Development of Efficient 3D Models for Tilting Pad Journal Bearings

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Abstract. This paper mainly focuses on the development of efficient three-dimensional (3D) models of TPJBs, able to in parallel simulate both the rotor dynamics behaviour and lubricant supply plant. The proposed modelling approach tries to obtain a good compromise between the typical accuracy of standard 3D models and the high numerical efficiency of simpler and less accurate models. In this work, the whole model has been developed and validated in collaboration with Nuovo Pignone General Electric S.p.a. which provided the required technical data. In particular, the experimental data are referred to a suitable lube oil console system, built at the GE testing center located in Massa-Carrara (MS, Italy) for the verification of such components.

1 Introduction

The authors aim to develop a model of Tilting Pad Journal Bearing including both the fluid dynamic and rotordynamic characteristics and preserving a competitive numerical efficiency. Therefore, the goal of this new model is to replace (partially or completely) both the classical lumped parameter models (poorly accurate for the modern industry requirements) and the complete 3D models (too time consuming). The general layout of the model, consists of the following parts: TPJB model (including oil film model and pad dynamic model), rotor model, lubricant supply model (including duct model and sump model) and lube oil console model. The pad model and the lubricant supply model are repeated \( n \) times depending on the considered Tilting Pad Journal Bearings, making the model highly modular. The whole model is built in the COMSOL Multiphysics \( \text{©}4.2a \) and MATLAB\( \text{®}R2011a \) environments.
2 General Architecture of the Model

An innovative feature of the developed architecture is the modularity of the approach, indeed, the behaviour of the pad/sump module can be repeated \( n \) times (where \( n \) represents the pads and sumps number), allowing the simulation of different bearings by adding or reducing the pad number.

To integrate different numerical solvers, the model solving architecture is divided into two parts: Partial Differential Equations (PDE) problem (the Reynolds equation for the oil film model is solved through the COMSOL solvers) and Ordinary Differential Equations (ODE) problem (in MATLAB, both the motion equations of rotor and pads and the 1D lumped equations of the lubricant supply model are solved).

This approach analyses separately the mechanical part and the fluid dynamics one reaching a good trade off between numerical efficiency and accuracy. Such decoupling is possible if the temporal integration step \( \Delta t \) is small enough; under this assumption, within the temporal interval \( \Delta t \), the fluid dynamic problem is solved as steady-state (a simplified fluid dynamic model is used, neglecting turbulent state flow). This hypothesis is acceptable if the oil film thickness is thin enough (compared to the other directions [2]).

An implicit, variable order and variable step ODE-solver, suitable for stiff problems (conventionally defined as *ode15s*), is used to simulate the time-dependent
part [5], [6].

The CFD oil film model (steady) reaches the solution using an iterative non linear solver type based on Newtonian methods [7]. The linear problem arising from the non linear solver are solved through an iterative method (BicGStab [8]).

As regards the mesh, it has been realized by means of bidimensional elements (this assumption is verified by the hypothesis underlying the Reynolds equation), to maintain acceptable performances and accurate results [20].

3 Model Description

In the following sections the different parts of the model will be described in detail.

In Figure 1 and 2, the $S_A$, $S_{pad}$ represent respectively the rotor and pad surfaces, while $S_1$, $S_3$ are the inlet and outlet oil sections of the control volume; finally $S_2$, $S_4$ are the lateral leakage surfaces of the control volume.

3.1 TPJB Model

In this research paper, basing on the technical data provided by Nuovo Pignone General Electric S.p.a., the CFD analysis is performed on the Journal Bearing JB200 (where the rotor diameter is 200 mm, the external diameter is 395 mm and the thickness is 170 mm) with 4 tilting pads set in the “X” configuration. According to the model architecture, the pad model is divided into: oil film model and pad dynamic model.
The Oil Film Model

The model inputs are defined by the rotor and pad kinematical quantities

\[
\mathbf{x_A} = (x_A, y_A, z_A, \alpha_A, \beta_A, \gamma_A)^T, \quad \dot{\mathbf{x_A}} = (\dot{x_A}, \dot{y_A}, \dot{z_A}, \dot{\alpha_A}, \dot{\beta_A}, \dot{\gamma_A})^T,
\]

\[
\mathbf{x_{pad}} = (\gamma_{pad}) \quad \text{and} \quad \dot{\mathbf{x_{pad}}} = (\dot{\gamma}_{pad}).
\]

The model is supposed time-independent and thus a steady-state problem is solved.

The oil film thickness \( h \) is given by:

\[
h = h(\mathbf{x_A}, y_A, z_A, \alpha_A, \beta_A, \gamma_{pad}, R_{p}, R_{rot}, \varphi, \chi)
\]  

(1)

where \( \varphi, \chi \) are the angles describing the oil film thickness orientation in a spherical reference system, \( R_{p} \) is the internal pad radius and \( R_{rot} \) is the rotor radius.

The forces that the oil film acts on rotor and pads can be calculated as follows:

\[
F_A = \int_{S_A} \sigma \mathbf{n_A} dS_A, \quad F_{pad} = \int_{S_{pad}} \sigma \mathbf{n_{pad}} dS_{pad}.
\]  

(2)

where \( \sigma \) is the fluid stress tensor and \( \mathbf{n_A}, \mathbf{n_{pad}} \) are the outgoing unitary vectors respectively from the rotor surface \( S_A \) and the pad surface \( S_{pad} \).

The momenta acting on the rotor and pads can be calculated according to the following equations:

\[
M_A = \int_{S_A} \mathbf{r_A} \times \sigma \mathbf{n_A} dS_A, \quad M_{pad} = \int_{S_{pad}} \mathbf{r_{pad}} \times \sigma \mathbf{n_{pad}} dS_{pad}.
\]  

(3)

The oil flow rates \( Q_{in} \), \( Q_{out} \), \( Q_{bord,l} \) and \( Q_{bord,r} \) are described as follows:

\[
Q_{in} = \int_{S_1} \rho v \mathbf{n}_1 dS_1, \quad Q_{out} = \int_{S_2} \rho v \mathbf{n}_2 dS_2,
\]

\[
Q_{bord,l} = \int_{S_3} \rho v \mathbf{n}_3 dS_3, \quad Q_{bord,r} = \int_{S_4} \rho v \mathbf{n}_4 dS_4.
\]  

(4)

where \( v \) represents the oil film velocity and \( \mathbf{n}_1, \mathbf{n}_2, \mathbf{n}_3, \mathbf{n}_4 \) are the outgoing unitary vectors respectively from surfaces \( S_1, S_2, S_3 \) and \( S_4 \) (see Figure 2).

The model outputs are the forces \( F_{x,A}, F_{y,A}, F_{z,A} \) and momenta \( M_{x,A}, M_{y,A}, M_{z,A} \) acting on the rotor (subscript \( A \)) and on the pad \( M_{z,\text{pad}} \) (subscript \( \text{pad} \)) and the inlet/outlet flow rates \( Q_{in} \), \( Q_{out} \) from the pad edges and the flow rate \( Q_{bord,l} \) and \( Q_{bord,r} \) exiting from the pad side edges.

Pad Dynamics Model

The Pad Dynamics Model describes the dynamics of rotation around the cylindrical pad pivot (then presents a single DOF). The inputs are represented by the momentum \( M^i_{z,\text{pad}} \) acting on the pad. The geometry of the pad is shown in Figure 2.

The pad motion equation is given by:

\[
I^i_{pad} \gamma^i_{pad} = M^i_{z,\text{pad}};
\]  

(5)

where \( \gamma \) indicates the rotation around the pad axis \( z \), \( I^i_{pad} \) defines the pad polar moment of inertia, and \( M^i_{z,\text{pad}} \) is calculated by equation 3.

The outputs are the pad position \( \gamma_{pad} \) and speed \( \dot{\gamma}_{pad} \).
3.2 The Rotor Model

The proposed model is based on a centrifugal compressor (called 3BCL1005) used to liquefy natural gas, directly coupled to a condensing-type steam turbine driver, where the characteristics are provided by Nuovo Pignone General Electric S.p.a. and is tested in a GE testing center located in Massa-Carrara (MS, Italy).

The inputs are defined by the forces \( F_{x,A}, F_{y,A}, F_{z,A} \) and momenta \( M_{x,A}, M_{y,A}, M_{z,A} \) acting on the rotor and possible external loads and torques \( (F_e, M_e) \).

The dynamics of the rotor considers the following 6 DOFs: rotation around the symmetry axis \((z\text{-axis})\), translation along the symmetry axis \((z\text{-axis})\) and translations and rotations around the other two axes \((x\text{-axis} \text{ and } y\text{-axis})\) approximating the rotor model to a rigid disk gyroscopic model (since the polar moment of inertia \( I_p \) is greater than the transversal moment of inertia \( I_t \)).

Hence, the motion equations (6), (7), (8) can be written:

\[
I_p \ddot{\gamma} + M_{z,e} + C_{mot} + M_{z,e} = \sum_{i=1}^{n} M_i^{z,A} + \sum_{i=1}^{n} F_i^{z,A}, \tag{6}
\]

\[
m_{A} \ddot{z} = \sum_{i=1}^{n} F_i^{z,A} + F_{z,e}, \tag{7}
\]

\[
M_{k} \begin{pmatrix} \ddot{x}_A \\ \ddot{y}_A \\ \ddot{z}_A \\ \dot{\alpha}_A \\ \dot{\beta}_A \\ \dot{\gamma}_A \end{pmatrix} = \sum_{i=1}^{n} \begin{pmatrix} \sum_{i=1}^{n} F_i^{x,A} + F_{x,e} \\ \sum_{i=1}^{n} M_i^{y,A} + M_{y,e} \\ \sum_{i=1}^{n} M_i^{z,A} + M_{z,e} \end{pmatrix}, \tag{8}
\]

where, \( n \) defines the pad number, \( m_A \) the rotor mass, \( C_{mot} \) is the motor torque, \( g \) is the modulus of the gravity acceleration vector, \( M_{k} \) and \( G_{k} \) represent respectively the mass and gyroscopic effect matrices.

Finally, \( F_{x,A}, F_{y,A}, F_{z,A}, M_{x,A}, M_{y,A}, M_{z,A} \) are the forces and momenta acting on the rotor and \( F_{x,e}, F_{y,e}, F_{z,e}, M_{x,e}, M_{y,e}, M_{z,e} \) are the possible external forces and momenta.

The outputs of the model are the rotor position \( x_A, y_A, z_A, \alpha_A, \beta_A, \gamma_A \), and velocities \( \dot{x}_A, \dot{y}_A, \dot{z}_A, \dot{\alpha}_A, \dot{\beta}_A, \dot{\gamma}_A \).

3.3 Lubricant Supply Model

The lubricant supply model has been realized following the Bond-Graph approach for the modelling of general multiphysics system [18], [19]. The lubricant supply model represents the supply and leakage of the system and it is divided in two components: the Duct model and the Sump model.

Duct Model The Duct Model describes the supply and leakage flow rates inside the sump.

The inputs are the sump pressure \( p_i \), the supply pressure \( p_s \) and the environment
This model is based on the Bond-Graph theory of lumped parameters modelling [18], [19], which implements elements of \( T \) (through) type (resistive elements). It represents the leakages and supply. The duct expressions are then shown; \( Q'_{orif} \) and \( Q'_{lk} \) are defined according to the following laws:

\[
Q'_{orif} = A_0 C_d \sqrt{\frac{2}{\rho} (p_s - p)^}, \quad Q'_{lk} = A_{lk} C_{dlk} \sqrt{\frac{2}{\rho} (p^i - p_{env})},
\]

where \( A_0 \) is the duct section, \( C_d \) is the duct flow coefficient, \( A_{lk} \) is the sump section, \( C_{dlk} \) is the sump flow coefficient, \( \rho \) is the lubricant density and \( p_{env} \) is the environment pressure.

The outputs are the supply and leakage flow rates, respectively \( Q'_{orif}, Q'_{lk} \).

**Sump Model** The Sump Model describes the balance of the flow rates inside the sump. The inputs are defined by the the inlet/outlet flow rates \( Q'_{in}, Q'_{out} \) from the pad edges, the supply and leakage flow rates, respectively \( Q'_{orif}, Q'_{lk} \). The sump scheme is shown in Figure 2; in particular, a sump is interposed between two adjacent pads. This model is based on the Bond-Graph theory of lumped parameters modelling [18], [19], which implements elements of \( A \) (across) type (capacitive elements). The constitutive equation of this element is:

\[
\frac{dp^i}{dt} = \frac{K}{V o l} \left( Q'_{in} - Q'_{lk} - Q'_{out} + Q'_{orif} \right),
\]

in which \( K \) is the Bulk module, \( V o l \) is the sump volume obtained by geometric measurements. This equation can be used considering that in this area (inside the bearing) the pressure is approximatively constant in the whole volume because the effects produced by the rotor on the sump are neglected. \( Q'_{in} \) is the input flow rate from the previous pad, \( Q'_{lk} \) represents the flow rate losses, \( Q'_{out} \) is the output flow rate from the sump to the next pad, \( Q'_{orif} \) is the flow rate introduced from outside.

The output of the model is the sump pressure \( p^i \).

### 4 Experimental Data Description

The Tilting Pad Journal Bearing model has been carried out and validated by the technical data of the \( JB200 \) (provided by Kingsbury), having 4 tilting pads with a “X” configuration. The TPJBs are installed on a 3BCL1005 centrifugal compressor provided by \textit{Nuovo Pignone General Electric S.p.a.}. The main features and operating conditions of the TPJB and of the rotor are shown in Table 1.

The TPJB model has been validated through a comparison with experimental data provided by General Electric Oil & Gas and coming from experimental testing of the entire system described in Figure 3 carried out at the GE testing center located in Massa-Carrara (MS, Italy).
Table 1. Bearing geometry and operating conditions.

![Simple scheme of the lubrication testing system in Massa-Carrara.](image)

The experimental data are obtained evaluating the measured drop pressure $p_{meas} - p_{env}$ and flow rate $Q_{meas}^{orif}$ (single value associated to the whole bearing considered) at a steady state condition defined by temperature $T_{meas}$ and rotational velocity of the rotor $\dot{\gamma}_{meas}^A$. In addition, two levels of temperatures are considered: 50°C, which is the nominal operating temperature, and 65°C, considered high temperature. The results are referred to a single Tilting Pad Journal Bearing (the one nearest to the rotor center of mass) exclusively for reasons of synthesis; the measures, however, were made on both the TPJBs and the test results concern the whole system.
The measured flow rate $Q_{\text{meas \ orif}}$ concerns the whole tilting pad journal bearing and consequently is validated by the sum of the flow rates adduced on the $n$ sumps:

$$Q_{\text{orif}} = \sum_{i=1}^{n} Q_{i\text{orif}}$$  \hspace{1cm} (11)

$Q_{\text{orif}}$ is the variable to be considered in the model and will be compared to $Q_{\text{meas \ orif}}$. The ideal test case are obtained applying both the ideal monodimensional formula [17] and the GE NP formula:

$$Q_{\text{id \ orif}} = A_0 C_d \sqrt{\frac{\rho}{2}} (p_{s\text{meas}} - p_{\text{env}}),$$

$$Q_{\text{GENP \ orif}} = D^2 k_{\text{GENP}} \frac{\beta^2}{\sqrt{1-\beta^2}} \sqrt{\frac{(p_{s\text{meas}} - p_{\text{env}})}{\rho}},$$  \hspace{1cm} (12)

where $Q_{\text{id \ orif}}$ and $Q_{\text{GENP \ orif}}$ are the computed flow rates considering the whole bearing as an ideal orifice, $p_{s\text{meas}} - p_{\text{env}}$ is the measured drop pressure, $C_d$ is the flow coefficient, $A_0$ is the orifice area, $D$ is the pipe diameter, $\beta$ is the ratio between the orifice and pipe diameters and $k_{\text{GENP}}$ is a numeric adimensional constant. The first equation of the 12 is usually used to model the Tilting Pad Journal Bearings behaviour from the hydraulic point of view; while the second equation is currently used within Nuovo Pignone General Electric S.p.a. for hydraulic modelling of the TPJBs. The values of the above mentioned parameters are provided by Nuovo Pignone General Electric S.p.a..

5 Validation and Results

In this section, the authors present the comparison between the drop pressure $p_{s\text{meas}} - p_{\text{env}}$ curve and the flow rate $Q_{\text{orif}}$ curve (simulated and experimental). In Figure 4 the flow rate results obtained with the proposed model are shown, compared to the previous experimental data and the ideal and GE NP formulas. The flow rates $Q_{\text{meas \ orif}}$, $Q_{\text{GENP \ orif}}$ and $Q_{\text{id \ orif}}$ concern the whole Tilting Pad Journal Bearing and consequently are validated respect to the sum of the flow rates ($Q_{\text{orif}} = \sum_{i=1}^{n} Q_{i\text{orif}}$) adduced on the $n$ sump.

$Q_{\text{id \ orif}}$ is obtained by the interpolation of the points $Q_{\text{orif}}$, which are numerically derived from the whole model (achieving good results in terms of accuracy). The $Q_{\text{id \ orif}}$ fits better to the measured data, compared to the ideal and GE NP curves. The maximum relative errors shown in Figure 4 are highlighted in Table 2 (it is worth to note that the maximum relative errors can be found at high $p_{s\text{meas}} - p_{\text{env}}$ and $Q_{\text{orif}}$).
Fig. 4. Proposed model results compared to experimental data for different formulae comparisons.

<table>
<thead>
<tr>
<th>proposed model</th>
<th>GE NP model</th>
<th>ideal model</th>
</tr>
</thead>
<tbody>
<tr>
<td>50°C 3.36%</td>
<td>13.43%</td>
<td>38.59%</td>
</tr>
<tr>
<td>65°C 3.77%</td>
<td>10.41%</td>
<td>29.91%</td>
</tr>
</tbody>
</table>

Table 2. Maximum relative errors.

6 Conclusions and Future Developments

An innovative TPJB modelling approach is described in this paper to increase the result accuracy with reduced computational time. The coupling between the fluid dynamics and rotordynamics effects is realized with the interaction between a 3D modelling and model parts still treated with 1D equations (the 1D formulation is implemented in the submodel that less affect the model accuracy). The proposed model aims to obtain a compromise between the accuracy of complete 3D models and the efficiency of lumped parameters models. The whole model has been developed and validated in collaboration with Nuovo Pignone General Electric S.p.a. which provided the required technical data and the test results. The results show a good agreement with experimental data (good accuracy), while the computation time are reduced compared to a totally 3D model (good efficiency). The following future developments stem from this research activity:

- analysis of the heat exchange inside the oil film and other components;
- implementation of flexibility in the rotor and pads models;
- implementation of the proposed model in rotor trains and more complex supply plants.

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References