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Effect of energy equation in one control-volume bulk-flow model for the prediction of labyrinth seal dynamic coefficients

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ABSTRACT

The influence of sealing components on the rotordynamic stability of turbomachinery has become a key topic because the oil and gas market is increasingly demanding high rotational speeds and high efficiency. This leads the turbomachinery manufacturers to design higher flexibility ratios and to reduce the clearance of the seals. Accurate prediction of the effective damping of seals is critical to avoid instability problems; in recent years, "negative-swirl" swirl brakes have been used to reverse the circumferential direction of the inlet flow, which changes the sign of the cross-coupled stiffness coefficients and generates stabilizing forces. Experimental tests for a teeth-on-stator labyrinth seal were performed by manufacturers with positive and negative pre-swirl values to investigate the pre-swirl effect on the cross-coupled stiffness coefficient. Those results are used as a benchmark in this paper. To analyse the rotor-fluid interaction in the seals, the bulk-flow numeric approach is more time efficient than computational fluid dynamics (CFD). Although the accuracy of the coefficients prediction in bulk-flow models is satisfactory for liquid phase application, the accuracy of the results strongly depends on the operating conditions in the case of the gas phase. In this paper, the authors propose an improvement in the state-of-the-art bulk-flow model by introducing the effect of the energy equation in the zeroth-order solution to better characterize real gas properties due to the enthalpy variation along the seal cavities. The consideration of the energy equation allows for a better estimation of the coefficients in the case of a negative pre-swirl ratio, therefore, it extend the prediction fidelity over a wide range of operating conditions. The numeric results are also compared to the state-of-the-art bulk-flow model, which highlights the improvement in the model.

1. Introduction

The current trend in the oil and gas turbomachinery market is to increase the "flexibility ratio", which is defined by API standards [1] as the ratio between the steady-state speed and the first critical speed. This makes the rotordynamic design more challenging. Because the internal sealing plays a pivotal role in the rotordynamic stability, the accurate numerical simulation of the dynamic phenomena associated with the sealing receives much attention.

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Nomenclature

A_i, A_{0i}	unsteady and steady cross-sectional area of the i-th cavity
a_{ri}, a_{si}	dimensionless length of the rotor and stator of the i-th cavity
B_i	tooth height of the i-th cavity
W	tooth width at the tip of the i-th cavity
H_i, Cr_i	unsteady and steady clearance of the i-th cavity
Dh_i, Dh_{0i}	unsteady and steady hydraulic diameter of the i-th cavity
h_i, h_{0i}	unsteady and steady enthalpy of the i-th cavity
H_1	perturbed clearance
L_i	cavity length of the i-th cavity
$\dot{m}_i, \dot{m}_{0i}$	unsteady and steady mass flow rate in the i-th orifice
NT	number of teeth
M	mass of the rigid rotor
P_i, P_{0i}	unsteady and steady pressure in the i-th cavity
ρ_i, ρ_{0i}	unsteady and steady density in the i-th cavity
R_s	rotor radius
V_i, V_{0i}	unsteady and steady tangential velocity in the i-th cavity
P_{in}	seal inlet pressure
P_{out}	seal outlet pressure
Ω	rotational speed of the rotor
ω	whirling speed of the orbit of the rotor
r_0	radius of the circular orbit of the rotor
τ_{ri}, τ_{r0i}	unsteady and steady rotor shear stress in the i-th cavity
τ_{si}, τ_{s0i}	unsteady and steady stator shear stress in the i-th cavity
f_{si}, f_{ri}	stator and rotor friction factor in the i-th cavity
ε	perturbation parameter
e	absolute roughness of the rotor and stator surface
μ_{1i}	flow coefficients of the i-th orifice
μ_{2i}	kinetic energy carry-over coefficient of the i-th orifice
ς_i	speed of sound under the i-th orifice
C_{eff}	effective damping of the seal
K_{eff}	effective stiffness of the seal
K_{xx}, K_{yy}	direct stiffness of the seal in the x and y-direction
K_{xy}, K_{yx}	cross-coupled stiffness of the seal in the x and y-direction
C_{xx}, K_{yy}	direct damping of the seal in the x and y-direction
C_{xy}, C_{yx}	cross-coupled damping of the seal in the x and y-direction
H_{xx}, H_{yy}	direct mechanical impedance of the seal in the x and y-direction
H_{xy}, H_{yx}	cross-coupled mechanical impedance of the seal in the x and y-direction
$F_{bx_i}(t), F_{by_i}(t)$	forces generated by the i-th AMB in the x and y directions
$F_{sx_i}(t), F_{sy_i}(t)$	forces generated by the i-th seal in the x and y directions
$x_{s_i}(t), y_{s_i}(t)$	displacements of the rotor in correspondence of the i-th seal in the x and y-direction
$x_G(t), y_G(t)$	displacements of the rotor centre of gravity in the x and y-direction

Seals are widely used in turbomachinery to reduce the leakage flow through the rotor-stator clearances from the high-pressure to the low-pressure region [2]. The clearance around the rotor is required to avoid rotor-to-stator rub, excessive wear and friction heating. In the case of labyrinth seals, the blades, both for teeth-on-rotor (TOR), teeth-on-stator (TOS) and interlocking configurations, impose a ''tortuous passage'' to the fluid [3]. The unsteady pressure field produces lateral forces on the shaft that can affect the rotordynamic stability of the machine [4].

The main destabilizing effects are caused by the non-symmetrical unsteady circumferential pressure distribution and by the unbalanced circumferential forces due to the orbit motion of the rotor within the seal.

Similar to the oil and gas market, the technology trend in the power-generation field is to increase the power and efficiency of turbomachines. This leads to minimizing the clearance between the rotor and the stationary parts, which increases the risk of rubbing [5]. Thus, vibration analysis is a critical issue in the preliminary design to correctly predict the rotating machinery dynamic behaviour. The effects of the seals for gas applications are modelled by the linear reaction/motion equations [2] in the standard beam model used in rotordynamics: