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Experimental Analysis of the Dynamical Response of a Flexible Rotor Including the Effects of External Damping

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Abstract: This study presents several experimental tests for the analysis of the dynamics of a flexible rotor. At first several modal analysis tests are made for different rotating speeds, with a shaker providing an asynchronous excitation. The Frequency Response Functions obtained from these tests are used to build the Campbell diagram of the rotor. In another series of tests, the synchronous response of the rotor is measured and the results are analyzed in parallel with the Campbell diagram. Secondly, the objective is to evaluate experimentally the effects of adding no-rotating damping for a rotor system to decrease the vibration amplitudes of the rotor within the range of operating speeds. The influence of the contribution of damping on such a system is evaluated by checking the coast up tests, mainly in the critical speeds zones. To accomplish such study, several alternatives of damping were tested to decrease the vibratory levels of the rotor at the synchronous critical speed.

Key words: Rotordynamics, campbell diagram, damping, critical speeds

INTRODUCTION

Rotating machinery is very common in the industry. For example, it is one of the main actors in the paper and textile industries, in the petrochemical industry and in the hydraulic networks. In this study, the vibratory phenomena associated to this kind of machine have been intensively studied[1-11] to assure the best operational conditions of the rotating machinery. Generally speaking, two of the most important aspects considered when designing rotating machinery is the good placement of the critical speeds[7,9,10] and the synchronous steady state response of the system[8,11].

One of the most important problems found in the domain of the rotating machinery is how to decrease the vibration levels at the operating conditions. The standard technique is the balancing. However, there are specific cases where it does not provide satisfactory results. One solution to this problem is to insert non rotating damping in the system. For the present study, an experimental device is proposed to install a damping system at the bearing supports and to evaluate the effects of damping on the vibration amplitudes within the range of operating speeds and more particularly near the first forward critical speed.

This study concerns the three aspects above cited: critical speeds, unbalance response and external damping. At first, a test rig that was developed at the Laboratoire de Tribologie et Dynamique de Structures of the Ecole Centrale de Lyon (France) will be presented. This test rig was built to provide experimental data for updating numerical models and help understanding the phenomena associated to non-linearities near the critical speeds. Then, the orbits and unbalance responses are measured, as well as the Campbell diagram, to help understanding the unbalance response peaks. Also orbits and operating shapes are measured. Finally, a special system consisting of an arrangement of rubber dampers, installed on the test rig, is presented and its effect and performance are evaluated.

THE TEST RIG

Description of the test rig: This test rig was designed and built[6] to accommodate three forward critical speeds in the rotating speed range, with all the parameters and physical dimensions coming from detailed finite element models. The test rig (Fig. 1) is composed of a horizontal shaft (1.7 m long, 0.04 m diameter), one 0.40 m disk, two bearings and two bearings support. The rotor is

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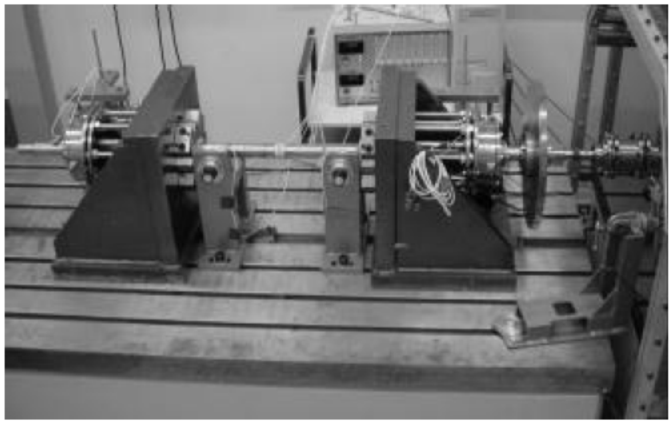

Fig. 1: The test rig

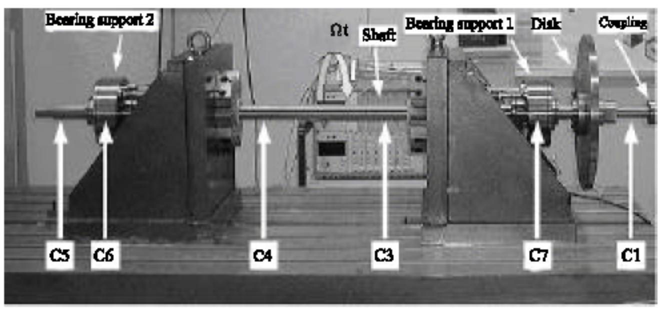

Fig. 2: Picture of the test rig, with the indications of the measuring planes and the rotating direction of the rotor

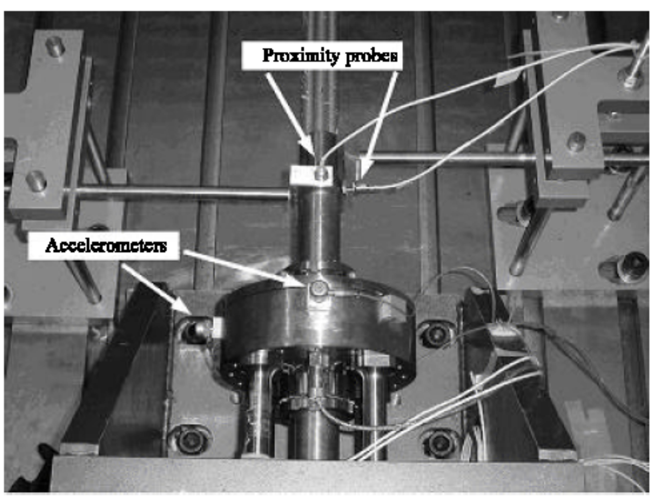

Fig. 3: Measuring planes C5 (proximity probes) and C6 (accelerometers)

connected to an electrical motor by a flexible coupling. The system is placed over a massive concrete table isolated from the environment by rubber slabs.

Instrumentation: Measurement of the vibration for rotating machines is relatively more difficult to realize than that of traditional structures, because of the rotation of a part of the system. Moreover, for a machine containing a flexible rotor, measures taken at the fixed parts of the system (like the bearing supports) is not sufficient to describe correctly the vibratory behavior of the system.

To have a good description of the dynamic of the rig, there exist six measurement planes each containing two transducers in an orthogonal arrangement. Four of these planes are placed to measure the transversal displacement of the shaft, by using non contact displacement transducers. The remaining two measurement planes are placed in the bearing supports and the planes consist of accelerometers. The test rig has also a phase meter and temperature transducers to monitor the bearing temperatures.

All the transducers are connected to an HP 3565A mainframe. The measuring planes are illustrated in Fig. 2 (planes C1, C3, C4, C5, C6 and C7), with the rotating direction showed by an arrow. Fig. 3 shows in detail the measuring planes C5 and C6.

RESULTS AND DISCUSSION

In this study, the dynamic behavior of the rotor is first undertaken by considering the classical unbalance responses. The orbits and deformation of the rotor are investigated for various rotational speeds. Then, an experimental procedure based on an asynchronous forcing is described to provide the Campbell diagram. An analysis of the unbalance response curves in parallel with the Campbell diagram is performed.

Unbalance response: In the field of rotating machinery, one of the classical tests carried out consists of studying the system's vibratory response for a constant acceleration speed profile. The results of these tests are function of the balancing quality and the value of the acceleration. The data acquisition system is configured to perform an order tracking up to the fifth order.

Figure 4 shows the orders 1 and 2 of a run up test measured at the bearing support 1. The abscissas are normalized with the first forward critical speed as the reference. In this figure, it is possible to see the first backward critical speed (at 0.86) and the first forward (at 1) in the order 1 curve. On order 2 curve, one can see in particular a peak at the speed 0.48 corresponding to the maximum 2X vibrations (vibrations whose frequencies are the double of the rotating frequency). It seems that the peaks at the adimensional rotating speeds 0.7 and 0.9 correspond to the second forward and backward modes.

Also, by considering the evolution of the phase and amplitude for the five orders, it is possible to synthesize the orbits at each measuring plane Ci (I = 1,3,..7) defined in Fig. 3. From the orbits the operational shape of the rotor is constructed using a degree three polynomial interpolation to connect the measuring planes. Fig. 5-7 show the orbits and the operational shape for rotating speeds corresponding to the observed 2X maximum, to the first backward critical speed and to the first forward critical speed. The orbits shown in these figures have a near elliptical shape, indicating that there exists some degree of asymmetry in the system. In the case of the orbits like the one obtained for the plane C1, the orbit shape approaches a circular form as the centrifugal forces became dominant.

The campbell diagram: The Campbell diagram is one representation of the evolution of the eigenvalues of the system with the rotating speed. One method available to establish this diagram for a rotating machine is to provide an asynchronous excitation to make the backward whirl mode appears clearly. In this study, the asynchronous excitation comes from an electromagnetic shaker installed on the bearing support 1, as shown in Fig. 8. The experimental procedure consists on evaluating the frequency response function of the system at several rotating speeds, using a sine sweep approach. The results obtained with this procedure are shown in the waterfall plot in Fig. 9. From this figure, it is easy to see the evolution of the first forward and backward whirl modes.

Based on the waterfall graphic, one can build the traditional Campbell diagram of the test rig, as illustrated in Fig. 10. This figure shows clearly the evolution of the first forward and backward modes, with a third degree polynomial function to fit the experimental data.

The interpolating curve shows that at rest the resonant frequencies are not coincident, indicating that the system has asymmetric properties. Actually, the first forward mode starts at 0.92 and the first backward mode starts at 0.90 Physically, this difference means a separation of about 0.5 Hz of the modes and this may be an important factor to explain the presence of the backward whirl at the unbalance response.

Moreover, the Campbell diagram can be superposed with the unbalance response to validate the localization of the critical speeds (Fig. 11). For the unbalance response, the figure contains the first and second orders. Clearly the extrema observed on the order 1 curve correspond on the Campbell diagram to the intersection of the synchronous excitation line (1X line) with the curves of evolution of the backward and forward modes. The extremum observed in the order 2 curve correspond with the intersection of the 2X line with the curve of evolution of the forward mode. Still on the Campbell diagram, it is possible to see also that at the frequency of intersection of the 2X line with the backward evolution line there is a small peak at the order 2 response curve. At the order 2 response curve, there are also two peaks about 0.7-0.9 that might correspond with the intersection of the 2X line with the evolution curves of the second backward and forward whirl, as indicated previously:

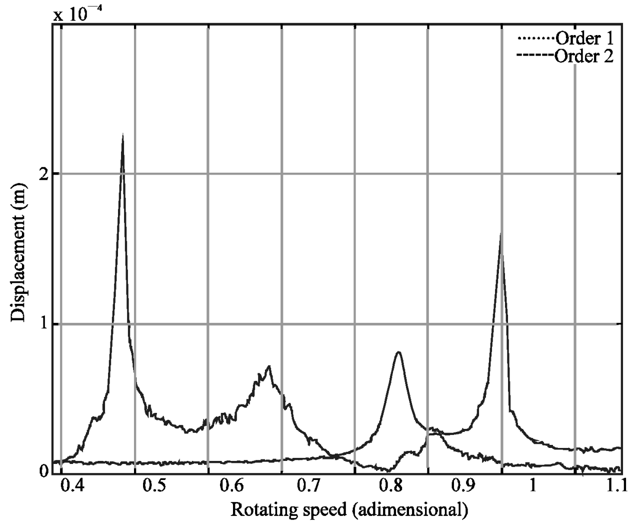

Fig. 4: Unbalance response measured at the bearing support 1 (vertical direction)

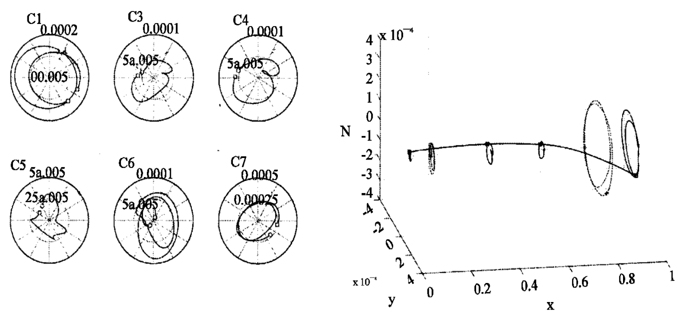

Fig. 5: Orbits (for all measuring planes) and operating shape for the 2X peak

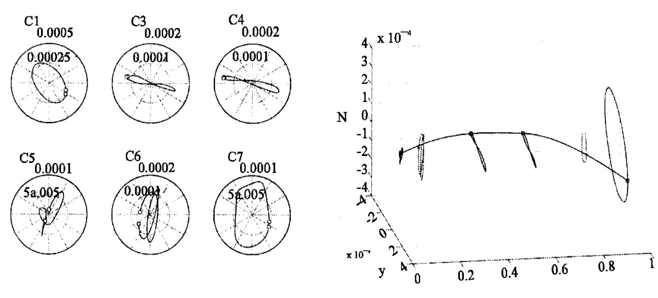

Fig. 6: Orbits (for all measuring planes) and operating shape for the backward critical speed

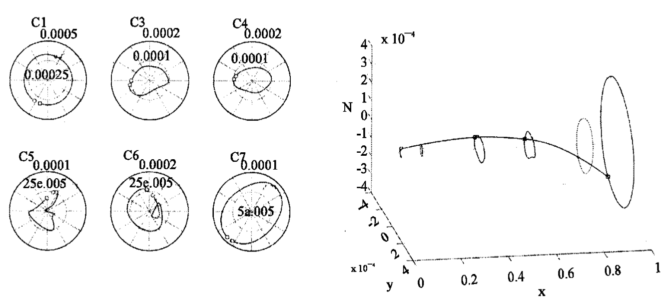

Fig. 7: Orbits (for all measuring planes) and operating shape for the forward critical speed

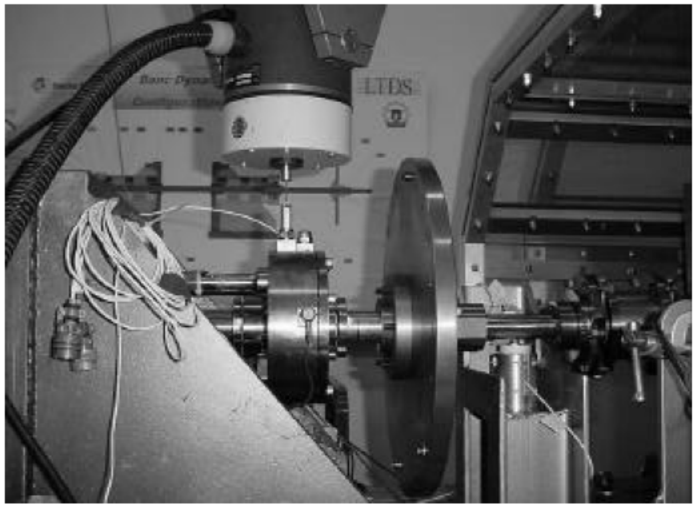

Fig. 8: Experimental setup for measuring the Campbell diagram

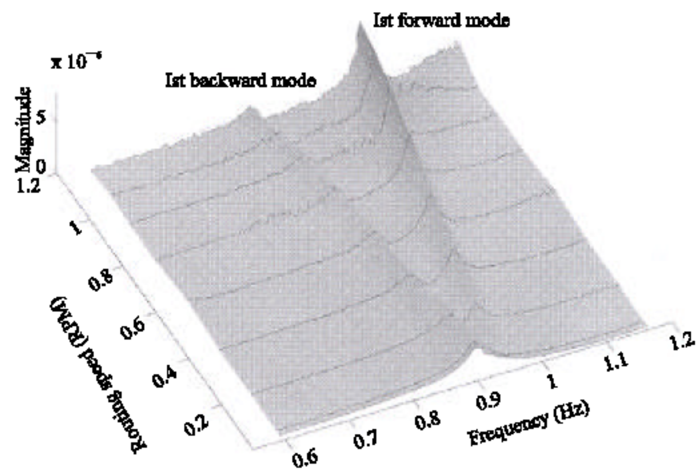

Fig. 9: Campbell diagram in waterfall form

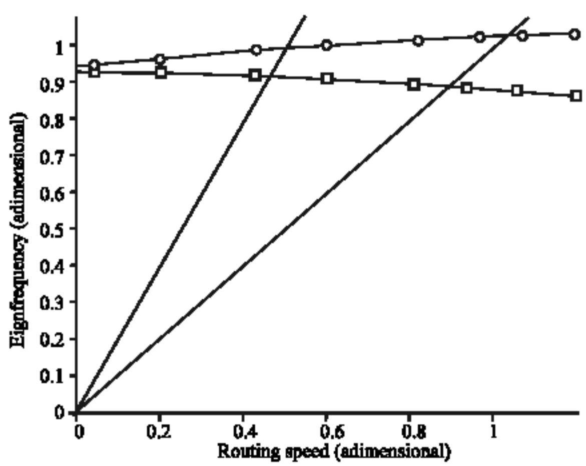

Fig. 10: The Campbell diagram of the test rig

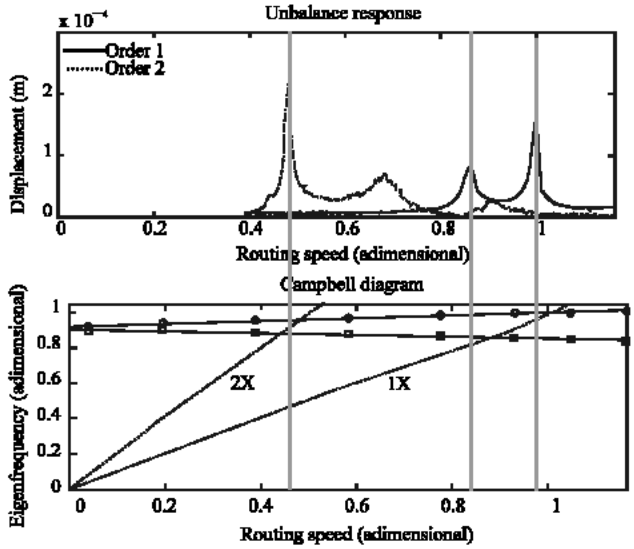

Fig. 11: Comparison between the Campbell diagram and the unbalance response

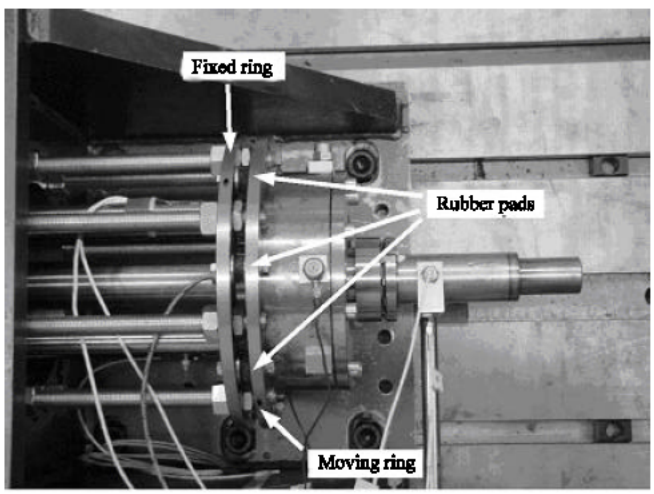

Fig. 12: The damping device

The damping system: The device (Fig. 12) consists of two rings that can have a relative radial displacement. One ring is fixed and the other is solidary to the bearing support. Between these rings, there is a set of anti-vibration mountings that work in shear. This arrangement of the anti-vibration mountings corresponds schematically to add a spring with complex stiffness in parallel to the stiffness of the bearing support.

The efficiency of the damping system is evaluated by checking the level of displacement of the bearing support 1, at the first forward critical speed. This value is then related to that obtained without dampers. Two configurations of the damping system are considered, one with four rubber pads and the other with eight rubber pads, both of which were installed symmetrically around the longitudinal axis of the bearing support.

The results are shown in Fig. 13. The figure shows that the damping system works very well, with the magnitude of the displacement 1.5 times smaller than the reference value for the device with four rubber pads and ten times smaller than that of the eight rubber pads device. These results show the efficiency of the damping system for decreasing the vibration level and the system can be used also to avoid a possible instability threshold after the first forward critical speed.

As a side effect, there is a small change for the values of the first forward critical speed. The four rubber pads device increases this critical speed by a factor of 1.02

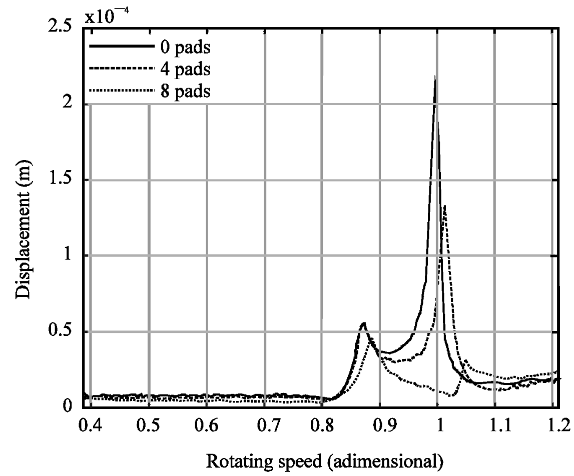

Fig. 13: The performance of the damping device

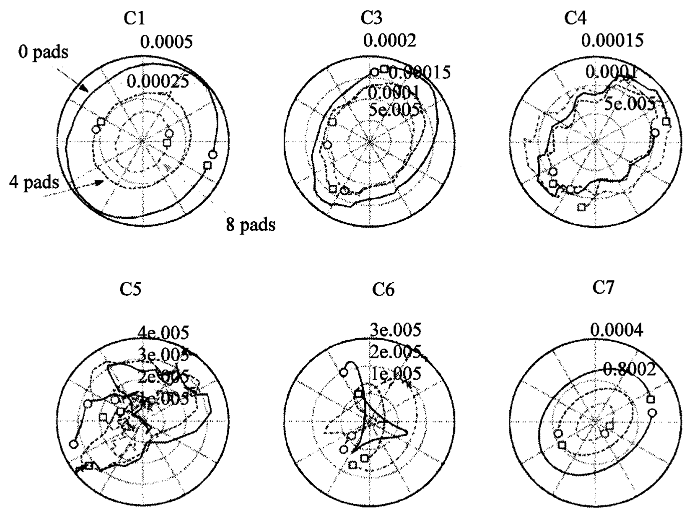

Fig. 14: The performance of the damping device for all measuring planes

while for the eight rubber pads device the increasing factor is equal to 1.05.

The global performance of the damping system can be viewed with the orbit plots shown in Fig. 14. The orbits are time measured at the first forward critical speed and they show that the damping system has a good performance for all measuring planes.

CONCLUSION

This research presents a test rig dedicated to the study of rotating machinery. By a complete experimental approach, the test rig had its dynamics described in details. The first backward and forward critical speeds were determined experimentally by two methods: the unbalance response and the Campbell diagram. It was found that there exists an important component 2X in the responses in the sub critical regime, probably coming from the combination of some level of asymmetry and the weight of the rotor (Dimarogonas[12]). The experimental Campbell diagram confirmed that the system had an asymmetric nature. Moreover, a device designed to damp the system and decrease the vibration levels was tested. This device consists of anti-vibration mountings installed at the bearing supports and it has shown to be very efficient in lowering the displacement levels without changing significantly the dynamic properties of the system.

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