Journal of Sound and Vibration 425 (2018) 70–81

Contents lists available at ScienceDirect

Journal of Sound and Vibration

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

# Numerical results of the influence of thermal effects on the turbo machine rotordynamics induced by light-rubs against a brush seal

R. Fay [a,*], D. Kreuzer [a], R. Liebich [a], T. Wiedemann [b], S. Werner [b]

[a] Berlin Institute of Technology, Straße des 17. Juni 135, Berlin 10623, Germany
[b] MAN Diesel & Turbo SE, Oberhausen 46145, Germany

ARTICLE INFO

*Article history:*
Received 21 March 2017
Revised 8 March 2018
Accepted 20 March 2018
Available online XXX

*Keywords:*
Rotor-dynamics
Turbomachinery
Rotor-stator-contact
Newkirk-effect
Brush seals
Stability analysis

ABSTRACT

Brush seals are an efficient alternative for labyrinth seals in turbomachinery. Brush seals show on the one hand a better leakage reduction in relation to their axial length and hence allow a shorter design of the machinery. On the other hand, the particularly small gap between bristles and the engine shaft increases the risk of rotor-stator-contact. The flexible brush seals induces basically light-rubs that in some cases might lead to spiral vibrations and thermal mechanical instabilities.

Spiral vibrations are caused by a thermal deflection of the rotor induced by a heat flow into the shaft. To predict areas of instabilites during the design process a tool was developed at the Berlin Institute of Technology. The model combines a rotor dynamic model and a thermal model. The thermal system is reduced using a stationary solution, so that the final system, on which the stability analysis is performed, is comparable to the established Kellenberger model.

The paper presents the numerical model for the predictions of unstable regions depending on rotational speed. This is illustrated by means of an example of an axial compressor manufactured by MAN Diesel & Turbo.



## 1. Introduction

Replacing labyrinth seals in turbo-machines with brush seals allows a significant reduction in axial length of the seal while maintaining the leakage characteristics. Hence, brush seals allow a wider range of possibilities in rotor design, for example to shift the eigenfrequencies of the rotor away from the operating speed or to reduce the leakage in order to increase the efficiency of the machine. As brush seals provide a comparably low stiffness due to the flexible bristles, the clearance to the rotor can be reduced to almost zero. But a small gap raises the probability of a contact between the rotor and the seal, leading to rub induced vibrations of the machine. Since the contact stiffness of the brush seal is small compared to the rotor stiffness, hard rubbing can be neglected and the friction can lead to the known Newkirk-effect [12]: The power loss due to friction at the contact-line between seal and rotor heats up the rotor asymmetrically. The non-uniform temperature distribution bends the rotor in direction of rotor's rubbing point. The rubbing point changes its circumferential position over time while the vibration amplitude changes as well. That combination leads to the typical spiral orbits observable in rotating coordinates that are sometimes increasing in an unstable manner.

* Corresponding author.
*E-mail address:* robert.fay@tu-berlin.de (R. Fay).

https://doi.org/10.1016/j.jsv.2018.03.024

A simple but very useful model of the Newkirk-effect was developed by Kellenberger [9,10]. He combined the second order differential equation of motion with a first order differential equation that describes the development of the thermal deflection according to the rub amplitude and the thermal parameters of the system. One result of Kellenbergers analysis was, that the stability of the combined system is primarily dependent on the ratio of the input heat coefficient $p$ to the heat loss coefficient $q$.

The model developed by Kellenberger was enhanced to be applicable to arbitrary continuum-rotors and was enhanced for different rub-characteristics [5,11,14,15]. The main challenge of Kellenberger's model is the determination of thermal parameters describing the heat input and loss. In order to improve the prediction of the thermal parameters, the temperature distribution has to be determined. For this reason a thermal model based on the finite volume method was derived [6]. In the present paper, the combined thermal and structural model is presented and explained also by means of an examplary axial compressor by MAN Diesel & Turbo SE. In order to provide as accurate results as possible all relevant effects are considered, such as speed-dependent and orthotropic bearing coefficients, the gyroscopic effect and shear deformation. The prediction of unstable speed regions is carried out using this proposed model of a thermo-elastic rotor behaviour.

## 2. The structural model

The numerical model consists of two parts that interact with each other. First, there is a FE-Beam-Model of the rotor and the support structure with its degree of freedom vector $\mathbf{r}$ as described in Eq. (1).

$$\mathbf{M}\cdot\ddot{\mathbf{r}}+[\mathbf{D}(\Omega)+\mathbf{G}(\Omega)]\cdot\dot{\mathbf{r}}+\left[\mathbf{S}+\mathbf{S}_{\text{Bearing}}(\Omega)\right]\cdot\mathbf{r}=\mathbf{f}_S(t) \tag{1}$$

The rotor mass matrix $\mathbf{M}$, the gyroscopic matrix $\mathbf{G}$ and the stiffness-matrix $\mathbf{S}$ are derived by using the Timoshenko-beam theory with cubic polynomials for the displacement $w$ and the bending angle $\gamma$ [16]. The matrix $\mathbf{S}_{\text{Bearing}}$ contains stiffnesses of the bearings and the support structure, where the bearing-parameters are dependent on the rotational velocity $\Omega$. The damping matrix $\mathbf{D}$ contains the local damping coefficients from the fluid dynamic bearings.

For the mass and the stiffness-matrices, different diameters are used in order to consider blades for example, which do not stiffen the rotor but increase the rotor mass. To ensure, that the masses and the rotational inertias are equal to the ones of the machine itself, single inertias are used in addition for the mass matrix. The final FE-modelling of the axial compressor can be seen in Fig. 1. The seals are located between the journal bearings. The model consists of 119 nodes, with 8° of freedom (DOF) each.

The bearing stiffness is linearized around each operating point and is thus available as a function of the rotor speed. The parameters for both bearings are shown in Fig. 2. Tilting pad bearings with four pads each are used and therefore the bearing stiffness is isotropic. Outside of the in Fig. 2 shown range of speeds the bearing stiffness are extrapolated based on the principle of the nearest neighbor. The bearing support structures are modelled as spring elements with a stiffness in horizontal direction $y$ of $c_{\text{Sup}} = 875\ \text{kN mm}^{-1}$ and in vertical direction $z$ of $c_{\text{Sup}} = 1\,750\ \text{kN mm}^{-1}$.

The rotor of the axial compressor is made of heat-treated steel. The thermal and structural parameters of that steel are listed in Table 1.

The damped critical rotor speeds (crs) are listed in Table 2. Since only the support stiffness varies with the direction, the mode shapes are quite similar for each direction. The first four mode shapes in horizontal direction are shown in Fig. 3. The mode shapes and eigenfrequencies of the vertical direction are fairly similar due to the very similar total stiffness in both directions. This is the result of the series connection of the bearing and support springs whereby the softness of the isotropic bearing dominates. Since the damping is local and viscous a general modal damping factor cannot be determined. In order to provide a damping factor for each crs, the real parts of the eigenvalues of the equivalent first order system of Eq. (1) are divided by the absolute values of the corresponding eigenvalues. These damping factors are shown in Table 2. The damping ratios of

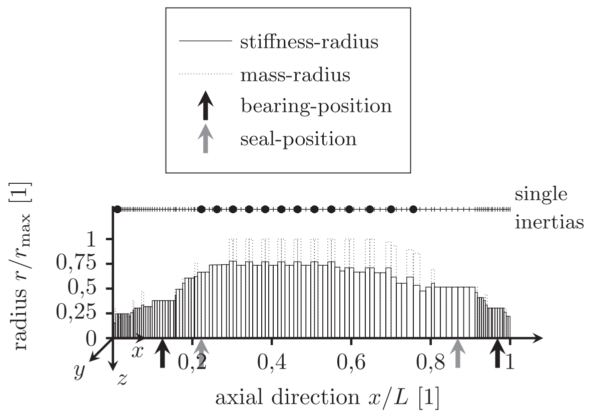


**Fig. 1.** Rotor model of the axial compressor.