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An experimental study of high-speed rotor supported by air bearings: test RIG and first experimental results

Guido Belforte, Terenziano Raparelli*, Vladimir Viktorov, Andrea Trivella, Federico Colombo

Department of Mechanics, Politecnic of Turin, 10128 Turin, Italy

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Abstract

The paper contains the description of the test rig and first experimental data regarding the mechanical and the thermal behaviour of a 7 kg rotor, 460 mm long and with a diameter of 50 mm, supported by externally-pressurized air-lubricated bearings. The rig allows to load the rotor radially and axially both in static and dynamic conditions and to measure the orbits on the supports. The rotating imbalance response at different speeds has been measured up to 60,000 rpm. The orbits are always synchronous with the rotational speed and their shape changes also with temperature (though this effect is almost negligible). The rotor is stable and does not present whirl instability.

1. Introduction

In high speed machining, production requirements call for a continual increase in removal rates, both to permit the use of smaller drill bits, and to reduce the processing times involved in milling and grinding. Increased cutting speeds are particularly desirable in operations such as drilling printed circuit boards, machining aluminum components for the aeronautics industry, mold and die finishing, and woodworking. Though highly dissimilar in terms of power requirements, rotational speeds and shear forces, these applications share a need for high tangential velocities.

The rolling bearings in common use deteriorate rapidly under high rotational speeds. Ceramic bearings are employed at the highest speeds, but even they do not provide particularly long service lives. One way of overcoming these problems is to use air bearings. In PCB drilling, for example, air bearing spindles are already in industrial use (see www.westwind-airbearings.com). As these units are fully pneumatic and feature no rolling elements, they obviously do not require bearing replacement, making service life practically infinite. Air bearings are not widely used in other industrial applications. This is due to a number of factors, including the lack of a readily usable design algorithm, instability problems, and the difficulty of predicting dynamic behaviour [1–3]. In addition, appropriate test benches and spindles are not available: while determining spindle stiffness characteristics is readily accomplished under static conditions, it is far more difficult under dynamic conditions, particularly at high speeds.

A large number of experimental studies of the behavior of air bearing rotors are presented in the literature. These investigations address a wide range of bearing types, including, among others, foil bearings [4–7], floating journal bearings installed on *O*-rings [8] and fixed journal bearings [9,10]. Though the literature is extensive, it is quite rare to find benches capable of loading the rotor with radial and axial forces in order to measure static and dynamic stiffness [11]. The Politecnico di Torino Department of Mechanics is currently investigating the behavior of high speed air bearing rotors. As part of this investigation, a test bench for analyzing spindles with this type of bearing has been designed and constructed. The bench is capable of monitoring rotor position in the bushing and subjecting the rotor to radial and axial forces during rotation, as well as under stationary conditions. The orbits described at different rotational speeds can thus be plotted, while it is also possible to evaluate bearing axial and radial stiffnesses, both of which are essential characteristics for determining spindle operating specifications.

* Corresponding author. Tel.: +39 11 5646929; fax: +39 11 564 6999.
E-mail addresses: terenziano.raparelli@polito.it (T. Raparelli), vladimir.viktorov@polito.it (V. Viktorov).

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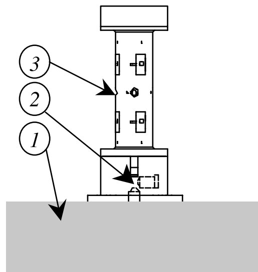

Fig. 1. Test bench outline.

This paper describes the test bench and the first work carried out on it, viz., bench and data acquisition system setup, dimensional checks, and initial measurements.

2. Test bench description

The test bench (see Fig. 1) is made of a heavy base (1) secured to the ground, axial and radial load devices (2), and the spindle under test (3). The load devices make it possible to load the rotor and thus determine axial and radial stiffness while the spindle is rotating.

2.1. Bench components

The spindle is shown in greater detail in Fig. 2, which illustrates how the housing (4) is constrained to the base through flange (5) and journal (6). Radial support is provided by bushings (7) and (8), while axial thrust is opposed by disks (9)–(11).

The housing is made of 18 Ni Cr Mo 5 steel, while the rotor (Fig. 3) (mass 7 kg, diameter 50 mm, length 459 mm) is made of 88 Mn V 8 Ku tool steel quenched and tempered to a hardness of 60 HRC. The rotor was also aged in liquid nitrogen for 5 h and dynamically balanced to a grade better than ISO quality grade G-2.5. The nose (12) to which loads are applied is secured to one end of the rotor, while the driving turbine (13) is integral with the other end.

Bushings are made of the same material as the housing and have an axial length of 100 mm. They are provided with six circumferential sets of four 0.25 ± 0.01 mm diameter radial holes, drilled in brass inserts as shown in Fig. 4.

Axial thrust (Fig. 5) is controlled by two disks (9) and (11) facing the flange on the journal. These disks are separated by a ring (10) whose thickness determines the size of the air gap. Supply air is delivered from an axial hole in the housing, is distributed through a circumferential slot, and then crosses a series of axial and radial channels machined in the disks to reach 0.25 ± 0.01 mm diameter axial nozzles (14) and (15), which are also machined in inserts.

Both the bushings and the disks were surface hardened and machined to produce a surface roughness of 0.2 and 0.4 μm, respectively at the air gaps.

Radial and axial forces are applied to nose (12) by means of load devices (2). These devices are made of a hollow cylinder containing a calibrated sphere with a diametral clearance of 40 μm. When the cylinder chamber is supplied, the sphere is pushed against the nose and at the same time supported, so that it can rotate against the nose without sliding. Radial and axial forces can thus be transmitted to the rotor even when the latter is in motion.

Supply air for the turbine (Fig. 6) crosses pre-distributor (16) in the axial direction to reach annular chamber (17), from which distributor (18) leads to eight tangential channels. Air is exhausted after actuating the turbine. Open loop speed control is accomplished by establishing turbine supply pressure. Spindle components are shown in Fig. 7.

2.2. Compressed air supply

As can be seen from the diagram in Fig. 8, bearing supply (1) is separate from turbine supply (2). Should the air supply

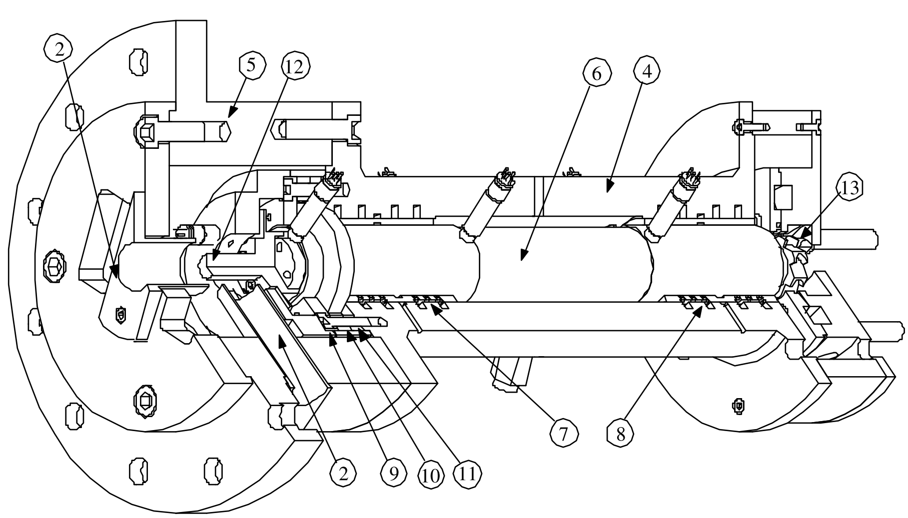

Fig. 2. Section of the spindle and the load devices.