

Dynamic Behavior Analysis of Rotor Supported by Damped Rolling Element Bearing Housing

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ABSTRACT

A typical rotating machinery system consists various components, such as rotor, support bearing, and disks. These components pass out energy into the system when coincident to critical speeds. Ignoring such event might lead to disastrous breakdown of the system. Due to necessity and vital contribution to most rotating machineries, the requirements on rolling element bearings have become stricter every day. In this experimental study, the dynamic behavior and displacement of rotor supported by damped rolling element bearing housing for different running speeds and load levels are analyzed and compared.

Keywords: Rotor, Rolling Element Bearing, Damped Bearing Housing, Critical Speed.

Sönümlü Yuvarlanmalı Yatakla Desteklenen Milin Dinamik Davranış Analizi

ÖZ

Yaygın olarak kullanılan dönen sistemler miller, diskler ve rulmanlar gibi çeşitli makine elemanlarından oluşur. Bu parçaların büyük bir çoğunluğu yuvarlanmalı yataklar üzerinde çalışırlar. Mil-yatak sisteminde yükleme konumunun belirlenmesi sistemin dinamik davranışları açısından büyük önem taşır. Sistemde uygun olmayan yüklemelerin yapılması milin mekanik enerjisini artırır. Bu tür dinamik davranışların oluşumunu dikkate almamak sistemde arızaya neden olabilir. Bu nedenle yuvarlanmalı yataklar makinelerin dinamik davranışları için hayati bir fonksiyona sahiptir. Bu deneysel çalışmada, farklı çalışma hızları ve yük seviyeleri için sönümlü yuvarlanmalı yatakla desteklenen mil, dinamik davranışları ve yer değiştirme yönünden incelenmiş ve karşılaştırılmıştır.

Anahtar Kelimeler: Mil, Yuvarlanmalı Yatak, Sönümlü Yatak, Kritik Hız.

1. INTRODUCTION

Rolling element bearing is a fundamental component of rotating machinery and diagnosis of its dynamic behavior is very important. Large amount of use of rolling element bearings indicates their vital contribution to the performance of rotating machinery. Rotating machinery plays an important role in industry due to wide range of their applications. It is very important to control their dynamic behavior for a safe operation. There are many studies for rolling element bearing in literature. However, very few studies on the damped rolling element bearing housing. Dynamic behavior control has been an area of theoretical and experimental research [1]. Extended literature reviews on the state of the art can be found in reference [2] and reader is also referred to the references [3-10] for further reading. Rotating machinery often has problems of instability at high running speed, which can result in sudden failures of the whole system or parts of

it. This problem can be solved by adding damping in the bearing housing [5].

This study presents an isolator as damping between bearing and its surrounding and an experimental analysis of bearing housing able to work as an effective vibration damper has been employed. The results showed that the damped bearing housing can be used for reduction in rotor resonance amplitude where mass and stiffness cannot be used.

2. EXPERIMENTAL SETUP

The experimental setup used in this study is schematically shown in Figure 1. A shaft having length of 850 mm and diameter of 12.7 mm located in the set. The shaft is supported on two ball bearings and driven by 0.5 HP alternative current induction motor attached to flexible coupling and a variable speed control unit. Six disks having diameter of 126.2 mm and weight of 1507, 1508, 1506, 1500.5, 1506.5, and 1506.5 grams (from the left to the right respectively) are used in order to load the shafts for enhancing the spectrum amplitude of the

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system. The disks are mounted on the shaft at different locations and unbalance is introduced to each disk with unbalance screw separately as sets to excite the shaft for critical speeds. The vibration spectrum of the inboard and outboard bearing housings in the vertical and horizontal directions are measured with accelerometers having frequency range up to 10 kHz. Two accelerometers are mounted on each of bearing housings with 90° apart. The system incorporates Data Acquisition card supplies channels for vibratory response and rotational speed acquisition. Channels were set as Ch1 and Ch2 for inboard (close to motor) bearing housing while Ch3 and Ch4 for outboard bearing housing to measure vibration response in the vertical and horizontal directions respectively. Two Eddy Current Proximity probes (whose resolution is 1 μm - micrometer) for measuring the shaft relative displacements are fixed on rigid supports at 2 mm linear distance between probe tip and shaft surface. The vertically and horizontally mounted probes make it possible to reliably observe the vibration behavior. The data were gathered using the Spectra Quest™ software and hardware system.

The experimental test procedures were as follows: two test cases were utilized to understand dynamic behavioral changes of the shaft mounted on typical (plain) bearing housing and damped bearing housing. Seven experimental setups were employed under five shaft running speeds (10, 20, 30, 40, and 50 Hz) for each of the test cases. The set A was utilized for normal state without

unbalance screw in order to establish the base-line data. Set B, Set C, Set E, Set F, and Set G were used for different location with unbalance screw attached to the disk while the Set D was used with two unbalance screw attached to disk. The sets (Set A, Set B, Set C, Set D, Set E, Set F, and Set G) configurations are given in Figure 2.

3. RESULTS AND DISCUSSION

The vibration responses of plain and damped bearings were measured at different unbalance screw position and speeds. Figure 3 gives maximum peaks of vibration spectrum from inboard bearing housing in the vertical direction (Ch1) and the horizontal direction (Ch2), and also from outboard bearing housing in the vertical direction (Ch3) and the horizontal direction (Ch4) for all test cases with respect to running speeds of 10, 20, 30, 40, and 50 Hz. It can be observed from Figure 3 that values of vibration amplitude vary with sets configurations. This indicates that unbalanced weight distribution affect the amplitude. It should be noted that the spectrum from from bearing housing in vertical direction represents resonance compare to the horizontal direction for the most of the test cases. Thus, the unbalance force gravity acts perpendicular downward on the shaft. This is very clear for the shaft running speed of 50 Hz. The maximum amplitude value of 0.0500 gRMS, an average of overall vibration, was recorded from Ch4 for Set D of damped bearing housing at shaft speed of 50 Hz while 0.0450 gRMS from Ch4 for Set D of plain

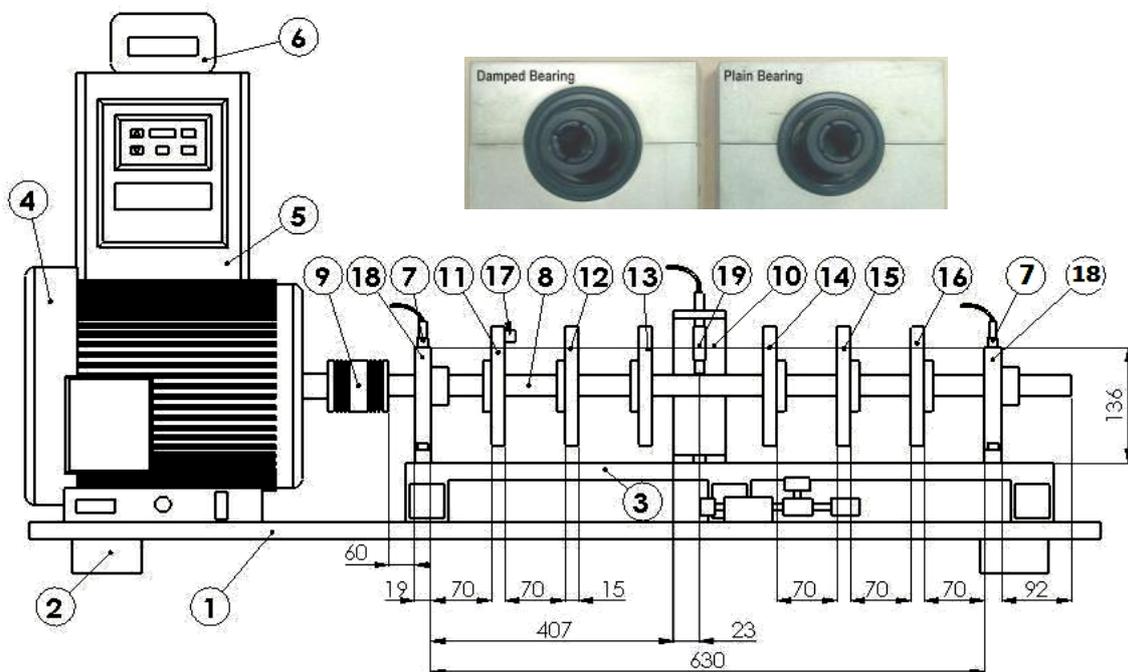


Fig. 1. Experimental setup

- (1) Basis; (2) Rubber isolators; (3) Extended rotor headpiece; (4) Motor; (5) Variable speed controller; (6) Tachometer; (7) Accelerometer; (8) Shaft; (9) Flexible coupling; (10) Rigid support for proxy probe; (11) Disk (1507 gr); (12) Disk (1508 gr); (13) Disk (1506 gr); (14) Disk (1500.5 gr); (15) Disk (1506.5 gr); (16) Disk (1506.5 gr); (17) screw (4.5 gr); (18) Bearing housing; (19) Proxy probe.

bearing housing. It can be said that the faster the shaft running speed the larger the magnitude of vibration response. It should be noted that the most of amplitude values for the plain bearing housing was higher than the damped bearing housing at shaft running speed of 20 Hz comparing to those of 10, 30, and 40 Hz at start-up. Also it should be note that it is not the case for shaft speed of 50 Hz. Figure 4 illustrates the vibration data captured from inboard and outboard bearing housing in the vertical and horizontal directions showing multiple channels in one graph for shaft running speed of 20 Hz. The vibration data collected from all channels are arranged as from the

bottom plot through the upper plot as ch1, ch2, ch3, and ch4 respectively. Although each spectrum is composed of 5000 individual peak, the frequency range of 0-1 kHz is presented. The amplitude peaks show the energy distributions at different frequencies for plain and damped bearing housing. The magnitude of amplitude spectrum for plain bearing is found to be quite unstable in comparison to damped bearing. It can obviously be seen that the vibration level for plain bearing housing is higher than that for damped bearing housing. It can be concluded that adding damping between bearing and its surroundings enables to reduce amplitude of the vibration

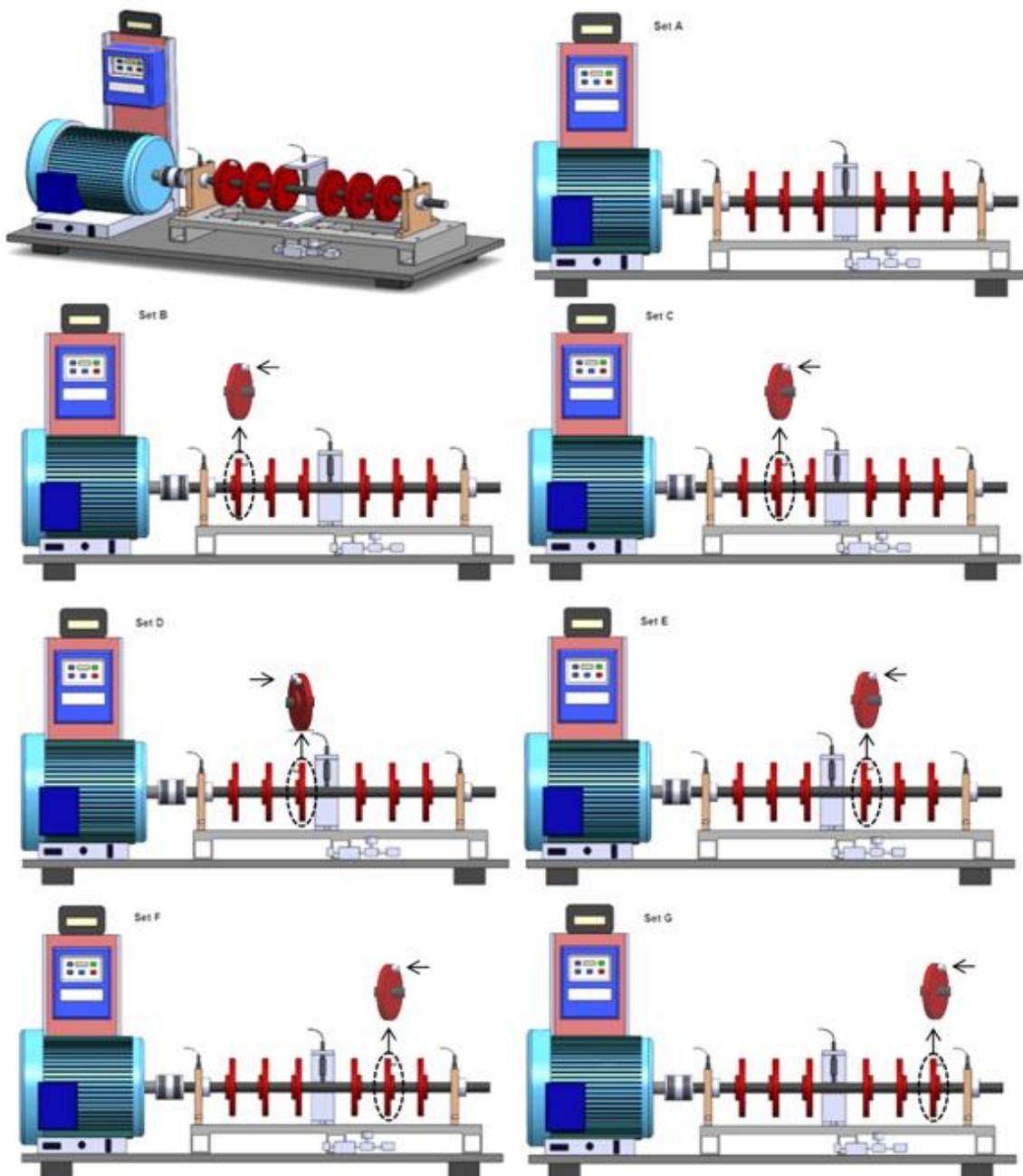


Fig. 2. Configurations of experimental setups

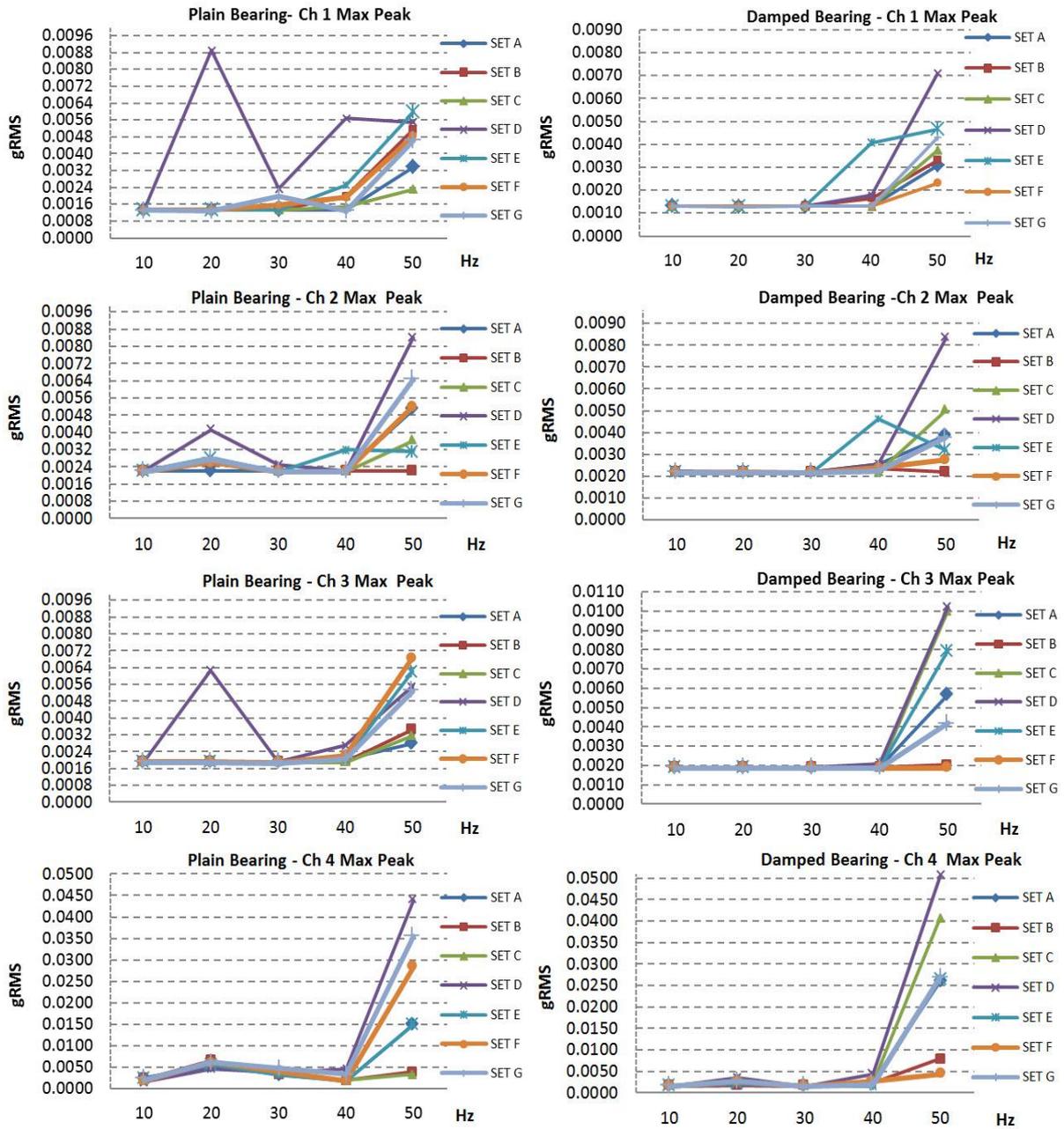


Fig.3. Maximum peaks of vibration spectrum versus shaft running speeds for all sets

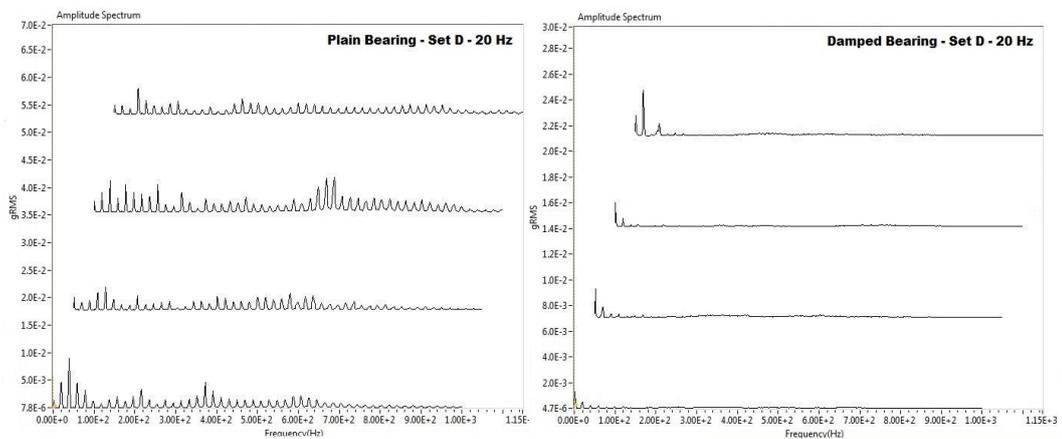


Fig. 4. The amplitude spectrum of bearing housing types for set D at shaft running speed of 20 Hz

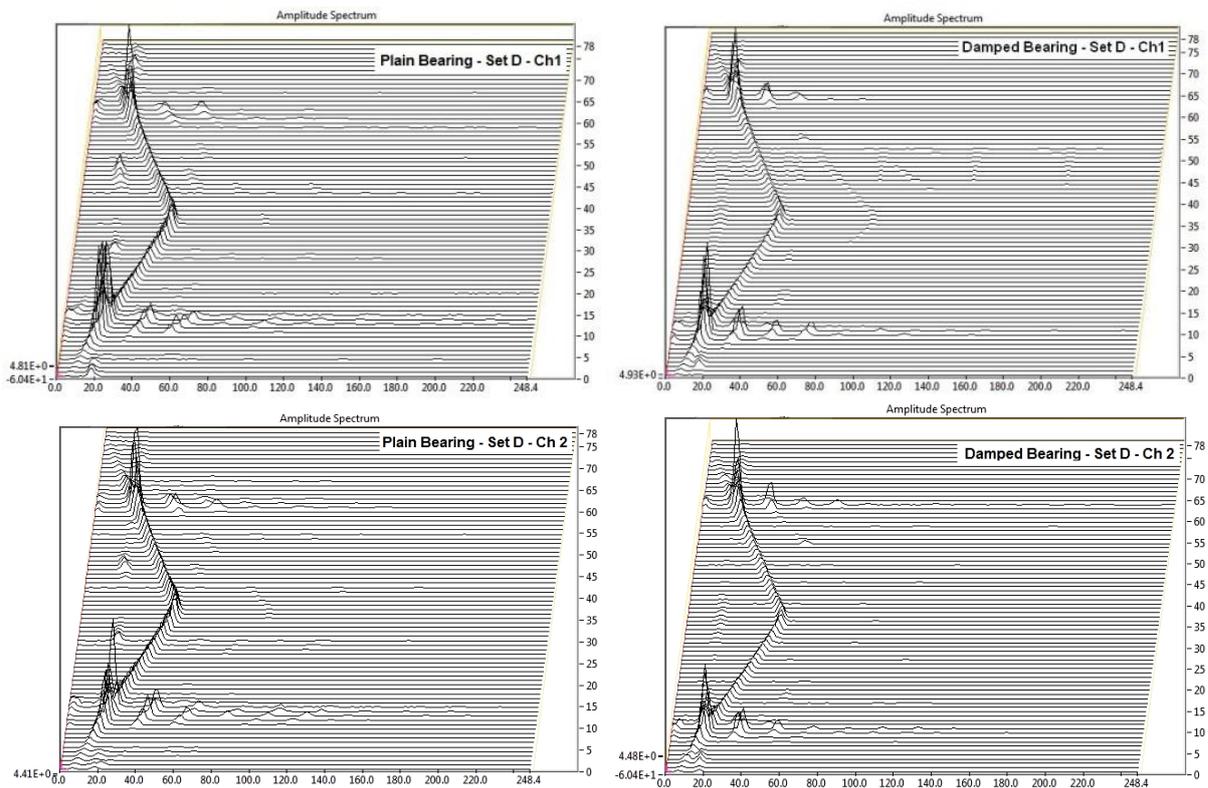


Fig. 5. Waterfall plot of run up and coast down in test cases for set D

Figure 5 shows the frequency waterfall in the vertical (Ch1) and horizontal (Ch2) directions for both plain and damped bearing housing. Set D is selected due to higher vibration. High vibration level was observed in both vertical and the horizontal direction of plain bearing case compare to the damped bearing. It can be seen that the surrounding isolator can damp some frequencies.

It can be noticed that a large frequency component of 1X causing 2X and 3X are barely recognized. Figure 6 shows the Bode plots of Set D during start-up processes. The speed controller allowed the system to operate in the range from 0 to 3000 rpm. The data were obtained by proximity probes for shaft running speed varying up to 3000 rpm. It can be seen that the maximum amplitude value in the vertical and horizontal directions for plain bearing is around 1400 rpm while for damped bearing is around 1200 rpm. It can be seen that vibration amplitudes of plain bearing housing are more propagating than the damped bearing housing.

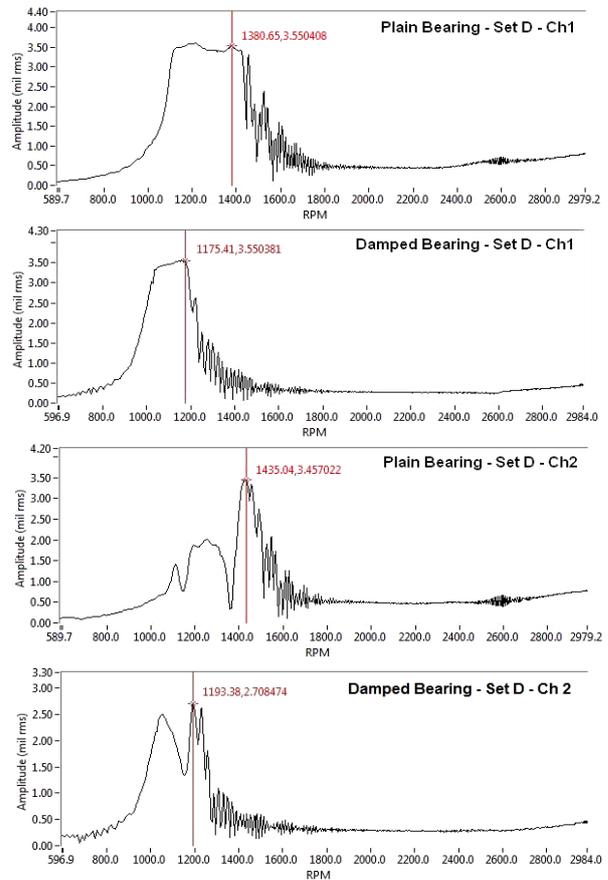


Fig. 6. Vibration amplitude values versus shaft running speeds

4. CONCLUSIONS

This study experimentally showed how the location of disk with respect to attaching unbalance screw and load position can significantly affect the dynamic behavior of shaft supported by plain and damped bearing housings. The experiment is mainly focused on the rotating machinery dynamic characteristics under unbalance conditions. The results showed that the location of the unbalance screw attached to the disk with respect to the load position can significantly affect the vibration response. The damped bearing housing can be used for reduction in rotor resonance amplitude where mass and stiffness cannot be used. The reduction of vibration transmission to the system can be obtained by applying an isolator between bearing and housing. Since damping is the most cost effective and appropriate means to attenuate resonant vibration, the damped bearing housing is less costly. The results show that the use of an isolator in the bearing housing can be very efficacious in rotating machinery vibration control.

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