EXPERIMENTAL INVESTIGATION RESULTS OF A HYBRID CERAMIC AND ACTIVELY COOLED BALL BEARING FOR GAS TURBINES

13 October 2016
Brussels, Belgium

Dr.-Ing. P. Glöckner
FAG Aerospace GmbH & Co.KG
HYBRID CERAMIC AND ACTIVELY COOLED BALL BEARING FOR GAS TURBINES

<p>| | |</p>
<table>
<thead>
<tr>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Motivation &amp; Goal</td>
</tr>
<tr>
<td>2</td>
<td>Test Bearing and Rig Test Head Design, Test Conditions</td>
</tr>
<tr>
<td>3</td>
<td>Experimental Investigation Results: Oil Flow</td>
</tr>
<tr>
<td>4</td>
<td>Experimental Investigation Results: Temperatures</td>
</tr>
<tr>
<td>5</td>
<td>Experimental Investigation Results: Bearing Power Loss &amp; Heat to Oil (HTO)</td>
</tr>
<tr>
<td>6</td>
<td>High Speed Capability</td>
</tr>
<tr>
<td>7</td>
<td>Conclusion &amp; Outlook</td>
</tr>
</tbody>
</table>
## Motivation & Goal

<p>| | |</p>
<table>
<thead>
<tr>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Motivation &amp; Goal</td>
</tr>
<tr>
<td>2</td>
<td>Test Bearing and Rig Test Head Design, Test Conditions</td>
</tr>
<tr>
<td>3</td>
<td>Experimental Investigation Results: Oil Flow</td>
</tr>
<tr>
<td>4</td>
<td>Experimental Investigation Results: Temperatures</td>
</tr>
<tr>
<td>5</td>
<td>Experimental Investigation Results: Bearing Power Loss &amp; Heat to Oil (HTO)</td>
</tr>
<tr>
<td>6</td>
<td>High Speed Capability</td>
</tr>
<tr>
<td>7</td>
<td>Conclusion &amp; Outlook</td>
</tr>
</tbody>
</table>
Motivation

Ball Bearing Friction Power

Goal: Reduction of bearing power loss and thermal stressing.
Motivation

For under-race lubricated bearings the outer ring is the temperature-critical component.
Motivation

Influence of Under - Race Oil Flow on Bearing Power Loss

\[ F_{\text{Thrust}} = 80 \text{ kN}, \ T_{\text{Oil in}} = 80 \ ^{\circ}\text{C} \]

A power loss reduction can be achieved by lower oil quantities …but we also need to control the raceway temperature
Motivation

Speed Index increases continuously for HP-Bearings

1) calculated with pitch diameter
Motivation & Goal

ECO-POLITICS
- The goals described by the European Commission and The Advisory Council for Aeronautics (ACARE) in "Flightpath 2050": reduction of 75% CO₂, 90% NOₓ, and 65% noise compared to capabilities of typical new aircraft in the year 2000.

END USER REQUIREMENTS
- End user require more efficient and performance-enhanced engine components. Approximately one third of an airline's total operating costs are contributed by kerosene costs.

TECHNICAL REQUIREMENTS
- Rolling element bearings determine significantly the mechanical efficiency of an aircraft engine
- Today's state of the art main shaft bearing feature squeeze film damping in order to reduce vibrational loads
Motivation & Goal

Goal:
Increase of Reliability, Performance and Efficiency of Aircraft Engine Ball Bearings

Further additional reduction compared to an all-steel bearing:
- bearing power loss: 10 %
- required total oil flow: 15 %
- damping of rotor vibrations
- temperatures: 10 K
- increase of max. rotor speed by 20%

Measures:

Use of:
- ceramic balls
- integrated squeeze film damper
- direct outer ring cooling concept
- plasma nitrided raceways
## Test Bearing and Rig Test Head Design, Test Conditions

<table>
<thead>
<tr>
<th>Section</th>
<th>Title</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Motivation &amp; Goal</td>
</tr>
<tr>
<td>2</td>
<td>Test Bearing and Rig Test Head Design, Test Conditions</td>
</tr>
<tr>
<td>3</td>
<td>Experimental Investigation Results: Oil Flow</td>
</tr>
<tr>
<td>4</td>
<td>Experimental Investigation Results: Temperatures</td>
</tr>
<tr>
<td>5</td>
<td>Experimental Investigation Results: Bearing Power Loss &amp; Heat to Oil (HTO)</td>
</tr>
<tr>
<td>6</td>
<td>High Speed Capability</td>
</tr>
<tr>
<td>7</td>
<td>Conclusion &amp; Outlook</td>
</tr>
</tbody>
</table>
Test Bearing Design

- Outer ring 2 (M50Nil)
- Outer ring 1 (M50Nil)
- Silver plated cage
- Ceramic balls (Si₃N₄)
- Inner ring 1 (M50Nil)
- Inner ring 2 (M50Nil)
- Oil distribution groove
- Piston Ring Grooves
- Squeeze Film Damper
- Oil-in cooling oil
- Outer ring cooling channel
- Duplex-hardened raceways
- Radial slots in split face for "under-race" lubrication

The information in this document is the property of FAG Aerospace GmbH & Co. KG and may not be copied or communicated to a third party for any purpose other than that for which it is supplied without express written authority of FAG Aerospace GmbH & Co. KG.
Test Bearing and Rig Test Head Design, Test Conditions

Bearing Components prior to Rig Test

- Outward outer ring featuring piston ring grooves for squeeze film damping
- Inward outer ring featuring helical cooling channel
- Loaded inner ring with radial oil slots in the split face
- Cage/ball assembly
Test Rig: AN62

- Location: Schweinfurt, Germany
- Speed Capability: depending on gearbox installed; w/ current gearbox up to 26000 rpm
- Motor Power: 120 kW (161 hp)
- Axial and radial loads up to 200 kN (45,000 lbf), misalignment testing
- Capability for tests with co- and counter-rotating bearings (intershaft bearings)
- Three independent oil systems
- Sensor technology: static, telemetry, vibrations, piezo, strain gages, chip detectors, cage speed, etc.
Test Bearing and Rig Test Head Design, Test Conditions

**Test Conditions:**
- Rotational Speed: 14000 - 24000 rpm \( (D_m \cdot n = 2.35 \text{ to } 4 \text{ Mio mm/min}) \)
- Axial Load: 26.7 - 80 kN \( (p_0 = 1540 \text{ to } 2440 \text{ MPa}, [224 \text{ to } 354 \text{ ksi}]) \)
- Oil In Temperature: 80°C and 110°C (176 and 230°F)
- Under-race Oil Flow: 5, 6, 8, 10, 12 (14, 15) l/min (1.3 to 4 gallons/min)
- Outer Ring Channel Oil Flow: 0, 1, 2, 3, 4, 6, 8, 10 l/min (up to 0.8 gallons/min)
- Engine Oil per MIL-PRF 23699

**Measured Variables:**
- Outer and Inner Ring Temperatures
- Bearing Power Loss
- Oil Flow Qty, Oil Pressure, Oil Temperatures
- Vibrational Acceleration
- Axial Load, Shaft Rotational Speed
Test Bearing and Rig Test Head Design, Test Conditions
Test Bearing and Rig Test Head Design, Test Conditions

Hertzian Stress

The diagram shows the Hertzian stress $p_0$ (in MPa) plotted against rotational speed (in rpm) for different bearing types and conditions. The following key points are highlighted:

- **F_{Thrust} = 26.7 kN, V_{nom} = 10 l/min, V_o = 2 l/min, T_{Oil In} = 80 °C**

- **Graph Legend**:
  - Orange line: hybrid bearing, inner ring raceway
  - Red line: hybrid bearing, outer ring raceway
  - Purple dashed line: all-steel bearing, inner ring raceway (calculated)
  - Blue line: all-steel bearing, outer ring raceway (calculated)

- **Key Observations**:
  - Identical Hertzian stresses for outer and inner race of steel brg
  - Identical Hertzian stresses for outer and inner race of hybrid brg

The information in this document is the property of FAG Aerospace GmbH & Co. KG and may not be copied or communicated to a third party for any purpose other than that for which it is supplied without express written authority of FAG Aerospace GmbH & Co. KG.
<table>
<thead>
<tr>
<th></th>
<th>Section Title</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Motivation &amp; Goal</td>
</tr>
<tr>
<td>2</td>
<td>Test Bearing and Rig Test Head Design, Test Conditions</td>
</tr>
<tr>
<td>3</td>
<td>Experimental Investigation Results: Oil Flow</td>
</tr>
<tr>
<td>4</td>
<td>Experimental Investigation Results: Temperatures</td>
</tr>
<tr>
<td>5</td>
<td>Experimental Investigation Results: Bearing Power Loss &amp; Heat to Oil (HTO)</td>
</tr>
<tr>
<td>6</td>
<td>High Speed Capability</td>
</tr>
<tr>
<td>7</td>
<td>Conclusion &amp; Outlook</td>
</tr>
</tbody>
</table>
Experimental Investigation Results

Pictures of rig tested hybrid bearing

- Loaded inner ring half
- Outer ring
- Bearing with one inner ring half removed, showing the ceramic balls and the cage
- Cage/ball assembly
- Bearing side view
Experimental Investigation Results: Oil Flow

\[ \eta = \frac{m_{W, left} + m_{W, right}}{m_{nom}} \]

Scoop Efficiency very similar between all-steel and hybrid bearing → enables direct comparison of steel and hybrid bearing
Experimental Investigation Results: Temperatures

<table>
<thead>
<tr>
<th></th>
<th>Motivation &amp; Goal</th>
</tr>
</thead>
<tbody>
<tr>
<td>2</td>
<td>Test Bearing and Rig Test Head Design, Test Conditions</td>
</tr>
<tr>
<td>3</td>
<td>Experimental Investigation Results: Oil Flow</td>
</tr>
<tr>
<td>4</td>
<td>Experimental Investigation Results: Temperatures</td>
</tr>
<tr>
<td>5</td>
<td>Experimental Investigation Results: Bearing Power Loss &amp; Heat to Oil (HTO)</td>
</tr>
<tr>
<td>6</td>
<td>Summary and Conclusion</td>
</tr>
</tbody>
</table>
Experimental Investigation Results: Temperatures

\[ F_{\text{Thrust}} = 80 \text{ kN}, \ n = 15000 \text{ rpm}, \ T_{\text{Oil In}} = 80 \degree \text{C}, \ V_o = 0 \text{ l/min} \]

- **hybrid bearing**
- **all-steel bearing**

**Graph:**
- Mean Outer Ring Temperature [\(^\circ\text{C}\)]
- Nominal Under-Race Oil Flow \( V_{\text{nom}} \) [l/min]

The information in this document is the property of FAG Aerospace GmbH & Co. KG and may not be copied or communicated to a third party for any purpose other than that for which it is supplied without express written authority of FAG Aerospace GmbH & Co. KG.
Experimental Investigation Results: Temperatures

n = 15000 rpm, $V_{nom} = 10$ l/min, $T_{oil \, in} = 80$ °C

- **hybrid bearing, outer ring loaded side, $V_o = 0$ l/min**
- **hybrid bearing, outer ring loaded side, $V_o = 2$ l/min**
- **all-steel bearing, outer ring loaded side, $V_o = 0$ l/min**
- **all-steel bearing, outer ring loaded side, $V_o = 2$ l/min**

Thrust Load $F_{Thrust}$ [kN] vs. Ring Bulk Temperature [°C]
Experimental Investigation Results: Temperatures

\[ F_{\text{Thrust}} = 80 \, \text{kN}, \quad V_{\text{nom}} = 10 \, \text{l/min}, \quad T_{\text{Oil In}} = 80 \, ^\circ\text{C} \]

- hybrid bearing, \( n = 14000 \, \text{rpm} \)
- hybrid bearing, \( n = 15000 \, \text{rpm} \)
- hybrid bearing, \( n = 16000 \, \text{rpm} \)
- hybrid bearing, \( n = 17000 \, \text{rpm} \)
- hybrid bearing, \( n = 19000 \, \text{rpm} \)
- hybrid bearing, \( n = 20000 \, \text{rpm} \)
- all-steel bearing, \( n = 13000 \, \text{rpm} \)
- all-steel bearing, \( n = 15000 \, \text{rpm} \)
- all-steel bearing, \( n = 16000 \, \text{rpm} \)
- all-steel bearing, \( n = 17000 \, \text{rpm} \)
<table>
<thead>
<tr>
<th></th>
<th>Title</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Motivation &amp; Goal</td>
</tr>
<tr>
<td>2</td>
<td>Test Bearing and Rig Test Head Design, Test Conditions</td>
</tr>
<tr>
<td>3</td>
<td>Experimental Investigation Results: Oil Flow</td>
</tr>
<tr>
<td>4</td>
<td>Experimental Investigation Results: Temperatures</td>
</tr>
<tr>
<td>5</td>
<td>Experimental Investigation Results: Bearing Power Loss &amp; Heat to Oil (HTO)</td>
</tr>
<tr>
<td>6</td>
<td>High Speed Capability</td>
</tr>
<tr>
<td>7</td>
<td>Conclusion &amp; Outlook</td>
</tr>
</tbody>
</table>
HTO = \( m_w \cdot c_p \cdot (T_{Oilout} - T_{OilIn}) \)
Experimental Investigation Results: Bearing Power Loss & Heat to Oil (HTO)

$F_{\text{Thrust}} = 80 \text{ kN}, V_o = 0 \text{ l/min}, T_{\text{Oil, In}} = 80 ^\circ \text{C}$

Hybrid Bearing Power Loss $P_{\text{Total}}$ [kW] vs. Nominal Under-Race Oil Flow $V_{\text{nom}}$ [l/min]

- $n = 20000 \text{ rpm}$
- $n = 19000 \text{ rpm}$
- $n = 17000 \text{ rpm}$
- $n = 16000 \text{ rpm}$
- $n = 15000 \text{ rpm}$
Experimental Investigation Results: Bearing Power Loss & Heat to Oil (HTO)

- $F_{\text{Thrust}} = 26.7 \text{ kN}$, $n = 19000 \text{ rpm}$, $T_{\text{OilIn}} = 80 \degree \text{C}$

- $\text{HTO} = 30.4 \text{ kW}$
- $\text{HTO} = 25.3 \text{ kW}$

- $\Delta \text{HTO} = 5.1 \text{ kW (17\%)}$
- $\Delta V_{\text{total}} = 4.5 \text{ l/min (30\%)}$
- $\Delta V_{\text{nom}} = 7 \text{ l/min (46\%)}$
Experimental Investigation Results: Bearing Power Loss & Heat to Oil (HTO)

F_{Thrust} = 40 \text{kN}, \ \ V_{\text{nom}} = 10 \text{l/min}, \ \ T_{\text{Oil \ in}} = 80 \degree \text{C}

Bearing Power Loss $P_{\text{Total}}$ [kW]

Outer Ring Channel Oil Flow $V_o$ [l/min]
<table>
<thead>
<tr>
<th></th>
<th>High Speed Capability</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Motivation &amp; Goal</td>
</tr>
<tr>
<td>2</td>
<td>Test Bearing and Rig Test Head Design, Test Conditions</td>
</tr>
<tr>
<td>3</td>
<td>Experimental Investigation Results: Oil Flow</td>
</tr>
<tr>
<td>4</td>
<td>Experimental Investigation Results: Temperatures</td>
</tr>
<tr>
<td>5</td>
<td>Experimental Investigation Results: Bearing Power Loss &amp; Heat to Oil (HTO)</td>
</tr>
<tr>
<td>6</td>
<td><strong>High Speed Capability</strong></td>
</tr>
<tr>
<td>7</td>
<td>Conclusion &amp; Outlook</td>
</tr>
</tbody>
</table>
High Speed Capability

\( F_{\text{thrust}} = 26.7 \text{ kN}, V_{\text{nom}} = 10 \text{ l/min} \)

\( V_o = 2 \text{ l/min}, T_{\text{oil in}} = 80 \degree \text{C} \)

\( D_m \cdot n = 4.02 \text{ Mio mm/min} \)

Speed Index > 4 Mio mm/min at \( T_{OR} < 200\degree \text{C} \) achieved by selective adjustment of under-race and outer ring channel oil flow quantity
High Speed Capability

$F_{\text{Thrust}} = 26.7 \text{ kN}$, $V_{\text{nom}} = 10 \text{ l/min}$, $V_o = 2 \text{ l/min}$, $T_{\text{Oil in}} = 80 \degree \text{C}$

- mean outer ring temperature
- bearing power loss
- radial acceleration

Range of $2f_{\text{elg}}$
## Conclusion & Outlook

<table>
<thead>
<tr>
<th>1</th>
<th>Motivation &amp; Goal</th>
</tr>
</thead>
<tbody>
<tr>
<td>2</td>
<td>Test Bearing and Rig Test Head Design, Test Conditions</td>
</tr>
<tr>
<td>3</td>
<td>Experimental Investigation Results: Oil Flow</td>
</tr>
<tr>
<td>4</td>
<td>Experimental Investigation Results: Temperatures</td>
</tr>
<tr>
<td>5</td>
<td>Experimental Investigation Results: Bearing Power Loss &amp; Heat to Oil (HTO)</td>
</tr>
<tr>
<td>6</td>
<td>High Speed Capability</td>
</tr>
<tr>
<td>7</td>
<td>Conclusion &amp; Outlook</td>
</tr>
</tbody>
</table>
Conclusion & Outlook

Specific Benefits:
- Oil Flow Savings up to 50 %
- Power Loss Reduction up 25 %
- up to 25 K lower temperatures
- High speed capability above $4 \cdot 10^6$ mm/min

Overall Benefits
- Increase in mechanical efficiency (bearing & engine)
- Reduction of weight (smaller oil pump etc.)
- Reduction of engine fuel consumption
- Reduction of engine emissions
- Improvement in material fatigue strength (from lower brg temperature)
- Increase of reliability
- Reduction of total cost
Conclusion & Outlook

Fuel Consumption Benefit by using Direct Outer Ring Cooled Hybrid Ball Bearing

Calculation Example:
- Bearing Power Loss Reduction: 4 kW per bearing
- #3 ball bearing (HP shaft) with DORC
- 4 kW saving per gas turbine
- Heat Value (Jet A1): 42500 kJ/kg
- Overall Engine Efficiency: 38%
- Gas Turbine fleet: approx. 5000 engines in service

→ Kerosene savings per engine: 7800 kg/a (CO₂ savings: 25 t/a)
→ Kerosene savings for fleet: 39000 t/a (CO₂ savings: 123000 t/a)

→ USD savings* for fleet: 16 Mio USD/a (45000 USD/d)
→ USD savings** for fleet: 31 Mio USD/a (84000 USD/d)

* US$/bbl = 53; ** US$/bbl = 100;
Thank you for your attention!