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Linear and non-linear stability analysis of an aerostatic spindle for micromachining applications

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In this work the stability analysis of an aerostatic spindle is carried out. The linearized perturbation method is employed to calculate the stiffness and damping coefficients of the journal bearings. These coefficients are then used to perform the linearized stability analysis, which is compared with a non-linear analysis. In order to validate the model, the numerical rotor unbalance response is compared with the experimental measurements.

Keywords: perturbation method, aerostatic bearings, high speed spindle, orbit method

1. Introduction

In different applications domain, such as micromachining and micro-turbomachinery, the miniaturization processes ask for higher and higher rotational speeds, which can be reached only supporting the spindles with aerostatic journal bearings. The design of such devices can be organized in three steps. The first step involves the static design, in which parameters such as load capacity, stiffness and air flow are considered. The second step is preliminary to the stability analysis and is aimed at calculating the linearized stiffness and damping coefficients of the bearings as a function of the perturbation frequency and of the rotational speed. The third step involves the stability analysis, which can be carried out in linear form, using these coefficients, or alternatively, in non-linear form with the so-called orbit method.
This paper shows the design of an aerostatic bearing for a high speed spindle employed in micromachining operations.

2. The design steps

The rotor can be considered rigid in the operating range of speeds and an equivalent lumped masses system is considered to perform the unbalance response analysis. Figures 1 sketches the spindle with distributed mass and the equivalent model with masses lumped in correspondence of the journal bearings.

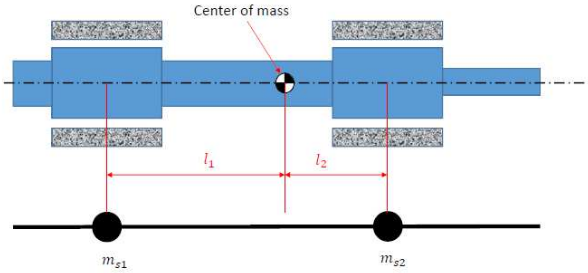

Fig. 1 Scheme of the equivalent lumped masses spindle.

The steady state problem is firstly considered solving the Reynolds equation (1).

$$\oint_{\Gamma_{i,j}} \left(\frac{ph^3}{12\mu RT} \nabla p - \frac{ph}{RT} \boldsymbol{U}' \right) \cdot \boldsymbol{n} \, dl + G_{in} = 0 \qquad (1)$$

Supposing a sinusoidal perturbation along x direction with amplitude Δx and frequency ν, the perturbed pressure distribution $\Delta p = \Delta p_k + j\nu\Delta p_c$ is then obtained after linearization of the problem around the steady state solution. Then, coefficients of stiffness and damping are obtained from relations (2)

$$k_{xx} = -\frac{\iint \Delta p_k \cos\vartheta \, r d\vartheta dz}{\Delta x}$$
$$c_{xx} = -\frac{\iint \Delta p_c \cos\vartheta \, r d\vartheta dz}{\Delta x}$$
$$k_{yx} = -\frac{\iint \Delta p_k \sin\vartheta \, r d\vartheta dz}{\Delta x} \qquad (2)$$
$$c_{yx} = -\frac{\iint \Delta p_c \sin\vartheta \, r d\vartheta dz}{\Delta x}$$

Considering a simple single mass model, illustrated in Figure 2, the rotor unbalance response is studied numerically and the linearized stability analysis is performed [1].

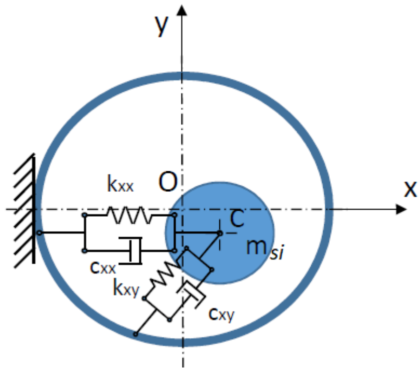

Fig. 2 Sketch of the one mass model.

The numerical results are compared with experimental tests in order to validate the model. The single mass model is found to be sufficiently accurate to describe both the cylindrical and the conical whirl motions.

References

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