Assessment of Porous Type Gas Bearings: Measurements of Bearing Performance and Rotor Vibrations

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Gas film bearings are effective means to support light weight rotating equipment w/o complicated lubrication systems.

Systems/products under development/use include air cycle machines, PV turbochargers, compact HS/HP compressors, water aeration pumps, etc.

Need to characterize most favourable gas bearing technology (cost, performance, etc).
Gas bearings with compliant surfaces

- Bump-type foil bearing
- Overleaf bearing
- Metal mesh foil bearing
Porous type gas bearings (PTGB) have sub-micron sized holes distributed in their matrix material, allowing for an even distribution of gas flow on the lubricated surface.

When compared to orifice restricted hydrostatic gas bearings, PTGBs have higher stiffness & damping.

Commercial applications: medical imaging tomography, high-speed spindle, coordinate measuring machine.

http://www.newwayairbearings.com/catalog/components
Other types of gas bearings

- Flexure-pivot tilting pad bearings (hydrodynamic & hybrid)

- Carbon graphite porous surface bearings

Demonstration of bearing porosity
Literature on Porous Gas Bearings

Montgomery, et al., 1955

Castelli, V. P., 1979

San Andrés et al., 2015, STLE Annual Meeting, Dallas, TX.
Three-pad tilting pad Carbon-Graphite PGBs

Tilting pad bearings with external pressurization allow operation with minute gaps, offer high stiffness and little friction, and provide enhanced stability due to nil cross-coupled stiffnesses.
2015: I test rig for high speed PGBs

- **Three-Pad PGBs**
  - Axial Length, $L_b$: 30 mm
  - Width, $w$: 30 mm
  - Radius, $R_b$: 14.3 mm
  - Arc Length, $\theta_p$: 82.5°
  - Pivot Offset: 50%

- **Rotor**
  - Rotor mass, $M$: 0.89 kg
  - Rotor diameter, $D$: 28.5 mm
  - Rotor length, $L$: 190 mm
  - Polar moment of inertia, $I_p$: 0.91 kg·cm²
2015 evaluation of rotor friction coefficient

COASTDOWN TESTS
Surface speed: $R_r \cdot \Omega_{\text{max}} = 82 \text{ m/s}$
$t > 3 \text{ min} \text{ to come to rest}$

Increase in supply pressure $P_s$ results in a longer duration of rotor coast down. For low $P_s$ and $\Omega/\Omega_0 < 0.1$ – rotor rubs and decelerates fast.

<table>
<thead>
<tr>
<th>Supply pressure [bar]</th>
<th>Bearing drag coefficient, $C_\theta$ [10^{-3} \text{ N\cdot m\cdot s/ rad]}</th>
<th>Friction factor, $f$</th>
</tr>
</thead>
<tbody>
<tr>
<td>5.1</td>
<td>0.43</td>
<td>0.007</td>
</tr>
<tr>
<td>6.5</td>
<td>0.30</td>
<td>0.005</td>
</tr>
<tr>
<td>7.9</td>
<td>0.25</td>
<td>0.004</td>
</tr>
</tbody>
</table>

As $P_s$ increases, the drag $C_\theta$ decreases and friction factor $f$ decreases from 0.007 to 0.004.
2015 response in small test rig

The rotor response is mainly synchronous (1X). Incipient SSV motions appear/die above critical speed. Rotor-bearing system is stable at highest speed (55 krpm).
Current work and objective

- **Air flow measurements**
  Estimate the bearing pads permeability coefficient ($\kappa$).

- **Rotor speed coast down tests**
  Estimate bearing drag torque ($T_{\text{bearing}}$) & drag coefficient ($C_\theta$).
  Extract friction factor ($f$) & equivalent film thickness ($h$).

- **Imbalance response tests**
  Quantify rotor response due to mass imbalances.

- **Static load tests**
  Calibrate a hydrostatic load mechanism and estimate bearing direct stiffness ($K_{yy}$).
### Solid Steel Rotor

<table>
<thead>
<tr>
<th>Metric</th>
<th>SI Units</th>
<th>English Units</th>
</tr>
</thead>
<tbody>
<tr>
<td>Mass, $M$</td>
<td>29.1 kg</td>
<td>64.2 lbs</td>
</tr>
<tr>
<td>Weight, $W_r$</td>
<td>285 N</td>
<td>64.2 lbf</td>
</tr>
<tr>
<td>Specific Load, $\frac{1}{2}W_r/(L_b D_r)$</td>
<td>19.5 kPa</td>
<td>2.8 psi</td>
</tr>
<tr>
<td>Diameter, $D_r$</td>
<td>100 mm</td>
<td>3.9 inch</td>
</tr>
<tr>
<td>Rotor length, $L_r$</td>
<td>457 mm</td>
<td>18 inch</td>
</tr>
</tbody>
</table>

### PGBs test rig

- **Test rotor**
- **Porous bearings**
- **Polymeric coupling**
- **Motor**
- **VFD**
- **Air feeding NDE**
- **Air feeding DE**
- **Air feeding main line**
- **Tachometer**

### 3-Phase AC Motor

- **Power** 7.46 kW
- **Max. Speed** 18 krpm
Five-Pad porous type tilting-pad bearing

Shaft rotation

Pressurized air supply into pad

Bearing cartridge

Porous Pads

Section A-A

Rotor

Load

Load cartridge

Pad cartridge

Porous pad

Thermocouples

Five-Pad PGBs

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Axial Length, $L_b$</td>
<td>73 mm</td>
</tr>
<tr>
<td>Width, $w$</td>
<td>51 mm</td>
</tr>
<tr>
<td>Radius, $R_b$</td>
<td>50 mm</td>
</tr>
<tr>
<td>Arc Length, $\theta_p$</td>
<td>60°</td>
</tr>
<tr>
<td>Pad Height, $H$</td>
<td>26 mm</td>
</tr>
<tr>
<td>Pivot Offset</td>
<td>50%</td>
</tr>
<tr>
<td>Pad preload</td>
<td>0</td>
</tr>
</tbody>
</table>
Isometric, top and side views of PGB

**Specific Load**, \( \frac{1}{2} \frac{W_r}{L_b D_r} \)

<table>
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<tr>
<th>SI Units</th>
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<tbody>
<tr>
<td>19.5 kPa</td>
<td>2.8 psi</td>
</tr>
</tbody>
</table>

- **Weight**, \( W_r \): 285 N (64.2 lbf)
- **Diameter**, \( D_r \): 100 mm (3.9 inch)
- **Bearing length**, \( L_b \): 73 mm (2.9 inch)

**Low static load**

- \( H_{c_{\text{min}}} = 2.2 \text{ mm} \)
- \( H_{c_{\text{max}}} = 9.1 \text{ mm} \)
Air flow rate through PGBs

Objective: **Estimate the bearing pads permeability coefficient** ($\kappa$). **Air temperature:** $T_s = 21 \, ^\circ\text{C}$

<table>
<thead>
<tr>
<th>$P_s$</th>
<th>Supply pressure</th>
<th>1.0~7.77 bar</th>
</tr>
</thead>
<tbody>
<tr>
<td>$P_a$</td>
<td>Ambient pressure</td>
<td>1.0 bar</td>
</tr>
<tr>
<td>$p_r$</td>
<td>Pressure ratio, $P_s/P_a$</td>
<td></td>
</tr>
<tr>
<td>$\kappa$</td>
<td>Permeability coefficient</td>
<td>$m^2$</td>
</tr>
</tbody>
</table>
Air flow vs supply pressure to bearings

\[
\frac{P_s}{P_a} = 1.0 \rightarrow 4.4: \text{ Flow rate grows non-linearly}
\]

\[
\frac{P_s}{P_a} > 4.4: \text{ Linear growth}
\]

Without rotor

With rotor, similar trend. Rotor lifts at \( \frac{P_s}{P_a} = 4.4 \). Slightly lower flow rates (7\%~15\% less)

Note: expiratory flow rate of average adult human (~450 LPM).

Note specific pressure (load) on bearings is just 0.195 bar (2.8 psi).
**Bulk-Permeability (κ) of a porous pad**

\[
\kappa = \frac{\mu t}{A_{pad} \frac{G \mathcal{R} T}{N_{pad} A_{pad} \frac{G \mathcal{R} T}{2 (P_s^2 - P_a^2)}}}
\]

<table>
<thead>
<tr>
<th>(A_{pad})</th>
<th>Area of a single pad surface [m²]</th>
</tr>
</thead>
<tbody>
<tr>
<td>(G)</td>
<td>Air flow rate [kg/s]</td>
</tr>
<tr>
<td>(t)</td>
<td>Thickness of porous layer [mm]</td>
</tr>
<tr>
<td>(\mu)</td>
<td>Air viscosity, 19.8 ×10⁻⁶ [Pa⋅s]</td>
</tr>
<tr>
<td>(\mathcal{R})</td>
<td>Gas constant, 286.7 [J/kg⋅K]</td>
</tr>
</tbody>
</table>

**Bulk Permeability Coefficient, \(\kappa \text{ [m}^2\text{]}\)**

- Drive-end
- Non-drive end

**Pressure Drop, \(P_s - P_a \text{ [bar]}\)**

\(A_{pad} \sim K_{DE} \sim K_{NDE} = 1.2 \times 10^{-15} \text{ m}^2\)

\(\kappa\) on the low end of reported in literature \(\{10^{-16} \sim 10^{-10} \text{ m}^2\}\).

\(\kappa\) with rotor < \(\kappa\) without rotor

By 7% ~ 17% (DE), by 7% ~ 19%, (NDE):
Air gap is not an effective flow restriction.
**Objective:**
Estimate bearing drag torque ($T_{bearing}$) & drag coefficient ($C_\theta$) → friction factor ($f$) & equivalent film thickness ($h$).

**Air temperature:** $T_s = 21$ °C

<table>
<thead>
<tr>
<th>$\Omega$</th>
<th>Rotor angular speed [rad/s]</th>
</tr>
</thead>
<tbody>
<tr>
<td>$\Omega_0$</td>
<td>Max. rotor angular speed, 838 [rad/s]</td>
</tr>
<tr>
<td>$P_s$</td>
<td>Supply pressure, 5.84~7.77 [bar]</td>
</tr>
</tbody>
</table>
COASTDOWN TESTS
Surface speed: $R_r \cdot \Omega_0 = 42 \text{ m/s}$
$t > 2 \text{ min to come to rest}$

Increase in supply pressure $P_s$ results in a longer duration of rotor coast down. For $\Omega/\Omega_0 < 0.1$ – rubs and rotor decelerates fast.
Bearing drag and friction factor

From rotor speed ($\Omega$) deceleration test:

\[
I_p \frac{\partial \Omega}{\partial t} + 2T_{bearing} + T_{drag\,motor} = 0
\]

Obtain:

Bearing drag coefficient $C_\theta$

Friction factor

\[
f = \frac{T_{bearing}}{\left( \frac{1}{2} W_R R_B \right)}
\]

Equiv. film thickness

\[
h = \frac{N_{pad} \mu L_b R_b^3 \theta_p}{C_\theta}
\]

Rotor inertia $I_p$: 0.038 kg⋅m²

Bearing drag torque $T_{bearing} = C_\theta \Omega$

$T_{drag\,motor}$: 0.028 N⋅m (dry friction)

As $P_s$ increases, the drag coefficient ($C_\theta$) decreases; friction factor $f = 0.033 \rightarrow 0.019$
Imbalance response tests

- Quantify the dynamic response of rotor-PGB system:
  - Influence of supply pressure $P_s$ on system 1X response
  - Influence of added imbalance ($m$) on system response
Free-free modes of rotor

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Measured</th>
<th>Predicted</th>
<th>Difference</th>
</tr>
</thead>
<tbody>
<tr>
<td>Mass [kg]</td>
<td>29.12</td>
<td>29.18</td>
<td>0.2 %</td>
</tr>
<tr>
<td>Center of gravity from DE [cm]</td>
<td>23.47</td>
<td>23.45</td>
<td>0.1 %</td>
</tr>
<tr>
<td>$I_p$ [kg.m$^2$]</td>
<td>0.0359</td>
<td>0.0373</td>
<td>3.9 %</td>
</tr>
<tr>
<td>$I_t$ [kg.m$^2$]</td>
<td>0.5565</td>
<td>0.5544</td>
<td>0.4 %</td>
</tr>
<tr>
<td>First natural frequency [Hz]</td>
<td>1,888</td>
<td>1,881</td>
<td>0.4 %</td>
</tr>
<tr>
<td>Second natural frequency [Hz]</td>
<td>4,488</td>
<td>4,501</td>
<td>0.3 %</td>
</tr>
</tbody>
</table>

First and second natural frequencies are well above rotor max. speed (18 krpm, ~ 300 Hz). **Rotor is rigid.**
Added mass imbalance

ISO 1940-1, $u_{G=2.5} = 2.6 \text{ g·mm/kg}$ for rigid rotor operating at 10 krpm.

<table>
<thead>
<tr>
<th>Added imbalance</th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>$m$</td>
<td>$u = m \cdot e / M$</td>
</tr>
<tr>
<td>$m_1 = 3.45 \text{ g}$</td>
<td>$u_1 = 6.70 \text{ g·mm/kg}$</td>
</tr>
<tr>
<td>$m_2 = 4.20 \text{ g}$</td>
<td>$u_2 = 8.10 \text{ “}$</td>
</tr>
<tr>
<td>$m_3 = 6.90 \text{ g}$</td>
<td>$u_3 = 13.30 \text{ “}$</td>
</tr>
</tbody>
</table>

Air supply pressure: $P_s = 3.77, 5.84, 6.53, 7.22, 7.77 \text{ bar}$

$T_s = 21 \degree \text{C}$ Temperature
Waterfall plots

Super synchronous frequency motions: Low magnitude

Sub-synchronous motions: None

NDEH: Non-drive end, horizontal  NDEV: vertical

Frequency [Hz]

Amplitude: 2 μm pp/div

Time [s]

$P_e = 7.77$ bar, NDE

$P_e = 7.77$ bar, NDEV

$10 \text{ krpm}$
1X response amplitude and phase

1X amplitude increases steadily until reaching critical speed at ~ 9 krpm (rigid body conical mode)
Influence of supply pressure ($P_s$)

As $P_s$ increases:
Slightly higher critical speed and
little influence on 1X peak amplitude
1X response for various imbalances

Response at $P_s = 7.77$ bar

- $P_s = 7.77$ bar, DEH
- 3.45 g
- 4.20 g
- 6.90 g

Normalized responses

- $P_s = 7.77$ bar
- $m_1 = 3.45$ g
- $m_2 = 4.20$ g
- $m_3 = 6.90$ g

Response at $P_s = 3.77$ bar

- $P_s = 3.77$ bar, DEH
- 3.45 g
- 4.20 g
- 6.90 g

At same rotor speed:
- Higher $m$ $\rightarrow$ larger 1X amplitude.
- Higher $P_s$ $\rightarrow$ lower 1X amplitude

Linear rotor-bearing system.
Static tests with a hydrostatic load mechanism

- Quantify static stiffness of bearings

\[ F_s \]
Static load tests

<table>
<thead>
<tr>
<th>Static load supported by DEB</th>
<th>Static displacement at DEB</th>
</tr>
</thead>
<tbody>
<tr>
<td>$F_s$</td>
<td>$F_{DE}$ [N]</td>
</tr>
<tr>
<td>23</td>
<td>13</td>
</tr>
<tr>
<td>32</td>
<td>16</td>
</tr>
<tr>
<td>42</td>
<td>20</td>
</tr>
<tr>
<td>51</td>
<td>24</td>
</tr>
<tr>
<td>60</td>
<td>27</td>
</tr>
</tbody>
</table>

Increasing $P_s$ gives a modest increase in PGB static stiffness.
Rotor seizure & pads restoration
Seizure: observations and restoring

- **Test with**
  - Highest imbalance \( m_3 = 6.90 \text{ g} \).
  - Lowest pressure \( P_s = 5.15 \text{ bar} \).
  - When operating at \(~8.5 \text{ krpm}\), (bottom) pad temperature increased \(~15 \degree\text{C} \(< 3 \text{ s}\)).
  - Motor self-protection mechanism shut down power.
  - Rotor expanded and seized at DE bearing \(\rightarrow\) quick stop \((< 5 \text{ s})\).
  - Flexible coupling & limited torque maintained integrity of the rotor & bearings.
  - Visible wear on DE bearing.

- **Restoring process**
  - A simple smoothing process restored the PGBs to a usable condition.
  - Max. rotor speed = 8 krpm for following tests. A precaution
1X response post seizure

Prior to seizure

\[ m_3 = 6.90 \text{ g}, \, P_s = 7.77 \text{ bar} \]

Post seizure

\[ m_3 = 6.90 \text{ g}, \, P_s = 7.77 \text{ bar} \]

Similar rotor response amplitude at the NDE bearing before and after seizure. Rotor motion at the DE bearing a little lower.
Conclusion

From test data for a solid steel rotor supported on tilting pad porous gas bearings (specific load =0.195 bar).

(a) Porous pads have low permeability $\rightarrow$ little flow rate.
(b) Min. pressure to lift test rotor = 4.4 bar $\gg$ specific load.
(c) Rotor speed deceleration driven mainly by viscous drag.
(d) Friction factor $(f)$ $\sim$0.030 at $P_s = 5.84$ bar; $\sim$0.019 at $P_s = 7.77$ bar $\rightarrow$ nearly friction-free operation. Estimated film thickness $(h)$ increases as $P_s$ increases.
(e) The amplitude of rotor synchronous response increases steadily with rotor speed $(\Omega)$ as it approaches the 1$^{st}$ critical speed (conical mode).
(f) Linear rotor-bearing system: amplitude of rotor response is proportional to imbalance.
(g) Seizure event delayed test results and prevented operation $> 8$ krpm.
Acknowledgments
TAMU Turbomachinery Research Consortium & New Way Air Bearings

Questions (?)

Learn more at http://rotorlab.tamu.edu