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This material is based upon work supported by the TAMU Turbomachinery Research Consortium (TRC) and NASA GRC.
To develop a detailed, physics-based computational model of gas-lubricated foil journal bearings including thermal effects to predict bearing performance.

In support of **oil-free systems**

- reduce overall system weight, complexity (low footprint)
- increase system efficiency due to low drag power losses
- extend maintenance intervals.

Since 2003, supported by NSF, NASA, Capstone and TRC
Gas foil bearings (+/-)

- Increase reliability & good load capacity (< 20 psi)
- Dispense with oil lubrication
- Reduced weight & number of parts
- High and low temperatures
- Tolerate high vibration and shock load.

- Load capacity: less than rolling or oil lubricated bearings
- Endurance: wear during start up & shut down (lift off speed)

- Thermal management for high temperature applications (gas turbines, turbochargers)
- Predictive models lack validation for GFB operation at HIGH TEMPERATURE
<table>
<thead>
<tr>
<th>Year</th>
<th>Topic</th>
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<tbody>
<tr>
<td>2008-13</td>
<td><strong>Metal Mesh Foil Bearings</strong>: construction, verification of lift off performance and load capacity, identification of structural stiffness and damping coefficients, identification of rotordynamic force coefficients</td>
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<tr>
<td>2008-10</td>
<td><strong>Performance at high temperatures</strong>, temperature and rotordynamic measurements. Extend nonlinear rotordynamic analysis</td>
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<tr>
<td>2007-09</td>
<td><strong>Thermoelastohydrodynamic model for prediction of GFB static and dynamic forced performance at high temperatures</strong></td>
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<td>2005-07</td>
<td>Rotordynamic measurements: instability vs. forced nonlinearity?</td>
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<tr>
<td>2005-06</td>
<td>Model for ultimate load capacity, Isothermal model for prediction of GFB static and dynamic forced performance</td>
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<tr>
<td>2004-09</td>
<td>Measurement of static load capacity, Identification of structural stiffness and damping coefficients. Ambient and high temperatures</td>
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Temperature and rotordynamic measurements of a heated rotor supported on gas foil bearings without cooling flow

- Measure temperature of bearings and rotor and the motions of rotor for increasing shaft temperatures
- Identify post-test condition of test rotor and bearings after sudden failure
- Estimate bearing clearance and top foil temperature to until bearing seizure occurs
- Compare the experimental results to predictions from an in-house computational program
Gas bearings (when airborne) are nearly friction free, with small drag power and temperature rise.

However, the material structure of a foil bearing tends to heat up quickly since the lubricant, a gas, has low density and low thermal conductivity.

Rises in temperature change material properties (solids and gas) and the bearing clearance.....

Applications demand large cooling flows to control thermal growth of components and to remove efficiently mechanical energy from the rotor mainly.
Cooling effectiveness

-Even a little flow can cool the bearings & rotor. Too large cooling flows stop being effective.

-The cooling effect of the supplied flows is most distinct when the rotor is hottest and at the highest rotor speed.
The current experimental results show poor operating procedure and ignorance on the predicted system behavior.

Inadequate thermal management caused a sudden system failure (bearing seizure) while operating at a rotor speed of 37 krpm with the shaft OD temperature at ~250 °C.

The following details the test procedure, temperature measurements, and discusses the possible causes leading to the failure.
The test rig
Hot rotor-FB test rig

Instrumentation for high temperature.

Drive motor (max. 50 krpm)

- Cartridge heater
- Fiberoptic displacement sensors
- Infrared thermometers
- Cooling air supply for coupling
- Infrared tachometer
- Flexible coupling
- Drive motor
- Test GFBs
- GFB support housing
- Hollow rotor

Significant temperature gradient along rotor axis
Heat source warms (unevenly) rotor and its bearings
Hollow rotor (AISI 4140): 1.31 kg
Length: 200.7 mm, OD 36.46 mm and ID 17.90 mm.
NO coating on rotor surface
Hot rotor-FB test rig

Instrumentation

Thermocouples (K-type)

- Heater temperature controller
- Infrared thermometer
- Cartridge heater
- Foil bearings
- Drive motor
- Infrared thermometers
- Motor controller
- Displacement sensors
- Temperature digital panel meter
- Tachometer
- Signal conditioner
- Oscilloscopes
- FFT Analyzer
- Data acquisition board
- PC

Thermocouples:
- 1 x heater, 1 x Bearing housing enclosure
- 2 x 4 FB outboard, 2 x Bearing housing
- 2x Drive motor, 1 x ambient
- + infrared thermometers 2 x rotor surface

(Total = 17)
The test bearings
Generation II FB

Three (axial) bump strip layers, each with 24 bumps.

Patented solid lubricant coating (up to 800°F) on top foil surface.

<table>
<thead>
<tr>
<th>Parameter [Dimension]</th>
<th>Drive end</th>
<th>Free end</th>
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<tbody>
<tr>
<td>Cartridge inner diameter [mm]</td>
<td>37.98</td>
<td>37.92</td>
</tr>
<tr>
<td>Cartridge outer diameter [mm]</td>
<td>44.64</td>
<td>44.58</td>
</tr>
<tr>
<td>Axial bearing length [mm]</td>
<td>25.40</td>
<td>25.40</td>
</tr>
<tr>
<td>Number of bumps</td>
<td>24 × 3</td>
<td>24 × 3</td>
</tr>
<tr>
<td>Bearing radial (assembly) clearance [mm]</td>
<td>0.076</td>
<td>0.042</td>
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</table>

Other data proprietary
Four (4) thermocouples placed in machined axial slots.
Test Procedure
Test procedure

- Rotor spins at 37 krpm

- Th is set at 200°C, 400°C, 600°C (~60 minute intervals)

- Failure at ~190 minute of elapsed test time
Temperature measurements

Diagram showing temperature measurements:
- T1~T4
- Te
- T6~T9
- Th
- Enclosure
- Bearing housing
- Tr_{FE}
- Tr_{DE}
Rotor OD temperature

Significant axial thermal gradient from the rotor free end towards its drive end

$T_{FE} >> T_{DE}$
FB sleeve OD temperatures are different depending on their circumferential location; a result related to the heater warming the shaft and bearings unevenly, even without rotor spinning.
Steady state temperatures

- Steady state temperature (~60 minute heating)

Transient temperature (~10 minute heating)

* Steady state temperature (~60 minute heating)

Temperature [°C]

Heater set temperature, \( \text{Th} \) [°C]

1 2 3 4

- \( \text{Th} = 200^\circ \text{C} \)
- \( \text{Th} = 400^\circ \text{C} \)
- \( \text{Th} = 600^\circ \text{C} \)

- Large thermal gradient in the rotor

- \( T_e \) and \( T_{\text{DE}} \) rise above the temperatures in the bearing sleeves (\( T_4, T_9 \)) as the heater temperature increases

None of the temperature measurements gives any evidence of an impending failure!
Rotor motion measurements
Waterfall of rotor response

37 krpm
No cooling flow into bearings

Rotor motion components at its free end show no distinctive differences as the rotor temperature increases.
Synchronous rotor response

37 krpm

Rotor temperature does not affect the size and orientation of the FE rotor orbits.
Post-test conditions
HT on FE rotor OD renders a much darker color!

Rotor surface where the FE bearing is held shows considerable wear and noteworthy scratches.

NO protective coatings on Rotor surface

Before operation

AFTER incident

Smears of top foil protective coating

Drive end bearing location

Free end bearing location
Drive end foil bearing after incident

Inboard

Wear marks

Outboard

Minor wear marks on the inboard top foil

Bearing still functional!
Free end foil bearing after incident

Melted top foil

Galled coating

Ripple traces

Beyond repair!

Melted top foil

Inboard

Outboard

Outboard

Inboard

Melting temperature of the foil material is 1,430°C!

Protective coating evaporated, and distinct sections of the top foil melted, in particular at the location where the top foil contacts the bump foil crests
Close up view of FE bearing

Most metal melted right at the line contacts with the crest of the bump foils → Bearing seizure!

Large amount of energy generated, producing an increasing temperature, first melted the protective coating (max. 400°C) and later the top foil.
Thermal expansion of test rig components
Predicted thermal expansion

Thermal expansion of the rotor OD is more pronounced since it is hotter than the bearing sleeve.
FE bearing eventually loses its clearance, followed by sudden bearing seizure and system failure!
Damage on inboard side of bearing?

Inboard side of the bearing sleeve shrunk more than outboard side or at the mid-span.

Trapped air in the enclosure became a thermal sink causing FE bearing inboard temperature to be higher than that on outboard side of the bearing!
Temperature predictions versus test data
<table>
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<tr>
<th>TAMU GFB TEHD model</th>
<th>Reynolds eqn. for hydrodynamic pressure generation</th>
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<td>Energy transport eqn. for mean flow temperature</td>
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<td>Various surface heat convection models</td>
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<td></td>
<td>Mixing of temperature at leading edge of top foil</td>
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<tr>
<td>Gas film</td>
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<th>Top foil &amp; underspring</th>
<th>Thermo-elastic deformation eqns.</th>
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<td>Finite Elements and discrete parameter for bump strips.</td>
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<td>Thermal energy conduction paths to side cooling flow and bearing housing.</td>
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<tr>
<th>Bearing clearance</th>
<th>Material properties (gas &amp; foils) = f (Temperature)</th>
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<td>Shaft thermal and centrifugal growth</td>
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<td>Bearing thermal growth</td>
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</table>

San Andrés and Kim (2008)

Software licensed by TAMU
Recap: test rotor and FB

- Outer flow stream
- Top foil
- Bearing housing
- "Bump" layer
- Thin film flow
- PCo, TCo
- Cooling stream
- Pa
- Hot air (out)
- Natural convection on exposed surfaces of bearing OD and shaft ID
- View of rotor, heater cartridge + side cooling stream
Bearing temperature predictions & tests

Thermal mixing coefficient $\lambda=0.65$

37 krpm

Predictions (DE bearing) agree well with test data!!

Discrepancies are due to large temperature gradient along heater axial length
Estimation of the temperature raise in the top foil
Dry friction contact generated large amounts of energy that could not be convected quickly to the bearing sleeve or removed by the gas film since there was no forced cooling flow. Hence, the temperature increased very rapidly.

Estimation of top foil temperature

\[
\left( C_p M \right)_{Foil} \frac{dT_{Foil}}{dt} + Ah_\infty (T_{Foil} - T_\infty) = \varphi_{friction}
\]

- Foil specific heat and mass
- Contact area
- Top foil temp. increase
- Generated Mechanical power due to rubbing

\[
\varphi_{friction} = \mu W (R_{SO} \Omega)
\]

\[
T_{Foil}(t) = T_o e^{-t/\tau} + \left[ T_\infty + \frac{P_{friction}}{Ah_{air}} \right] \left( 1 - e^{-t/\tau} \right)
\]

Solution is valid for \(T_{Foil} < 1,430 \, ^\circ C \) \( (T_{melt}) \)

\[
\tau = \frac{(C_p M)_{Foil}}{Ah_\infty}
\]

130 W (with 6 N and 37 krpm)
Rapid growth of the foil temperature to reach its melting point in ~14 s.
Conclusions

✓ Unusual operating condition, w/o cooling flow and too large increment in rotor temperature (up to 250°C) led to the incident which destroyed one of the test bearings.
✓ Bearing clearance decreases notably as rotor temperature increases until seizure occurs.
✓ Upon contact between the rotor and top foil, dry-friction quickly generated vast amounts of energy that melted the protective coating and metal top foil.
✓ In spite of the loss of one foil bearing, the other system components (rotor, 2nd bearing and instrumentation) remain functional, showing little damage.

Predictive tool validated & benchmarked to reliable test data base.
✓ Even small quantities of air could be effective to promote the evacuation of hot air. The qualitative assessment for GFB system thermal management requires considerable experience!
✓ A cautious predictive design and conservative operation of the GFB system, along with an adequate thermal control strategy, could have prevented the bearing seizure and the rotor bearing system failure.

Both (a) engineering thermal management with adequate cooling scheme and (b) adequate assessment of thermal effects prior to actual operation are mandatory for high temperature operation.
Acknowledgments

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• TRC (2004-10)

Learn more http://rotorlab.tamu.edu