to allow relative movement between plate and threaded bars.

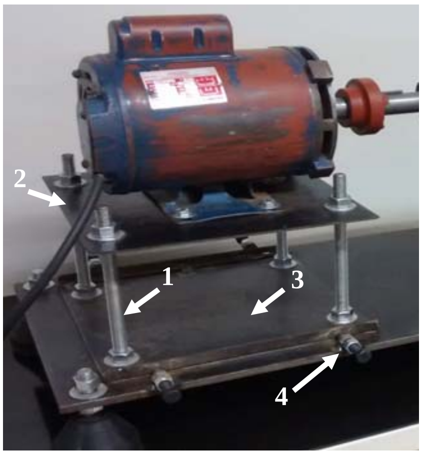

Figure 8a. Electric motor apparatus.

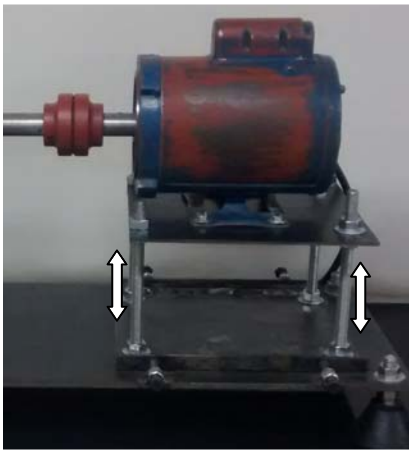

Figure 8b. Horizontal plane misalignment.

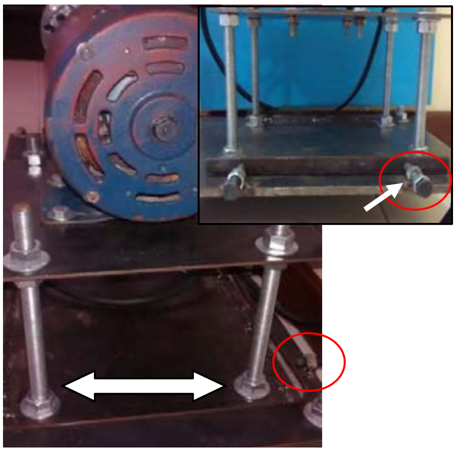

Figure 8c. Vertical plane misalignment.

A flexible bush pin type flange coupling was used in order to connect both manufactured and motor shafts. It was used shaft coupling key for this purpose. Finally, all the components listed before were mounted in the plate base of 0.008 m thickness. Figure 9 shows the finalized test bench.

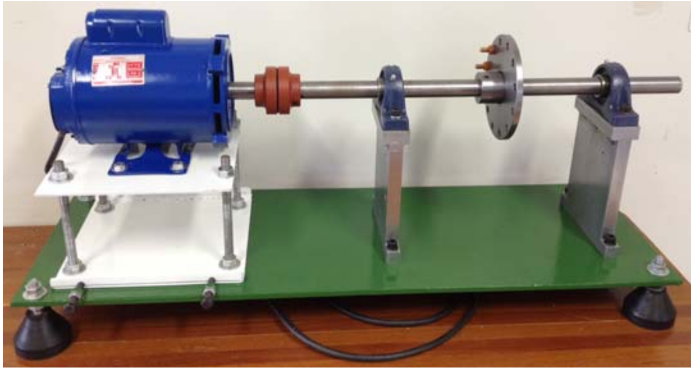

Figure 9. Test bench manufactured.

5. EXPERIMENTAL TESTS

In this section, the experimental results obtained with the test bench manufactured are shown. It was induced three configurations in it: misalignment, unbalance and bearing defect. First of all, the test bench was installed and so the base plate has been aligned using bubble level device.

In order to impose parallel and angular misalignment in the horizontal plane, it was adjustments in the threaded bars height. A laser shaft alignment tool (TMEA 1P/2.5 SKF®) was used in order to measure the misalignment induced. This tool consists of two measuring units and a display unit. The measuring units were fitted with brackets and chains for attachment to the shafts. Both measuring units emit laser, which is projected on the position sensor detector of the other unit. Figure 10a shows this equipment fixed in the test bench. In the left top corner (Fig. 10a) is the display in detail illustrating a parallel misalignment status. Figure 10b and 10c illustrate the measuring units.

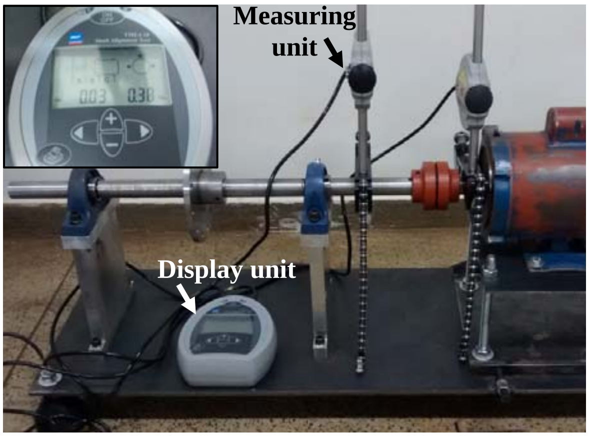

Figure 10a. Laser shaft alignment tool.

Figure 10b. Measuring units.

Figure 10c. Measuring units in details.

The measurements and analysis of vibration signals were done using a digital system analysis of vibrations (SDAV Teknikao®) which has two accelerometer input channels and an USB data acquisition board. The piezoelectric accelerometer was fixed in the plummer block top as shown in Fig. 11a. It was used 2 kHz sampling rate and 30 seconds of time sampling.

Figure 11b shows the graphical results of the misalignment experiments. It is easy to note the higher vibratory energy (global energy) when the system is misalignment, and higher values of *rms* (root mean square) acceleration in the axial direction which is an intrinsic characteristic of misalignment faults (Benevenuti, 2004).

The next test was to identify an unbalance condition in the test bench. First, it was made the alignment of the shaft and the vibration signal was acquired. After, a mass (bolt and screw) of 0.020 kg was inserted in the perforated disc in order to simulate an unbalance condition (Fig. 12a). Both vibration signals (unbalanced and balanced conditions) were analyzed in the frequency domain as shown in Fig. 12b.

The vibration signal was acquired and analyzed for all three directions: axial, horizontal radial and vertical radial. So, here is shown (Fig. 12b) only the radial vertical acceleration because this reached the most vibratory energy on the rotation frequency when compared with the other two signals. It is important to note that the acceleration acquired in all considered directions had high vibratory energy in the rotation frequency (59 Hz) when compared with the balanced system analyzed. So the radial direction it is the most relevant for this kind of fault in a mechanical system (Benevenuti, 2004).

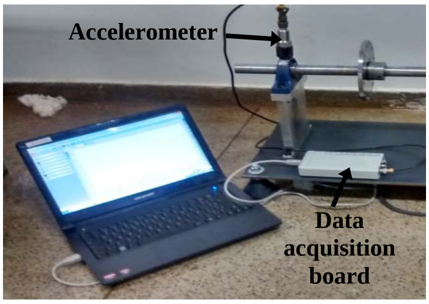

Figure 11a. Experimental setup acquisition.

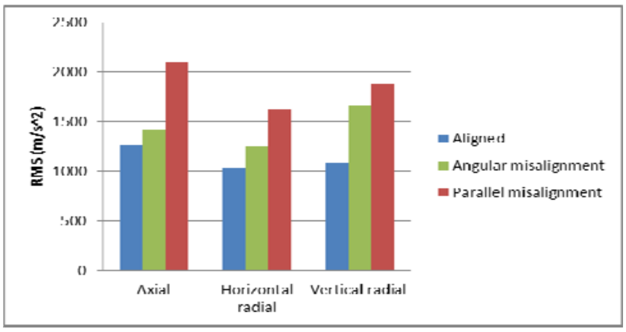

Figure 11b. Results of misalignment experiments.

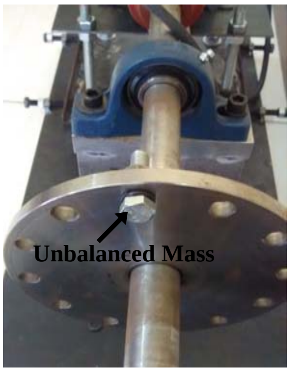

Figure 12a. Unbalanced Mass in the perforated disc.

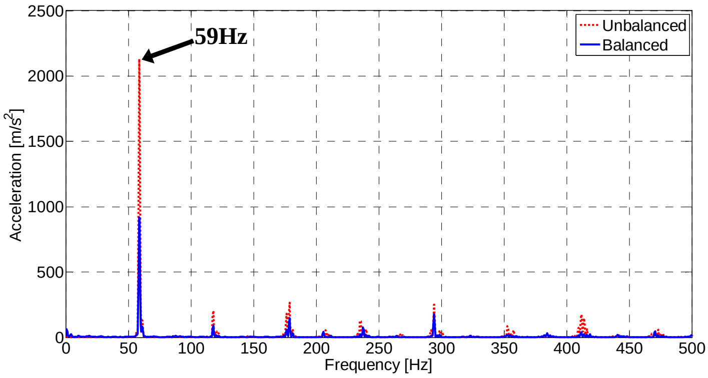

Figure 12b. Vibration signal of balanced and unbalanced conditions.

Lastly, the bearing defect was induced in order to acquire the vibration signal and to detect it using envelope analysis. The external bearing (close to shaft end) was replaced by one with induced outer race defect. This last was made by own authors who dismounted the bearing and caused this defect.

According to manufacturer specifications (SKF® Model YAR 205-2F) is possible to estimate the bearing characteristic frequencies using: bearing geometry and the speed at which bearing rotate (Bezerra, 2004). The mean bearing characteristic frequencies are: ball pass frequency of inner race (BPFI), ball pass frequency of outer race (BPFO) and bass spin frequency (BSF). Using envelope analysis (Fig. 13) is possible to detect the bearing defect. It is easy to note in Fig. 13 the dominant frequency in the envelope graphic which is 214.2 Hz, corresponding to the BPFO and which is the induced defect. Although, all the defects simulated in the test bench proposed was detect by vibration signal.

Table 1. Estimated bearing characteristic frequencies.

BPFI	BPFO	BSF
324.30 Hz	214.16 Hz	23.79 Hz

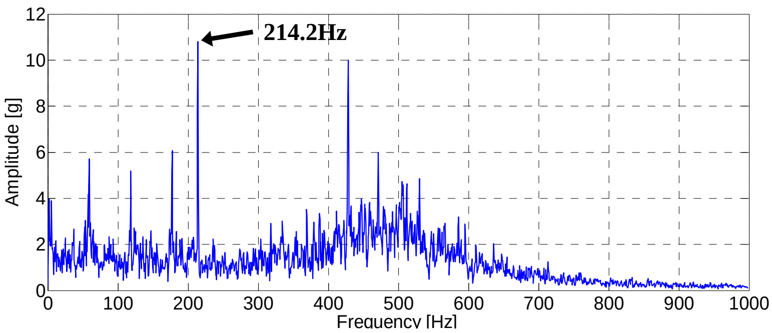

Figure 13. Envelope of bearing with inner race defect.

A cost study was done by the authors (Lima, 2014) in order to quantify the financial economy obtained by manufacturing the test bench instead to buy one from commercial suppliers. Similar commercial equipment was quoted R$ 70,000.00, while the test bench manufactured by the authors had a cost R$1,100.00 by using recycled components. If all components were acquired from commercial suppliers, the final test bench manufactured in the academic institution would be a cost of R$ 4,156.00. It is important to highlight this saving once it was one goal of this project.

6. CONCLUSIONS

This paper describes a test bench proposed to vibration tests and analysis. It aims to attend beginners in experimental tests in rotating machinery, with capacity to simulate misalignment, unbalance of rotary components and bearing deterioration. The main contribution of this paper is to show how a test bench for this purpose may be designed and manufactured in academic institutions. The final product may be used for teaching and researches in vibration and dynamics areas, as well as, for training in industries and specialized laboratories. The results obtained from acquired vibration signal shows that the system meets its goal.

7. ACKNOWLEDGEMENTS

The authors thank SENAI-Gama/DF for technical support; the Processes of Fabrication Laboratory of the UnB-FGA and their technician team for manufacturing support.

8. REFERENCES

Abreu, R. D. A, Brito, J. N. and Lamim Filho , P. C. M., 2007. "Identificação de falhas em rolamentos através da detecção de envelope e da decomposição em wavelets". In *Proceedings of the XIV CREEM*, Uberlândia, Brazil.

Beer, F., Johnston Jr., E. R., Eisenberg, E. and Mazurek, D., 2006. *Vector Mechanics for Engineers: Statics and Dynamics*. McGraw-Hill, 8 ed.

Benevenuti, E.L., 2004. *Metodologia para monitoração e diagnóstico de vibração das bombas moto-operadas do circuito primário de refrigeração do reator IEA-R1*. M.Sc. thesis, Instituto de Pesquisas Energéticas e Nucleares, São Paulo, Brazil.

Bezerra, R. A., 2004. *Detecção de falhas em rolamentos por análise de vibração*. Ph.D. thesis, Universidade Estadual de Campinas, Campinas, Brazil.

Bhattacharya, A. and Dan, P. K., 2014. "Recent trend in condition monitoring for equipment fault diagnosis". Int J Syst Assur Eng Manag, 5 (3), p. 230–244.

Fredriksson, G. and Larsson, H., 2012. *An analysis of maintenance strategies and development of a model for strategy formulation-A case study*. M.Sc. thesis, Chalmers University of Technology, Göteborg, Sweden.

Garcia, M.S., 2005. *Análise de defeitos em sistemas mecânicos rotativos a partir da monitoração de vibrações*. M.Sc. thesis, Universidade Federal do Rio de Janeiro, Rio Janeiro, Brazil.

Gonçalves, A. C. and Silva, J. B. C., 2011. "Predictive Maintenance of a Reducer with Contaminated Oil under an Excentrical Load through Vibration and Oil Analysis". J. of the Braz. Soc. of Mech. Sci. & Eng., Vol. XXXIII, N. 1.

Liggan, P. and Lyons, D., 2011. "Applying predictive maintenance techniques to utility systems". *Pharmaceutical Engineering*, Vol. 36 No 6.

Lima, I. A. M., 2014. Proposição de uma bancada didática para análise de vibração gerada por desalinhamento e desbalanceamento. Graduate thesis, Universidade de Brasília, Brasília, Brazil.

Mahmood, S. T., 2011. *Use of Vibrations Analysis Technique in Condition Based Maintenance*. M.Sc. thesis, Royal Institute of Technology, Stockholm, Sweden.

Meola, T., 2005. *Monitoramento em tempo real da qualidade de sinais de vibrações utilizando inteligencia artificial*. M.Sc. thesis, Universidade Federal de Uberlândia, Uberlândia, Brazil.

Mobley, R. K., 2002. *An Introduction to Predictive Maintenance - Establishing a predictive maintenance program*. Elsevier Inc.

Pacholok, M., 2004. *Uso da termografia para avaliação do desalinhamento de eixos e máquinas rotativas: Uma Ferramenta auxiliar á análise e de vibrações*. M.Sc. thesis, Pontifícia Universidade Católica do Paraná, Paraná, Brazil.

Silva, B.T.V., 2012. *Bancada para análise de vibração: Análise de falha em máquinas rotativas*. M.Sc. thesis, Universidade de Taubaté, Taubaté, Brazil.

Yuan, Y., Jiang, X. and Liu, X., 2013. "Predictive maintenance of shield tunnels". *Tunnelling and Underground Space Technology*, Vol. 38 p. 69–86.

9. RESPONSIBILITY NOTICE

The authors are the only responsible for the printed material included in this paper.