specificity of each market. It could provide banks with the knowledge of efficient financial intermediation without the confines of collateral.

ROTOR DYNAMIC FORCES

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Any movement of the axis of rotation of the impeller-shaft relative to its casing induces fluid forces on the shaft and the casing, which in turn increase or decrease the rotor deflection or vibration. Contributions to these rotor dynamic forces can arise from seals, the rotor side space, the flow through the impeller, leakage flows, or the flow in the bearings themselves.

When designing high speed turbomachines the resulting fluid forces and rotor deflection must be known to begin with. For this reason rotor dynamic coefficients are required as input data for the prediction of rotor vibration. In general the dynamic behavior of the rotor system is described by linear models with time invariant system parameters in terms of differential equations, expressing the dynamic equilibrium of inertia, damping, stiffness and external forces.

Assuming a linear relationship of force and displacement and neglecting influences of high derivatives of the motion, this force-displacement model may be described with the following rotor dynamic stiffness, damping and mass matrices:

$$\begin{Bmatrix} -F_x \\ -F_y \end{Bmatrix} = \begin{bmatrix} K_{xx}K_{yx} & \\ -K_{xy}K_{yy} & \end{bmatrix}\begin{Bmatrix} x \\ y \end{Bmatrix} + \begin{bmatrix} D_{xx}D_{yx} & \\ -D_{xy}D_{yy} & \end{bmatrix}\begin{Bmatrix} \dot{x} \\ \dot{y} \end{Bmatrix} + \begin{bmatrix} M_{xx}M_{yx} & \\ -M_{xy}M_{yy} & \end{bmatrix}\begin{Bmatrix} \ddot{x} \\ \ddot{y} \end{Bmatrix} \quad (1)$$

For small motion around a centered position the cross-coupled terms of the damping and stiffness matrices become equal in magnitude based on their rotational symmetry. According to experimental findings the cross-coupled inertia terms of the mass matrix may be neglected and are set to be zero. However, the direct inertia term M cannot be neglected except in cases where laminar seal flow dominates, e.g. for slide bearings. Thus, the coefficient matrices can be simplified in the case of small concentric perturbations as follows:

$$\begin{Bmatrix} -F_x \\ -F_y \end{Bmatrix} = \begin{bmatrix} K & K_c \\ -K_c & K \end{bmatrix} \begin{Bmatrix} x \\ y \end{Bmatrix} + \begin{bmatrix} D & D_c \\ -D_c & D \end{bmatrix} \begin{Bmatrix} \dot{x} \\ \dot{y} \end{Bmatrix} + \begin{bmatrix} M & 0 \\ 0 & M \end{bmatrix} \begin{Bmatrix} \ddot{x} \\ \ddot{y} \end{Bmatrix} \qquad (2$$

Labyrinth seals are often the major source for radial fluid force in turbomachinery and may, in the worst case, destabilize their rotors. On the other hand, the seals are the key elements controlling the leakage flow of turbomachines. Optimizing leakage flow and friction will contribute to improving the efficiency of the machines. In practice one has to find a compromise between the often conflicting demands arising from efficiency improvement and rotor dynamic stability.

For seals the coefficient matrices are independent of the type of rotor motion but strongly dependent upon the operational conditions of the seal, such as the leakage flow or the rotor speed and also depend largely on the preswirl of the flow entering the seal. These particular effects were systematically investigated in a series of research work at the Swiss Federal Institute Zurich. On the basis of experimentally measured forces under varying operational conditions of the seal and rotor motion the stiffness, damping and mass matrices of equation (2 could be determined. Selected results of these studies will be used as reference for comparison with the results of the study being presented.

To calculate the rotor dynamic behavior of turbomachinery seals well-validated tools, such as the bulk-flow theory of Childs, are available. The difficulty in applying these tools very often lies in making a preliminary estimate of the flow conditions into the seal part. To avoid this difficulty a procedure was tested to determine rotor dynamic coefficients on the basis of numerical flow calculations using a commercial CFD code. Choosing an adequate, rotating co-ordinate system, rotor dynamic coefficients could be determined directly through the integration of calculated pressure distributions.