Precision Engineering 40 (2015) 7–13

Contents lists available at ScienceDirect

Precision Engineering

journal homepage: www.elsevier.com/locate/precision

Review

Robust rotor dynamics for high-speed air bearing spindles

R. Dupont*

Levicron GmbH, Sauerwiesen 6, 67661 Kaiserslautern, Germany

ARTICLE INFO

Article history:
Received 11 February 2014
Received in revised form
29 November 2014
Accepted 24 December 2014
Available online 17 January 2015

Keywords:
Air bearing
Spindle
Rotor-dynamics
Resonance speed
Rigid mode criticals
Dynamic stiffness
Bearing gap
Vibration analysis

ABSTRACT

For ultra-precision machining machine tool components need to operate outside critical frequencies of the machining system to avoid insufficient surface finish caused by vibrations. This particularly applies to tooling spindles as those are generally the component of a machine tool with low stiffness and damping values. Surface finish and shape of a machined part rely directly on the overall accuracy in motion of the tooling spindle over the entire machining parameter and speed range. Thus spindle designs for an operation outside critical frequencies combined with high stiffness and damping values are crucial for ultra-precision machining.

For sufficient stiffness properties bearing gaps of gas bearings have to have a size of only a few microns and show a distinct sensitivity on temperature and for journal bearings also on speed. This again means that bearing properties change with temperature and speed. Considering a spindle system comprising a rigid shaft rotating in a radial/axial bearing system with changing stiffness and damping properties leads to a resonance speed map with changing rigid mode resonance speeds.

This paper treats the influence of shaft speed and temperature on bearing gaps from which rigid mode resonance speeds for a shaft spinning in a bearing system are derived. The quoted influence of centrifugal load and temperature on bearing stiffness, damping and load capacity can be applied to any kind of gas bearing. Therefore the calculation of bearing stiffness, damping or load capacity is not treated in detail. The reader will be shown that there are simple design rules for air bearing systems and shafts of high-speed tooling spindles to avoid critical speeds through the entire speed range. Finally, methods of how to prove the initial design goals and how to verify dynamics of high-speed spindles in production will be presented to the reader. It will also be shown that there are production high-speed spindles available which do not include any critical speed within their speed range and thus show robust rotor dynamics with extremely low errors in motion.

Procedures in design, validation and application treated in this paper shall give the reader not only design guidelines for spindles to avoid critical spindle speeds within its speed range, but also recommendations for machine tool builders and end-users for a machine operation taking machine and rotor dynamics into account. As the knowledge for this paper is predominantly based on the experience and work of the author himself only a few references are used. However presented testing results entirely confirm the approach presented in this paper.

Contents

* Tel.: +49 6301 66800 0; fax: +49 6301 66800 910.
E-mail address: r.dupont@levicron.com

http://dx.doi.org/10.1016/j.precisioneng.2014.12.008
0141-6359/

Symbols

f	turning frequency (spindle speed [Hz])
r	cylindrical vibration mode amplitude
C	radial bearing damping
I_θ	transverse rotor moment of inertia
I_0	polar rotor moment of inertia
J	distance between shaft center of gravity and journal bearing center
K	radial bearing stiffness
M	spinning mass
U_s	static imbalance
U_d	dynamic imbalance
ε_0	relative static shaft eccentricity related to static nominal bearing gap
$\varepsilon(\omega)$	relative static shaft eccentricity related to actual nominal bearing gap
φ_r	phase angle of cylindrical vibration mode
φ_θ	phase angle of conical vibration mode
ω	shaft angular frequency
ω_r	cylindrical resonance angular frequency
ω_θ	conical resonance angular frequency
θ	conical vibration mode amplitude

1. Introduction

CNC Machining includes rotating and/or oscillating machine components like motor spindles or linear motors. Abstracting machine components to mass–spring–damper models leads to a system with specific natural frequencies. This includes pumps, fans, axes controllers, the machining process, but predominantly work-holding and tooling-spindles as these are in direct contact with the work piece and create its surface and shape.

For spindle systems imbalances and cutting edges cause vibrational excitations with the turning frequency or multiples of it. Reduction of the bearing system to spring and damper elements and the shaft to its mass and inertia properties lead to a model with characteristic rigid resonance modes.

Because it is characteristic for high-speed spindles bearing characteristics like stiffness and damping change with speed due to aerodynamic effects, thermal influences and centrifugal loads. Thus natural frequencies of high-speed spindles drastically change with speed.

For ultra-precision and micro-machining a spindle operation outside its resonance frequencies is crucial to reduce tool wear and to avoid tool breakage and insufficient surface finish. Up until now it was usual to measure critical spindle speeds and leave those out during machining without a sufficient prediction of spindle characteristics at speed. A more anticipating method – which is described in this paper – is to calculate and predict critical speeds and use the findings to optimize the spindle system to operate below those critical speeds. To achieve this it is necessary to calculate shape and size of the bearing gaps with speed and to take aerodynamic effects into account to get stiffness and damping functions with speed and shaft eccentricity.

It will be shown that it is possible to design high-speed air bearing spindles running below their critical speeds. It will finally be explained how to measure and verify critical speeds of spindle systems and how to derive actual static and dynamic bearing stiffness with speed from it.

Finally the reader will be shown that there are high-speed air bearing spindles available which operate below critical speeds and thus show safe and robust dynamics with outstanding surface finish as a result.

2. Rigid shaft-bearing model of high-speed spindles and stiffness with speed

Gas bearing spindles need to be operated at least 20% below their critical shaft bending speed to avoid tremendous vibrations and the bearing system from being damaged. This leads to a specific shaft-bearing-system where the shaft is considered to be rigid and can thus be reduced to its mass and inertia properties [1]. The bearing system – only taking the radial bearings into account for now – can be reduced to its stiffness and damping properties where both are not constant and depend on shaft eccentricity, rotation speed and thermal conditions. With speed also aerodynamic effects as well as centrifugal and thermal expansions of bearing and/or shaft parts come into play. This approach is valid for any bearing type and thus the calculation of bearing stiffness and damping with speed depends on the bearing system being used and obliges the reader.

The assumptions made so far lead to the following shaft-bearing model.

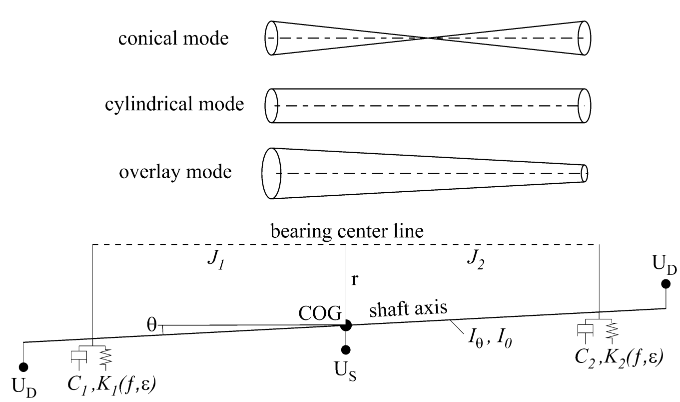

Fig. 1. Shaft-bearing model with speed-dependent bearing properties and a rigid shaft [1].

Journal bearings in high-speed gas bearing spindles work with bearing gaps of 0.0002–0.001 of their bearing diameter. Considering a spindle shaft made from steel and increasing its temperature by 10 °C means a reduction in bearing gap size of 55–11% based on the bearing gap thickness range stated before. This example and the fact that in air bearings the radial stiffness is proportional to the reciprocal value of the square of the bearing gap [1,5] makes clear that for high-speed spindles the bearing gap size and thus stiffness and damping are a function of speed and thus temperature and centrifugal load [2,5]. To predict spindle dynamics it is therefore necessary to know the bearing gap size with speed and requires a mathematical model that contains of thermal and centrifugal loads. Thermal loads are predominantly caused by shear losses within the bearing gaps and motor losses.