Computer-Aided Design of Pocket Elliptical Journal Bearings, Part 2: Application*

Fabrizio STEFANI**
** Department of Mechanical Engineering, University of Genoa, Via all’Opera Pia 15/A, 16154, Genoa, Italy
E-mail: stefani@unige.it

Abstract
In a companion paper, part 1, a computer-aided design method for pocket elliptical bearings has been developed. Its application is discussed in the current paper, which mutually related goals are: to understand how bearing design and particularly pocket geometry affect the bearing performance, to determine optimal values of the driving variables that control bearing profile and pocket width, to choose a suitable supply groove layout.

As explained in part 1, the developed method does not include an automatic optimization tool. Therefore, in order to reduce the number of the variables that control the bearing geometry involved in the optimization task, the clearance of the assembled bearing (pocket clearance) is kept constant. Hence, the optimization of the bearing shape is performed by changing pocket opening, pad eccentricities, pocket width, lubricant supply mode.

Key words: Journal Bearing, Computer Aided Design, Lubrication, Finite Element Method, Optimization, Turbomachinery

1. Introduction
In a companion paper, part 1, a computer-aided design method for pocket elliptical bearings has been developed. Its application is discussed in the current paper, which mutually related goals are: to understand how bearing design and particularly pocket geometry affect the bearing performance, to determine optimal values of the driving variables that control bearing profile.

As explained in part 1, the developed method does not include an automatic optimization tool. Therefore, in order to reduce the number of the variables that control the bearing geometry involved in the optimization task, the clearance of the assembled bearing (pocket clearance) is kept constant. Hence, the optimization of the bearing shape is performed by changing pocket opening, pad eccentricities and the design variants presented in part 1.

Nomenclature

\[ \begin{align*}
  d & \quad \text{shim thickness, mm} \\
  c_h & \quad \text{barrier clearance, mm} \\
  c_c & \quad \text{pocket clearance, mm} \\
  c_L & \quad \text{lubricant specific heat, J/(kg °C)} \\
  c_p & \quad \text{pad clearance, mm} \\
  C & \quad \text{relative clearance} \\
  C_c & \quad \text{relative pocket clearance}
\end{align*} \]
$C_p$ relative pad clearance
$c_{V1}$ vertical clearance of bottom pad, mm
$c_{V2}$ vertical clearance of top pad, mm
$e_x$ journal x coordinate, mm
$e_y$ journal y coordinate, mm
$e_1$ bottom pad eccentricity, mm
$E_1$ bottom pad ellipticity
$e_2$ top pad eccentricity, mm
$E_2$ top pad ellipticity
$g$ gravitational acceleration, m/s$^2$
$h$ film thickness, µm
$h_{min}$ minimum film thickness, µm
$k_L$ lubricant conductivity, W/(m °C)
$L$ length, mm
$L_b$ barrier length, mm
$L_s$ supply groove length, mm
$m$ journal mass, kg
$m_1$ bottom pad preload
$m_2$ top pad preload
$M_c$ non-dimensional critical mass
$O_c$ bearing center
$O_{i1}$ pad i center
$O_{i2}$ journal center
$P$ heat dissipation, kW
$p$ pressure, MPa
$p_{max}$ peak pressure, MPa
$p_s$ supply pressure, MPa
$Q$ supply flow rate, l/s
$Q_{in}$ inlet flow, l/s
$Q_{out}$ outlet flow (side loss), l/s
$Q_{i1}$ upstream groove inlet flow, l/s
$Q_{i2}$ downstream groove inlet flow, l/s
$R$ journal radius, mm
$R_a$ average surface roughness of bearing and journal surface, µm
$R_p$ pocket radius, mm
$R_p$ pad radius, mm
$s$ lathe spindle offset (2nd turning operation), mm
$t$ cut depth, mm
$T$ temperature, °C
$T_b$ average bush temperature, °C
$t_s$ supply groove average depth, mm
$T_s$ average shaft temperature, °C
$t_{H1}$ cut depth @ horizontal diameter, mm
$t_{H1}$ cut depth @ bottom pad center, mm
$t_{H2}$ cut depth @ top pad center, mm
$T_0$ lubricant reference temperature, °C
$T_s$ supply temperature, °C
$T_{out}$ outlet temperature, °C
$x$ vertical coordinate in the bearing reference system, m
$y$ horizontal coordinate in the bearing reference system, m
$Y$ performance indicator
2. Method

In design practice suitable non-dimensional charts, like the classic Raimondi and Boyd (1) plots for more conventional (circular, partial arc, etc.) bearings, might be used, in order to avoid the use of the computer program for each application case.

Nevertheless, in the case of pocket bearings, it would be very difficult (if not impossible) both to draw and to read the charts, as the number of the independent non-dimensional parameters that rule the lubrication problem is too much high for a graphical parametric representation. Dimensional analysis or, equivalently, the application of Buckingham’s $\pi$ theorems show that, for a given supply pressure $p_s$ and groove configuration, i.e. for constant $p_s$, $\vartheta_s$, $\beta_s$ and $L_s$ (Fig. 5 of part 1), usually locked in bearing design charts, the hydrodynamic problem still depends on 8 non-dimensional variables: 4 gap driving parameters (relative clearance $C$, bottom pad ellipticity $E_1$, pocket angle $\alpha_t$, top pad ellipticity $E_2$), the ratios between bearing and barrier axial length to radius ($L/R$ and $L_b/R$, respectively), the Sommerfeld number and the mean Reynolds number $\Omega_1$ (lubrication regime is usually non-laminar).

In addition the hydrodynamic problem is always coupled with the thermal problem and so the number of leading parameters further increases. In other words, a generalized study that takes into account all of the design variants is very hard to carry out and each case (for given size and working conditions) must be calculated independently.

Therefore, in order to test the procedure explained in part 1, an existing pocket elliptical bearing for turbogenerators is redesigned and its actual design is checked for modifications. Such case study is described in §3, where the corresponding input data are also given. In §4 an initial design is set by means of the geometrical design sequence. Afterwards, in §5 parametric analyses are presented to show the influence of the driving variables and design variants. Finally, on the basis of the previous calculations, two optimal designs are chosen and evaluated (§6).

3. Case study

All of the nominal independent data assumed a priori in the following are relevant to an

---

$Y_{ref}$ reference performance indicator

$Y_g$ global performance indicator, %

$Y_{gs}$ global performance indicator including stability, %

$z$ coordinate along journal axis in bearing reference system, m

$W$ journal load, N

$\alpha$ pocket half-opening angle, deg

$\alpha_t$ pocket opening angle, deg

$\beta_t$ lubricant temperature-viscosity coefficient, $1/\text{°C}$

$\beta_s$ supply groove circumferential extent, deg

$\gamma$ weight of the performance indicator

$\varrho$ polar coordinate, deg

$\vartheta_s$ supply groove circumferential location, deg

$\mu_L$ lubricant viscosity, Pa s

$\mu_0$ lubricant reference viscosity, Pa s

$\rho_L$ lubricant density, kg/m$^3$

$\omega$ shaft rotation speed, rpm

$\Omega$ mass-square speed parameter, kg/s$^2$

$\Omega_c$ critical parameter, kg/s$^2$
actual pocket elliptical journal bearing, which is located near the compressor of a gas turbine power plant (near the turbine a tilting pad journal bearing is usually preferred). The existing bearing is designed with “barriers” (length \( L_b > 0 \), §3.2 of part 1) and it complies with design B (§2 of part 1), i.e. the ellipticities of both pads are equal. Its journal nominal diameter is 425 mm, so that it can be considered a medium-sized bearing (turbogenerator journal diameters are usually in the range 100-1000 mm). The bearing is equipped with two feed grooves, opposite with reference to the load line, but it is fed only through the groove upstream the load-carrying film region (groove layout of Fig. 4 (c) in part 1, supply mode: inlet 1).

Table 1 reports all of the data required by FEM analyses (bearing nominal working conditions, lubricant data and bearing dimensions), with the exception of the gap driving variables that must be determined by means of geometrical design. The working conditions include journal load \( W \) as well as feed temperature \( T_s \) and supply pressure \( p_s \). The chosen thermal model (§4.2, part 1) also requires the average bush and shaft temperatures, \( T_B \) and \( T_S \). The lubricant is ISO VG 46 oil (parameters \( \mu_0 \), \( T_0 \) and \( \beta_L \) are used to simulate the variation of viscosity with temperature). Among bearing dimensions, feed grooves are characterized by means of the circumferential coordinate \( \beta_s \) of their center, the angular size \( \beta_s \) and the axial length \( L_s \) (Fig. 5, part 1). Their average thickness \( t_s \) is used only to take into account heat dissipation in the grooves by means of a simple model \(^{(3)}\).

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Symbol [unit]</th>
<th>Nominal/basic value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Length ( L ) [mm]</td>
<td>373</td>
<td></td>
</tr>
<tr>
<td>Barrier length ( L_b ) [mm]</td>
<td>35</td>
<td></td>
</tr>
<tr>
<td>Supply groove circumferential extent ( \beta_l ) [deg]</td>
<td>12.75</td>
<td></td>
</tr>
<tr>
<td>Supply groove axial length ( L_a ) [mm]</td>
<td>344</td>
<td></td>
</tr>
<tr>
<td>Supply groove circumferential location ( \beta_l ) [deg]</td>
<td>96.375</td>
<td></td>
</tr>
<tr>
<td>Supply groove average depth ( t_l ) [mm]</td>
<td>0.35</td>
<td></td>
</tr>
<tr>
<td>Supply temperature ( T_s (°C) )</td>
<td>41</td>
<td></td>
</tr>
<tr>
<td>Supply pressure ( p_s ) [MPa]</td>
<td>0.67</td>
<td></td>
</tr>
<tr>
<td>Load ( W ) [N]</td>
<td>105013</td>
<td></td>
</tr>
<tr>
<td>Shaft rotation speed ( n ) [rpm]</td>
<td>3000</td>
<td></td>
</tr>
<tr>
<td>Bush average temperature ( T_B (°C) )</td>
<td>50</td>
<td></td>
</tr>
<tr>
<td>Shaft average temperature ( T_S (°C) )</td>
<td>50</td>
<td></td>
</tr>
<tr>
<td>Lubricant reference viscosity ( \mu_0 ) [Pa s]</td>
<td>0.02635</td>
<td></td>
</tr>
<tr>
<td>Lubricant reference temperature ( T_0 (°C) )</td>
<td>50</td>
<td></td>
</tr>
<tr>
<td>Lubricant temperature-viscosity coefficient for the law ( \mu(T) = \mu_0 \cdot \exp\left[-\beta(T-T_0)\right] )</td>
<td>( \beta ) [°C(^{-1})]</td>
<td>0.0272</td>
</tr>
<tr>
<td>Lubricant density ( \rho ) [kg m(^{-3})]</td>
<td>890</td>
<td></td>
</tr>
<tr>
<td>Lubricant specific heat ( c_p ) [J kg(^{-1})°C(^{-1})]</td>
<td>2000</td>
<td></td>
</tr>
<tr>
<td>Lubricant conductivity ( \kappa ) [W m(^{-1})°C(^{-1})]</td>
<td>0.14</td>
<td></td>
</tr>
<tr>
<td>Average surface roughness ( R_a ) [µm]</td>
<td>5</td>
<td></td>
</tr>
</tbody>
</table>

Although all of the dimensions in Table 1 are independent design variables with reference to the lubrication problem, their values are typical for fixed-pad journal bearings for similar applications \(^{(4)}\)-\(^{(5)}\) so that their choice is not discussed in the present paper.

A comprehensive THD analysis of the same bearing, including a comparison with data measured during the turbomachine operation, has been presented in \(^{(6)}\), where the predictions of a suitable model agree with the measurements satisfactorily. Therefore the same model is used in all of the following calculations.

For the present case study, FEM analyses take advantage of a 1000 node mesh (100 x 10, along circumferential and axial directions, respectively) of first-order isoparametric elements. The highly non-linear problem relevant to a single analysis, on the average, requires 2000 time-steps to be solved. The corresponding computational time is about 40 min on a conventional PC with average performance (AMD Athlon 3 GHz processor).
4. Geometrical design evaluation

An initial geometrical design, i.e. a good starting point for a subsequent optimization, compliant with the design of the actual bearing, is required. The driving variables chosen for this purpose, from which the remaining can be found by means of the geometrical design sequence (§4.1, part 1), are given in Table 2.

Table 2 Design variables and performance indicators for the initial design

<table>
<thead>
<tr>
<th>Parameter type</th>
<th>Dimension</th>
<th>Symbol [unit]</th>
<th>Nominal value</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Driving dimension</strong></td>
<td>Journal radius</td>
<td>R [mm]</td>
<td>212.5</td>
</tr>
<tr>
<td></td>
<td>Relative pocket clearance</td>
<td>Cc [1000]</td>
<td>1.7</td>
</tr>
<tr>
<td></td>
<td>Bottom pad ellipticity</td>
<td>E1</td>
<td>3.12</td>
</tr>
<tr>
<td></td>
<td>Pocket opening angle</td>
<td>α [deg]</td>
<td>110.0</td>
</tr>
<tr>
<td></td>
<td>Top pad ellipticity (preload)</td>
<td>E2 [mm]</td>
<td>3.12 (0.08)</td>
</tr>
<tr>
<td><strong>Driven dimension</strong></td>
<td>Pocket clearance</td>
<td>c1 [mm]</td>
<td>0.361</td>
</tr>
<tr>
<td></td>
<td>Pocket radius</td>
<td>R [mm]</td>
<td>212.861</td>
</tr>
<tr>
<td></td>
<td>Cut depth</td>
<td>t [mm]</td>
<td>0.766</td>
</tr>
<tr>
<td></td>
<td>Pad radius</td>
<td>R [mm]</td>
<td>213.627</td>
</tr>
<tr>
<td></td>
<td>Pad clearance</td>
<td>c2 [mm]</td>
<td>1.127</td>
</tr>
<tr>
<td></td>
<td>Relative pocket clearance</td>
<td>Cc [1000]</td>
<td>5.3</td>
</tr>
<tr>
<td></td>
<td>Bottom pad eccentricity</td>
<td>e1 [mm]</td>
<td>1.330</td>
</tr>
<tr>
<td></td>
<td>Top pad preload</td>
<td>Mx</td>
<td>1.18</td>
</tr>
<tr>
<td></td>
<td>Top pad eccentricity</td>
<td>e2 [mm]</td>
<td>0.966</td>
</tr>
<tr>
<td></td>
<td>Lathe spindle offset (G2 turning operation)</td>
<td>s [mm]</td>
<td>0.282</td>
</tr>
<tr>
<td></td>
<td>Shim thickness</td>
<td>d [mm]</td>
<td>2.096</td>
</tr>
<tr>
<td></td>
<td>Horizontal cut depth</td>
<td>t2 [mm]</td>
<td>0.766</td>
</tr>
<tr>
<td></td>
<td>Cut depth (g) bottom pad center</td>
<td>t1 [mm]</td>
<td>0.564</td>
</tr>
<tr>
<td></td>
<td>Cut depth (g) top pad center</td>
<td>t1 [mm]</td>
<td>0</td>
</tr>
<tr>
<td><strong>Performance indicators</strong></td>
<td>Minimum film thickness</td>
<td>hmin [mm]</td>
<td>110.0</td>
</tr>
<tr>
<td></td>
<td>Peak pressure</td>
<td>ps [MPa]</td>
<td>5.1</td>
</tr>
<tr>
<td></td>
<td>Maximum temperature</td>
<td>T [°C]</td>
<td>98.3</td>
</tr>
<tr>
<td></td>
<td>Heat dissipation</td>
<td>P [W]</td>
<td>146.8</td>
</tr>
<tr>
<td></td>
<td>Supply flow rate</td>
<td>Q [l/min]</td>
<td>3.0</td>
</tr>
<tr>
<td></td>
<td>Critical mass</td>
<td>Mc</td>
<td>85.7</td>
</tr>
<tr>
<td></td>
<td>Critical stability parameter</td>
<td>ΘC [kg/m]</td>
<td>2.3·10^9</td>
</tr>
</tbody>
</table>

The sequence, which steps are the rows of Table 2, starts from the journal radius R, which is known from the machinery requirements. Then, as the actual bearing is designed with barriers, Cc is chosen rather than Cp. As is usual for bearings with diameter lower than 450 mm, the lowest value in the permissible range 1.7/1000-2.2/1000 is chosen for the sake of safety, in agreement with the data provided by the manufacturer.

The initial value of the bottom pad ellipticity is also set accordingly (E1=3.12). It must be greater than 2, as explained in the theoretical part, but not oversized in order to avoid excessive clearance. The check on the resulting Cp reported below confirms that E1 is chosen reasonably.

The most usual value (110 deg) advised by manufacturers for the nominal pocket opening angle α is chosen. Design B is adopted so that the nominal value of E2 is c2/c1 (or, equivalently, m2 = t/c2), where c2, c1 and t are computed later. More straightforwardly, E2=E1 as design B implies equal vertical clearances for both pads (cV1=cV2).

The driven variables, also reported in Table 2, are calculated by means of the above-cited geometrical design sequence.

Particularly, the calculated value of pad clearance (Cp=5.3/1000) is greater than the values suggested by Kingsbury formula but it is satisfactory. Indeed, Equations (20) and (21) (part 1) give Cp=2.24/1000 and Cp=4/1000, respectively. As already explained, an higher, usually lower than 6/1000, is advisable in order to take into account a potential differential thermal expansion of the cinematic pair.

The resulting value of the bottom pad preload m1 (1.18, as shown in Table 2) is much higher than usual values for elliptical bearings, which range from 0.5 to 0.8 (2).
preloads do not further improve the dynamic performance of elliptical bearings, as they become inherently stable as soon as preload value exceeds 0.8. The high bottom pad preload in the case at the hand is due to the above-explained dimensioning, which is aimed to keep in pocket bearings the ellipticities (ratios between horizontal and vertical clearances) of conventional two-lobe bearings for both pads, in order to preserve large lubricant supply and optimal fluid dynamics in the feed grooves. To the same goal the shape of grooves is often complex, i.e. they are different from cylindrical recesses or simple slots in many cases. On the contrary, as the top lobe is a conventional “elliptical” lobe, the value of its preload \( m_2 \) is not unusual.

The evaluation of the initial design is completed by the assessment of the relevant performance indicators (§4.3, part 1), which values are reported in the final section of Table 2. They are used in Eq. (40) (part 1) as reference values \( Y_i^{ref} \) or, equivalently, the initial design is assumed as reference in order to evaluate new design solutions.

5. Parametric analyses

The influence of driving parameters and design variants is studied in order to improve the initial design that has been reasonably chosen in the previous paragraph. As an optimization based on the simultaneous variation of all of the driving variables would require a specialized numerical method (7), the problem is simplified and split into different parametric studies. Therefore, single or multiple parameters are varied in turn, leaving the others fixed at their initial values.

In order to simplify the problem, the relative pocket clearance is always kept constant \( (C_c = 0.0017) \). Indeed, the admissible design range for \( C_c \) is narrow (from 0.0017 to 0.0022) and such simplification does not limit the generality of results.

Nevertheless, by repeating the shape optimization for different values of \( C_c \) and by taking into account thermoelastic deformations \((8)\), a more significant optimum design might be also found.

A further simplification is obtained by assuming that top and bottom lobe does not influence each other. Indeed, such hypothesis is reasonable if two conditions are verified:

1. the top pad is completely cavitated,
2. the reformation boundary is locked by the upstream groove (inlet 1, inlet 1+2 and inlet 1&2 supply modes).

As far as condition 1 is concerned, if a pressurized region develops in the converging wedge of the top pad, interactions between the two pads occur through the journal equilibrium equations, as pressure distributions of both pads contribute to the hydrodynamic reaction that balances the external load.

With regard to condition 2, if the upstream groove is present, the pressurized film in the bottom pad is constrained to reform in the vicinity of such groove. Differently, if the upstream groove is not present, the reformation boundary can move and its location is determined by the upstream oil flow coming from the top pad. Hence, lubrication performances are determined by the interaction between the two pads.

In synthesis, the impact on performance of the three main driving variables that rule bearing profile is analyzed hereafter. Such variables are the pocket angle \( \alpha_t \), top pad ellipticity \( E_1 \), and bottom pad ellipticity \( E_2 \).

Fig. 1 compares different journal mobility plots (§4.4, part 1) obtained by varying the driving variables. Relative clearance \( C_p \) (Fig. 1 (a)) controls the area enclosed in the mobility contour but not its shape. Comparing Fig. 1 (b), (c) and (d) shows that \( E_1 \) modifies the radius of the pads and consequently \( C_p \), while it is not changed by parameters \( E_2 \) and \( \alpha_t \).

In the case of the following parametric analyses with constant \( C_c \), the relative pad clearance \( C_p \) is not directly chosen, but it is assessed on the basis of the main driving parameter \( E_1 \), which together with \( \alpha_t \) and \( E_2 \) determines the shape of the bearing profile.
Particularly, on one hand $\alpha_t$, $E_1$ control the pocket geometry as well as the bottom pad profile and on the other hand $E_2$ rules the top pad profile. Thanks to the above-mentioned assumption, the optimum profiles of the top and bottom pads can be searched by means of two independent parametric analyses by starting from the nominal bearing profile and by modifying one pad at a time.

![Fig. 1 Journal mobility plots for the initial design for different variable parameters: (a) relative pad clearance ($C_p = 2, 3, 4, 5, 6$), (b) bottom pad ellipticity ($E_1 = 2, 3, 4$), and (c) top pad ellipticity ($E_2 = 1, 2.06, 3.12$) and (d) pocket angle ($\alpha = 60, 90, 110$)](image)

Section 5.1 considers the effect of the pocket angle $\alpha_t$, which has not been studied hitherto in literature. On the contrary, the influence of clearance and ellipticity on fixed pad bearing performance is well-documented (1)-(2). Nevertheless, $E_1$ is varied together with $\alpha_t$, as the bottom lobe is not a regular elliptical bearing pad and the interaction between the two driving parameters characteristic of its profile is unknown.

Afterwards, the influence of top pad profile is considered in §5.2. As the effect of the top pad profile is strictly dependent of the feed groove layout, $E_2$ is varied together with the supply mode.

The effect of the barriers is finally studied (§5.3). As the aim of a barrier is only to decrease the side loss, its axial size $L_b$ is usually limited as much as possible (a large value of $L_b$ is highly detrimental for the load-carrying capacity). Hence, $L_b$ is not considered as a variable parameter and a design without barrier (initial design and $L_b = 0$) is simply compared with the nominal one.

5.1 Influence of pocket design

In the current paragraph, the initial design is modified by varying the pocket geometry (variable parameters: $\alpha_t$ and $E_1$, constant parameter: $E_2 = 3.12$).

A reasonable number (consistent with FEM analysis computational times) of discrete combinations of the driving dimensions have been considered (40 combinations, elapsed time: roughly 27 hours). For each combination, the geometrical design evaluation (Par. 4, part 1) is carried out. Intermediate performances among the chosen combinations are retrieved by means of cubic spline interpolation.

$E_1$ is varied in a range between 2 and 5. Greater values of $E_1$ are not included in the explored range. Indeed, if $E_1$ becomes greater than 5, $C_p$ rises over 8/1000, which is already over the advisable limit of 6/1000.
The performance indicators are plotted against $E_1$ and $\alpha_t$ in Figs. 2~7. A particularly important indicator is the minimum film thickness $h_{min}$ (Fig. 2), which is directly linked to the bearing load-carrying capacity. If the ratio between $h_{min}$ and the root mean square bearing surface roughness (averaged between journal and sleeve surfaces, according to ISO 12129-2:1995) is greater than 5, the lubrication is hydrodynamic \(^9\). The simulation model implemented in the version of FEMLub used in the present work holds in hydrodynamic regime, i.e. roughly for $h_{min} > 5 \cdot Ra = 25 \, \mu m$, where $Ra$ is the surface finish (Table 1). Lower and negative values of $h_{min}$ are predicted by FEM analyses for $\alpha_t<70$ deg and $\alpha_t<65$ deg, respectively, so that mixed and boundary lubrication regime may occur according to the adopted model. Hence, only the results for $\alpha_t>65$ deg are plotted in Fig. 2.

An optimum geometry that maximizes $h_{min}$, referred to as hydrodynamic optimum, is on the boundary of the design domain at the highest values of $\alpha_t$ and $E_1$ (110 deg and 5, respectively), as shown in Table 3, where the extrema of the performance indicators are reported.

Table 3 Extreme values of performance indicators and corresponding design variables (optima are bolded)

<table>
<thead>
<tr>
<th>Performance indicator</th>
<th>Maximum indicator</th>
<th>$\alpha_t$ [deg]</th>
<th>$E_1$ [deg]</th>
<th>Minimum indicator</th>
<th>$\alpha_t$ [deg]</th>
<th>$E_1$ [deg]</th>
</tr>
</thead>
<tbody>
<tr>
<td>$h_{min}$ [µm]</td>
<td>321.2</td>
<td>100</td>
<td>5</td>
<td>6.0</td>
<td>65</td>
<td>5</td>
</tr>
<tr>
<td>$p_{max}$ [MPa]</td>
<td>0.6</td>
<td>65</td>
<td>5</td>
<td>5.1</td>
<td>110</td>
<td>5</td>
</tr>
<tr>
<td>$T_{in}$ [°C]</td>
<td>101.2</td>
<td>110</td>
<td>2</td>
<td>88.7</td>
<td>65</td>
<td>5</td>
</tr>
<tr>
<td>$M_1$ [kN]</td>
<td>107</td>
<td>65</td>
<td>2</td>
<td>114</td>
<td>110</td>
<td>5</td>
</tr>
<tr>
<td>$Q$ [l/s]</td>
<td>6.5</td>
<td>65</td>
<td>5</td>
<td>2.8</td>
<td>110</td>
<td>2</td>
</tr>
<tr>
<td>$W$ [kN/m]</td>
<td>100</td>
<td>110</td>
<td>2</td>
<td>-29.1</td>
<td>94</td>
<td>5</td>
</tr>
<tr>
<td>$W_{c}$ [N/m]</td>
<td>$4.2 \times 10^5$</td>
<td>110</td>
<td>2</td>
<td>$-6.2 \times 10^5$</td>
<td>94</td>
<td>2</td>
</tr>
<tr>
<td>$f_1$ [N]</td>
<td>1.2</td>
<td>110</td>
<td>5</td>
<td>-49.7</td>
<td>65</td>
<td>5</td>
</tr>
<tr>
<td>$f_2$ [N]</td>
<td>18.7</td>
<td>110</td>
<td>2</td>
<td>-49.7</td>
<td>65</td>
<td>5</td>
</tr>
</tbody>
</table>

When $E_1$ rises, both the pad clearance $c_p$ and the bottom pad preload $m_1$, i.e. the wedge effect in the gap outside the pocket, increase (Fig. 1 (b)). As far as the rise of clearance is concerned, it reduces $h_{min}$, as the lubricant leaves the bearing more easily (the Sommerfeld number drops). On the contrary, in the matter of the growth of preload, whenever the active (pressurized) film region develops outside the pocket, the greater wedge effect increases $h_{min}$. Only this second effect is influenced by $\alpha_t$. Indeed, higher pocket angles yield wider active film regions \(^6\), while for the lower $\alpha_t$ the active film is confined to inner part of the pocket and the sharper wedge outside the pocket cannot affect the load-carrying capacity.

As a consequence, for lower values of $\alpha_t$ the minimum film thickness monotonously decreases by increasing $E_1$, while for higher $\alpha_t$ as a result of the opposite effect of $c_p$ and $m_1$ on film thickness, the trend of $h_{min}$ can be approximated by means of a convex function which minimum is located roughly at $E_1=4$.

Peak pressure $p_{max}$ is mainly linked to load-carrying capacity and to the extent of bottom pad active film region. Usually, the lower such capacity, the higher $p_{max}$. For the same capacity and resultant load $W$, which is obtained by integrating pressures, the wider the integration domain (the active film region) the lower $p_{max}$.

Peak pressure according to Fig. 3 grows as $E_1$ rises for all of the $\alpha_t$ values, except the highest, while it always drops with the increase in $\alpha_t$. The former behavior is an effect of the variation of the load-carrying capacity due to the $c_p$ and $m_1$ changes described above, the latter is also due to the growth of the active film region with the pocket size \(^6\).

The minimum value of $p_{max}$ is achieved at the hydrodynamic optimum (Table 3).

Fig. 4 shows that the supply flow rate $Q$ behaves like $p_{max}$, but $Q$ always grows as $E_1$ becomes large. Indeed, the increased wedge effect and clearance both contribute to rise the side flow. The reduction of the Poiseuille flow, due to the lower hydrodynamic pressure,
explains the drop of $Q$ as $\alpha_t$ is increased.

Heat dissipation $P$ (Fig. 5) is related to the efficiency of the bearing. For constant $E_1$ it decreases monotonically with $\alpha_t$, while for constant $\alpha_t$ its trend exhibits a minimum at intermediate values of $E_1$. Indeed, the variations of $P$ are consistent with those of $c_p$, $Q$, $T_{\text{max}}$, i.e. $P$ grows if clearance and temperature drop as well as flow increases. For the higher values of $\alpha_t$ the temperature drop and the flow increment due to the growth of $E_1$ are less remarkable so that $P$ monotonically decreases due to the rise of clearance.

![Fig. 2 Minimum film thickness $h_{\text{min}}$ as a function of bottom pad ellipticity $E_1$ and pocket opening angle $\alpha_t$](image1)

![Fig. 3 Peak pressure $p_{\text{max}}$ as a function of bottom pad ellipticity $E_1$ and pocket opening angle $\alpha_t$](image2)

![Fig. 4 Lubricant flow rate $Q$ as a function of bottom pad ellipticity $E_1$ and pocket opening angle $\alpha_t$](image3)

![Fig. 5 Heat dissipation $P$ as a function of bottom pad ellipticity $E_1$ and pocket opening angle $\alpha_t$](image4)

Table 3 indicates the friction optimum, i.e. the design with lowest $P$, which matches the hydrodynamic optimum.

The peak temperature $T_{\text{max}}$ (Fig. 6) is in agreement with the variations of flow and heat dissipation. Particularly, $T_{\text{max}}$ drops as $Q$ increases and $P$ decreases. The thermal optimum, i.e. the design with minimum $T_{\text{max}}$ is different from the hydrodynamic optimum (Table 3) in that it occurs at the minimum pocket angle, for which the large oil flow due to the high pressure gradient provides more efficient cooling.

The square critical parameter $\Omega^2$ (Fig. 7) is linked to the dynamic performance. When $\Omega^2$ is negative the bearing is inherently stable with reference to the simple linearized rotor model described in [3], so that a good dynamic behavior is anyway expected in a more complex system. This holds in a wide region of the explored domain $E_1\cdot\alpha_t$. Nevertheless, independently of $E_1$, between $\alpha_t=100$ deg and $\alpha_t=110$ deg, $\Omega^2$ varies abruptly and changes sign. As pad preload is the primary stabilizing device in a fixed pad bearing, in the $\alpha_t$ range where a stability boundary exists ($M_c>0$ or $\Omega^2>0$), it is higher for higher preloads in bearings operating with the same Sommerfeld number. Nevertheless, if on one hand the rise of $E_1$ (at constant $\alpha_t$) increases the preload, on the other hand it also increases the clearance $c_p$ and consequently reduces the Sommerfeld number so that, at least for low $E_1$, $M_c$ drops. In agreement with Eq. (39) of part 1, $\Omega^2$ further shrinks so that at $\alpha_t=110$ deg it decreases monotonically with $E_1$ (Fig. 7).

In the current (nominal) working conditions (constant journal mass and rotation speed, $m$ and $\omega$ respectively), for which $\Omega^2=m\omega^2=3.07\cdot10^9$ kg/s$^2$, the stability boundary is exceeded ($\Omega^2 > \Omega^2_c$) in the design domain region labeled as “Unstable” in Fig. 8, where
bearing dynamic status is plotted as a function of $\alpha_t$ and $E_1$. Unstable zone is very small anyway, as the nominal load is significant in relation with bearing dimensions, speed and oil actual viscosity, i.e. Sommerfeld number is quite low, in the order $10^{-2}$ to $10^{-3}$.

In synthesis, both global performance indicators $Y_g$ and $Y_g'$ in Table 3 find the global optimum for pocket design at $\alpha_t = 110$ deg. For this value, $Y_g'$ suggests that the unstable region is avoided and the increment of stability margin is sufficient.

By comparing $Y_g$ and $Y_g'$ data in Table 3, it is also evident that concern for the stability margin moves the optimal ellipticity from 5 to 2. Indeed the former design requires much higher clearance $c_p$ than the latter (the relative variation is about the 95% of the clearance relevant to the initial design). The higher the clearance, the lower the stability, as mentioned above.

A further increase of the most influential factor, i.e. the pocket angle $\alpha_t$, is not convenient. Indeed, it would decrease the stability at the expense of bearing safety, as shown by Fig. 9 where the global performance $Y_g'$ is plotted for pocket angles ranging from 65 to 150 deg. Although the global optimum implies a quite high working temperature, required to reduce viscosity and heat loss, the peak temperature (about 95°C to 100°C) is still acceptable. Indeed, the maximum admissible operative temperature of the white metal is 110°C and a calculated peak temperature 10°C lower is advisable in nominal conditions.

5.2 Influence of top pad design

A new parametric analysis allows exploring all of the intermediate bearing configurations between design A and B, i.e. top pad preload is varied, while the pocket and bottom pad geometry of the initial design are retained (variable parameter: $E_2$, constant parameters: $E_1 = 3.12$, $\alpha_t = 110$ deg). The choice of $E_2$ influences the oil circulation between the two pads, which depends of the supply mode, in turn. Consequently, the response to $E_2$ variation is determined by the particular supply mode, i.e. inlet 1, inlet 2, inlet (1)&2, inlet 1&2, inlet 1+2 ($\S$3.2, part 1).

Figures 10–14 plot against $E_2$ the main indicators and the global performance of the
bearing, \( P \) and \( T_{\text{max}} \), \( h_{\text{min}} \) and \( p_{\text{max}} \), \( Q \), \( Y_{g} \), \( Y_{g}^{s} \). Modes \( \text{inlet 1&2} \), \( \text{inlet 1+2} \) get the same performance indicators except for the oil supply rate \( Q \) (§4.2, part 1). Common results are labeled \( \text{inlet 1&/+2} \).

In the supply mode adopted in the initial design, which is referred to as \( \text{inlet 1} \) and is characterized by a single groove located upstream of the active film region, the expected variations in performance are very slight, as the incoming oil from the supply groove enters directly the active film region without passing through the top pad gap, which only carries the recirculated oil flow from the rupture boundary to the inlet groove. Accordingly, numerical results indicate that for \( \text{inlet 1} \) mode all of the performance indicators do not depend on top pad ellipticity \( E_{2} \), except for the power dissipation \( P \).

Another expectation is a similar behavior of modes \( \text{inlet 2} \) and \( \text{inlet (1)&2} \), as they share the same pressure-type boundary condition, i.e. lubricant feed is simulated in the same way. Again, numerical results confirm the expected behavior.

As depicted by Fig. 10, for all of the supply modes \( P \) shows a slightly increasing trend due to the growth of the friction linked to Couette flow in the top pad. Indeed, such friction contribution is inversely proportional to the film thickness, which decreases when the gap in the top pad is reduced by increasing \( E_{2} \). In any case, the variation of \( P \) in the explored range of \( E_{2} \), which maximum amount is 15 kW (10% of reference dissipation) for \( \text{inlet 1&/+2} \) mode, is not very significant and it slightly affects the temperature of the bearing and the remaining indicators.

Fig. 10 shows that the initial mode \( \text{inlet 1} \) causes the highest peak temperature, close to the maximum allowable operating temperature. By adopting supply modes that include inlet from the downstream groove (\( \text{inlet 2} \), \( \text{inlet (1)&2} \), \( \text{inlet 1&/+2} \)), the bearing runs colder and, unexpectedly, among such modes the higher peak temperature is predicted for the modes with two inlets (\( \text{inlet 1&/+2} \)). The reason is that the oil flow through the upstream groove, denoted by \( Q_{1} \) (§4.2, part 1), in modes \( \text{inlet 1&/+2} \) is always negative, i.e. the oil flows out the upstream groove. Such reverse flow, regardless of its recirculation, does not provide cooling of the bottom pad or, in other words, a portion of the additional oil that can be fed through the downstream groove does not enter the bottom lobe, as it flows out of the bearing through the upstream groove.

Fig. 11 shows that \( \text{inlet 2} \) and \( \text{inlet (1)&2} \) modes ensure the highest minimum film thickness \( h_{\text{min}} \). It is due both to the additional flow rate fed through the downstream groove, which yields a supplementary load-carrying capacity, and to the expansion of the active film region in the bottom pad, as described below.
Nevertheless, inlet 1&/+2 modes, which also include the downstream groove, provide roughly the same load-carrying capacity (or, equivalently, the same $h_{min}$) as inlet 1. Such behavior is mainly due to two causes. Firstly, reverse flow does not enter the active film region in the bottom pad, so that the above-cited advantage due to the additional flow rate is lost. In the second instance, whenever the downstream groove is fed (modes inlet 2, inlet (1)&2 and inlet 1&/+2), the top pad is not starved and in its converging wedge an active (pressurized) film region can develop. Pressure in the top pad is detrimental to the load-carrying capacity, as the resultant force includes a large component in the vertical direction with the same orientation as the external load. The first cause reduces the load-carrying capacity, the second increases the effective load carried by the bottom pad.

![Fig. 11 Minimum film thickness $h_{min}$ and peak pressure $p_{max}$ as a function of top pad ellipticity $E_2$ and supply mode](image)

Fig. 11 plots peak pressure $p_{max}$, which is lower when load-carrying capacity is higher and the extent of the bottom pad active film region is wider, as mentioned in §5.1.

Accordingly, due to both the above-described causes, $p_{max}$ for inlet 1&/+2 modes is always the highest and, particularly, it is higher than the one predicted for inlet 1 due to the detrimental secondary pressurized film region. Despite of such detriment, inlet 2 and inlet (1)&2 supply modes yield lower $p_{max}$ than inlet 1. Such behavior is mainly due to the exclusion of the upstream oil inlet, which moves the film reformation boundary. It is located roughly at $\vartheta = 60$ deg for inlet 2 as well as inlet (1)&2 and at $\vartheta = 90$ deg for inlet 1 as well as inlet 1&/+2. The upstream motion of reformation boundary causes the main active film region to enlarge and, consequently, the peak pressure to reduce perceivably.

Fig. 12 plots supply flow rates $Q$ and side loss $Q_{side}$. For all modes such two parameters are equal, within the convergence tolerance of mass-conserving algorithm, except for inlet 1&2 due to reverse flow (§4.2, part 1). By feeding the downstream groove, the supply flow rate must be much greater, up to a factor of 3 larger than flow for inlet 1 mode. Side loss is roughly the same for two-grooves modes (inlet 1&/+2) and inlet 1, i.e. the additional lubricant fed through the downstream groove does not enter the bottom pad and load-carrying capacity is similar in this two modes, as confirmed by the comparison of the corresponding $h_{min}$ (Fig. 11). The advantage of mode inlet 1+2 over inlet 1&2 in terms of supply oil saving is substantial.

Therefore, according to Fig. 13, the best global performance $Y_g$ regardless of stability is achieved by means of supply mode inlet 1+2 at $E_2=1$. Nevertheless, stability indicators for mode inlet 1+2 are lower than for inlet 1 and the best performance based on $Y_g^s$ (Fig. 14) is for mode inlet 2 at roughly $E_2=2.6$ due to a high local peak of the corresponding stability indicator.
The rise of a secondary active film region in the top pad is shown by Fig. 15, where the middle-plane pressure distributions for the four supply modes at the hand and for three values of the ellipticity $E_2$ (1, 1.51 and 3.12) are plotted.

![Fig. 12 Supply flow rate $Q$ and side loss $Q_{out}$ as a function of top pad ellipticity $E_2$ and supply mode](image)

![Fig. 13 Global performance $Y_g$ as a function of top pad ellipticity $E_2$ and supply mode](image)

![Fig. 14 Global and stability performance $Y_s$ as a function of top pad ellipticity $E_2$ and supply mode](image)

The top pad (secondary) hydrodynamic pressure develops for the higher values of $E_2$ and for supply modes with downstream inlet (Fig. 15 (b), (c) and (d)). Particularly, comparing Fig. 15 (a), (b) and (c) suggests that for inlet 2 and inlet (1)&2 modes at $E_2$=1.51, although the secondary active film region exists, the corresponding $p_{max}$ is lower than for inlet 1 due to the expansion of the active film region in the bottom pad. Modes inlet 2 and inlet (1)&2 (Fig. 15 (b) and (c), respectively) yield similar pressure trends, except for $\vartheta=90$ deg where the distribution is hydrostatic for the latter mode, in agreement with the presence of the inactive groove and the corresponding flow-type boundary condition.

Fig. 16 plots into the mobility areas the journal center locations corresponding to the pressure distributions in Fig. 15. All of the journal centers are inside the gray circles with radius $c_c$, which mark the mobility boundaries in the barrier zone. For the modes without upstream groove inlet (inlet 2 and inlet (1)&2, in Fig. 16 (b) and (c), respectively) at low values of $E_2$ the horizontal component of the journal center position $e_y$ is greater than in the remaining cases. Such displacement is due to the enlargement of the active film region in the bottom pad, which, in turn, increases the horizontal component of the hydrodynamic reaction. For higher values of $E_2$, due to the top pad pressure, the journal returns in the usual location, which is more centered and shifted to the inner part of the pocket (lower $e_y$, higher $e_x$). In such condition the journal is more blocked and stability is high.
5.3 Influence of barriers

The final parametric analysis compares the response of two design solutions, with and without the barriers. The former solution retains the barrier axial lengths of the initial design ($L_b=35 \text{ mm}$), while no barriers are simulated in the latter case ($L_b=0$). The same driving variables of the bearing profile are used and the most influential, the pocket angle, is varied in its design range (variable parameter: $\alpha_t$, constant parameters: $E_1=E_2=3.12$).

Fig. 17 shows minimum film thickness $h_{min}$ and supply flow rate $Q$ for both designs. Barriers reduce the bearing load-carrying capacity in all of the explored range and, due to the consequent reduction of film thickness, the design with barriers is not advisable for $\alpha_t < 80 \text{ deg}$. By serving the purpose for which they are designed, barriers allow the bearing to save roughly 2 l/min of oil for all of the pocket angles.

Fig. 18 plots the corresponding global performance $Y_g$. By including the stability indicator, performance does not change ($Y_g=Y_g\text{ss}$). For the bearing at the hand and by using the above-mentioned criteria (all of the weights $\gamma_i$ in Eq. (40) of part 1 are identical), the design with barriers is always favorable. Indeed, for the design with $\alpha_t=70 \text{ deg}$, which leads the bearing with barriers near the limit of mixed lubrication regime, barriers reduce $h_{min}$ by 55% and $Q$ by 83% of their corresponding reference (initial) values, 119 $\mu$m and 2.98 l/min respectively. Such example explains why the design with barriers still yields higher performance indicators for low pocket angles. A different choice of the weights, aimed to favor safety, would change such result, as the design without barriers is much safer at low
pocket angles.

Fig. 17 Minimum film thickness $h_{min}$ and supply flow rate $Q$ as a function of pocket opening angle $\alpha_t$ for design with and without barriers

6. Optimal design

By superimposing the results separately obtained for the top and bottom pads, two designs capable of improving the initial performance of the bearing according to $Y_g$ and $Y_{gs}$, respectively, are found.

Table 4 reports for both the designs driving variables, design variants, the expected global performance calculated by summing the corresponding performance indicators of the two parametric analyses ($\S\S 5.1$ and 5.2) and the actually computed global performance.

Design with barriers is always retained, as its superiority has been proved for the bearing data at the hand ($\S 5.3$).

Table 4 Driving variables, design variants and performance for the two best designs, with and without taking into account stability

<table>
<thead>
<tr>
<th>Barrier</th>
<th>Optimal design (performance indicator $Y_g$)</th>
<th>Optimal design (stability included, performance indicator $Y_{gs}$)</th>
</tr>
</thead>
<tbody>
<tr>
<td>$\alpha_t$ [deg]</td>
<td>110</td>
<td>110</td>
</tr>
<tr>
<td>$E_0$</td>
<td>5</td>
<td>2</td>
</tr>
<tr>
<td>$E_1$</td>
<td>1</td>
<td>3.12</td>
</tr>
<tr>
<td>Supply mode</td>
<td>Inlet 1+2</td>
<td>Inlet 1</td>
</tr>
<tr>
<td>Expected performance indicator</td>
<td>5.96</td>
<td>15.3</td>
</tr>
<tr>
<td>Calculated performance indicator</td>
<td>3.21</td>
<td>15.3</td>
</tr>
</tbody>
</table>

When stability is not taken into account, the best supply mode is inlet $1+2$, as shown by Fig. 13. In such case, the resulting global performance is lower than the sum of the performances of the two parts. Indeed, the two pads have a light mutual influence because the top pad is not completely cavitated (a low hydrodynamic pressure of few bars is generated between $\vartheta=270$ deg and $\vartheta=300$ deg) and, therefore, condition 1 of §5 is not completely met.

When stability is considered, the best supply mode is inlet 2, as shown by Fig. 14. Nevertheless, in such case both conditions 1 and 2 of §5 are not true and joining optimum top and bottom pads does not yield an optimum bearing. Hence, the second-best supply mode, inlet 1, is considered. Such case has been already calculated during the parametric analysis aimed to the bottom pad optimization.

Although for both the optimum designs the calculated performance improvement is not high, the initial design is already a near-optimal one, as it is adopted by an experienced manufacturer.

7. Conclusions

Although it is not possible to generalize the results of the application studies to all of the pocket bearing sizes and variants, an adequate number of design solutions have been
analyzed and some important conclusions can be drawn for medium-size pocket bearings.

As clearly shown by the global performance plot in Fig. 9, the response of numerical results agrees with the design practice of manufacturers, who usually adopt 110 deg pocket angle for bearings with barriers. If the second turning operation is extended to the whole bearing length, a wider range of the pocket angle can be employed and the bearing is safer, especially if low pocket angles are adopted, e.g. a 60 deg pocket is solely adequate to bearings without barriers.

In comparison with upstream groove, downstream supply grooves require much larger amount of supply oil but the bearing is safer and cooler. Anyway, lower temperatures cause higher friction and, therefore, upstream grooves are often employed for the sake of higher efficiency. Two-groove supply systems seem to strike balance between the two single-groove solutions, but oil recirculation is required as reverse flow may cause high oil consumption. The advantages of two-grooves design with lubricant recirculation (inlet 1+2 supply mode) might be overestimated, as the temperature rise of the recirculated flow has been neglected.

Numerical results have also proved the efficiency of barriers in reducing the side loss. On the contrary, the expedient of reducing the top pad clearance by increasing its ellipticity up to the maximum (design B solution) turns out to be effective only when a downstream groove is fed.

In the present study, due to the simplifying assumptions and the lack of a systematic search technique, only a moderate improvement of the bearing design has been carried out. Other work is required to perform a more effective optimization of the bearing by means of a suitable numerical method.

Acknowledgments

I would like to acknowledge Dr. Alessandro Reppa and his Department of Ansaldo Energia (Genoa) for providing information on the bearing in study.

References