ON THE PREDICTED PERFORMANCE OF OIL LUBRICATED THRUST COLLARS IN INTEGRALLY GEARED COMPRESSORS

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Integrally Geared Compressors

Compared to single shaft multistage compressors, industry selects IGCs for their:

• increased thermal efficiency,
• decreased footprint, &
• ease of access for maintenance and overhaul.

All pictures & components are a courtesy of Samsung Techwin
Components of an IGC

- pinion gears
- pinion G/B frame
- suction-adapter castings
- impellers
- bull gears
- IGV assemblies
- volute castings
- pinion thrust bearings
- journal bearings
- journal bearings
The bull gear/pinion gear

- Pinion (helical) gear
- Bull gear
- Impellers - compressor
- Pinion trust/radial bearing
- Bull gear trust/radial bearing
- THRUST COLLARS
The thrust collar (TC)

Lubricated zone in thrust collar transmits axial load from pinion shaft & gear to bull gear shaft.

Load is from gas pressure acting on the front and back sides of an impeller plus the axial component of the transmission contact force in a helical gear.
IGCs performance and customization

1. Ensure Operational Reliability
   - Minimize vibration
   - Design for coupling life and endurance
   - Lasting seals

2. Offer High Performance
   - On design
   - And off design
   - Minimize parasitic losses

3. Field Support
   - Interchangeability
   - Common components

4. Baseline Unit Cost
   - Master impellers
   - Standardized subcomponents
   - Controlled development

Design based on validated predictive tools

Build a process that targets user concerns on core
# The thrust collar (TC)

<table>
<thead>
<tr>
<th>Year</th>
<th>Author(s)</th>
<th>Title</th>
<th>Source</th>
</tr>
</thead>
</table>

- **1968**: Empirical formula for diametral interference fit. TestedSeven configurations. Gives design rule (taper angles).
- **1991**: Introduce IGCs in USA: operation and design guidelines. TCs save space and complexity as they dispense with large OD hydrodynamic thrust bearings.
- **2006**: Introduce FE structural models. A safety ring near a press fit TC increases its load capacity before TC slips.
- **2006, 2009**: Complete EHD analysis of TCs to optimize geometry for largest load at design speed. Only one taper angle.
Objective

To advance engineering analysis for prediction of the unsteady forced response of thrust collars in geared compressors.

**Unknown:**
Selection of taper angles to improve TC performance

**Design Practice:**
Nominal taper angle < 2 degrees.

No certain knowledge on taper angle and its effect on TC performance, in particular for load capacity.
Means to the objective

- Produce lubrication model for prediction of film pressure (static and dynamic), temperature rise, power loss, reaction force & moments in a thrust collar.

- Perform parametric studies to determine effect of taper angles on performance of thrust collar.

- Calibrate tool predictions against experimental or published results.

Numerical FE model for prediction of pressure field an extensions to include thermal energy transport

Maximize film thickness and stiffness, minimize flow rate and friction coefficient.

Parallel test program at TurboLab (Dr. Childs)
Kinematics of thrust collar

\( \omega_B : BG \) speed

\( \omega_{TC} : PG \) speed

Film thickness (exaggerated)

\( \phi : \) taper angle
**Generation of hydrodynamic pressure**

**Assumptions**

Laminar thin film flow.
Incompressible lubricant.
Rigid surfaces. Steady state.

\[
\frac{1}{r} \frac{\partial}{\partial r} \left( r \frac{h^3}{12 \mu} \frac{\partial p}{\partial r} \right) + \frac{1}{r} \frac{\partial}{\partial \theta} \left( \frac{h^3}{12 \mu} r \frac{\partial p}{\partial \theta} \right) = \frac{1}{r} \frac{\partial}{\partial r} \left\{ r \left[ b \omega_B \sin(\varepsilon) \right] \left( \frac{h}{2} \right) \right\} + \frac{1}{r} \frac{\partial}{\partial \theta} \left\{ \left[ b \omega_B \cos(\varepsilon) + r \omega_{TC} \right] \left( \frac{h}{2} \right) \right\}
\]

- \( \omega_B \): BG speed
- \( \omega_{TC} \): PG speed
- \( \mu \): oil viscosity
- \( h \): film thickness

\[
h_{(r, \theta)} = h_{R_i} + \left[ \left( R_1 - d + b_{(r, \theta)} \right) \tan(\phi_B) \right] - \left[ \left( R_1 - r \right) \tan(\phi_{TC}) \right]
\]

\( \phi \): taper angles
Numerical method of solution

Solve Reynolds equation with isoparametric finite element with bi-linear shape functions.

Greate GUI: engineering design tool.

INPUT: thrust collar configuration, operating speeds, oil type, and applied force,

OUTPUT: pressure field, drag torque and power loss, flow rate.
Numerical method of solution

Axial force coefficients

Perturbation of Reynolds equation for small amplitude axial displacements leads to PDEs to obtain axial and stiffness damping force coefficients

\[ K_z + i \omega C_z = - \int_{r_{\text{left}}}^{R_1} \int_{\theta_{\text{min}}}^{\theta_{\text{max}}} p_z r \, dr \, d\theta \]
Thrust collars TAPER ANGLES

BG taper angle > TC angle

TC taper angle > B angle

Surface taper angles define performance of TC
Effect of taper angles on TC performance

**Question:** How do small changes in taper angle affect the operating film thickness, friction, flow, and force coefficients of a TC?

\[ \bar{\phi}_B - \bar{\phi}_{TC} = 0.1 \text{ and } 0.2 \]

\[ \bar{\phi}_B = \bar{\phi}_{TC} \text{ nominal} \]

\[ \bar{\phi}_{TC} - \bar{\phi}_B = 0.1 \text{ and } 0.2 \]
### Example of TC performance

<table>
<thead>
<tr>
<th>Operating Conditions</th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>Load</td>
<td>$W$</td>
<td>1.0</td>
</tr>
<tr>
<td>Speed (BG/TC)</td>
<td>$\bar{\omega}$</td>
<td>10/115</td>
</tr>
<tr>
<td>Geometry</td>
<td>$R_2/R_1$</td>
<td>7.14</td>
</tr>
<tr>
<td></td>
<td>$d/R_1$</td>
<td>7.78</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Lubricant ISO VG 32</th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>Supply Temperature</td>
<td>$T_s$</td>
<td>40 °C</td>
</tr>
<tr>
<td>Dynamic Viscosity (40°C)</td>
<td>$\mu$</td>
<td>0.0275 Pa.s</td>
</tr>
<tr>
<td>Ambient Pressure</td>
<td>$p_a$</td>
<td>100 kPa</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Lubricated Zone</th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>Max. angle</td>
<td>$\theta_{max}$</td>
<td>47.3°</td>
</tr>
<tr>
<td>Length</td>
<td>$c/R_1$</td>
<td>1.47</td>
</tr>
<tr>
<td>Width at $\theta = 0$</td>
<td>$l/R_1$</td>
<td>0.36</td>
</tr>
<tr>
<td>Area</td>
<td>$A_{lub}/\pi R_1^2$</td>
<td>0.12</td>
</tr>
</tbody>
</table>

$W/A = 5.48$ MPa
Pressure & film thickness fields

\[ \varphi_B = \varphi_{TC} \]

\[ \varphi_B = 1.25 \varphi_{TC} \]

\[ \bar{P} = \frac{p}{p^*} = \frac{p}{A_{lub}}/W^* \]

\[ \varphi_B = 0.8 \varphi_{TC} \]
Pressure & flow field

\[ \varphi_B = \varphi_{TC} \]

Region of cold lubricant entering thrust collar.
Important for adequate location of oil source and to avoid starvation.

Max pressure \( \tilde{p}_{\text{max}} = 5.24 \)

Min film \( \tilde{h}_{\text{min}} = 0.591 \)
Peak film pressure

Film peak pressure increases with taper angle

\[ \bar{P} = \frac{p A_{\text{lub}}}{W^*} \]

\( \Phi_B > \Phi_{TC} \) by 0.1

\( \Phi_B = \Phi_{TC} \)

\( \Phi_{TC} > \Phi_B \) by 0.2

\( \Phi_{TC} > \Phi_B \) by 0.1
Minimum film thickness

Film thickness ~ uniform for taper angles > 0.6

$\varphi_B = \varphi_{TC}$ gives the largest minimum film thickness.

$\overline{h} = h / h^*$
Friction coefficient $f$

$\phi_B = \phi_{TC}$ gives the lowest drag friction.

As the TC taper angle increases, large pressures in the lubricated zone drive the increase in friction.

$f \sim 0.003 \Rightarrow$ Drag power $\sim$ a few kW.
Axial stiffness

Normalized Axial Stiffness

$\bar{K}_z = K_z \left( \frac{h^*}{W} \right)$

$W \approx 100 \text{ MN/m}$

$W = 1.0$

Stiffness decreases with taper angle. Magnitudes are large. $\varphi_B = \varphi_{TC}$ select
Axial damping

Damping decreases with taper angle. Magnitudes are good (damping ratio > 0.2)
Conclusions I

For the specific TC, taper angles

\[ \bar{\phi}_B = 0.4 \text{ and } \bar{\phi}_{TC} = 0.6 \]

lead to the lowest friction factor \((f \approx 0.0027)\), produce large axial stiffness and damping, while operating with an adequate minimum film thickness.

The peak hydrodynamic pressure is \(~300\) bar \((4.35\) kpsi\). Its effect on a local elastic deformation of the lubricated surfaces is yet to be determined.
Effect of load on TC performance

Question: How does load affect the operating film, friction and force coefficients of a TC?

Select:

\[
\frac{\phi_B}{\phi_{TC}} = 0.4, \quad \frac{\phi_B}{\phi_{TC}} = 0.6 \quad \text{and} \quad \frac{\phi_B}{\phi_{TC}} = 0.6
\]
Film thickness decreases with load. Largest for same B & TC taper angles.
Friction factor vs. load

Friction factor drops with load. All TC configurations give similar performance.

\[ f = \frac{\phi}{\omega_{TC} \bar{W}^* R_1} \]

- \( \bar{\phi}_B = \bar{\phi}_{TC} = 0.6 \)
- \( \bar{\phi}_{TC} = 0.4, \bar{\phi}_B = 0.6 \)
- \( \bar{\phi}_{TC} = 0.6, \bar{\phi}_B = 0.4 \)

\( \bar{W} = 1.0 \)
Both $K$ & $C$ increase with load. TC with $\phi_{TC} = 0.6 > \phi_B = 0.4$ gives largest $K$ & $C$. 

\[ K_z = K_z \left( \frac{h^*}{W} \right) \]

\[ C_z = C_z \left( \frac{h^* \omega_{TC}}{W} \right) \]
Conclusions II

As load increases, min. film decreases and peak pressure rises, force coefficients increase.

For the specified TC, taper angles

$$\bar{\phi}_B = \frac{0.4}{0.6}, \quad \bar{\phi}_B = \frac{0.6}{0.4}, \quad \text{and} \quad \bar{\phi}_B = \frac{0.6}{0.6}$$

Produce large axial stiffness and damping.

TC system damping ratio is good:

$$\xi = \left(\frac{\omega_{TC} C_z}{2 K_z}\right) = \left(\frac{\bar{C}_z}{2 \bar{K}_z}\right) \rightarrow 0.30 \text{ to } 0.42$$

Predictive tool ready for engineering path analysis and design.
Acknowledgments

Thanks to Samsung Techwin

Questions (?)

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

• Numerical solution of thermal energy transport equation

• Analysis for determining misalignment effects

• Determination of mechanical deformation effects
Thermal energy transport

Temperature raise causes drop in lubricant viscosity affecting performance of TC. Model uses an effective film temperature \( T_{\text{eff}} \) across lubricated area.

\[
\rho c_p \left[ \frac{\partial}{\partial t} \left\{ h T_{\text{eff}} \right\} + \frac{1}{r} \frac{\partial}{\partial r} \left\{ r q_r T_{\text{eff}} \right\} + \frac{1}{r} \frac{\partial}{\partial \theta} \left\{ q_\theta T_{\text{eff}} \right\} \right] + \bar{h}_B \left( T_{\text{eff}} - T_B \right) + \bar{h}_{TC} \left( T_{\text{eff}} - T_{TC} \right) = \Phi
\]

Convection of heat by fluid flow

Convection into BG and TC surfaces

\[ \Phi = \frac{h^3}{12 \mu} \left[ \left( \frac{\partial p}{\partial r} \right)^2 + \left( \frac{\partial p}{r \partial \theta} \right)^2 \right] + \frac{\mu}{h} \left[ \left( b \omega_B \right)^2 - 2 \left( b \omega_B \cos(\varepsilon) r \omega_{TC} \right) + \left( r \omega_{TC} \right)^2 \right] \]