VERTICAL INTEGRAL THRUST BEARING UPGRADE
Outline

- Pump Background
- Problem
- Solution
- Calculations
- Benefits
- Lesson learned
- Current status
Service

- Produced Water
- 2 pump in series
- Pump 1
  - Suction: 220 PSIG
  - Discharge: 1720 PSIG
- Pump 2
  - Suction: 1720 PSIG
  - Discharge: 3320 PSIG
- Temp 130°F
Background

- Pump originally designed and installed by a reputable OEM.
- Pump had been repaired and upgraded several times by various repair shops.
Original Pump design

- Cast Iron bronze construction
- 20 + pumps at site
- Each pump had various upgrades and design changes. Such as the bearing bushing in the housing.
Upgraded pump shown

Upgrade

- Customer wanted to do a complete overhaul on a set of pumps.

- This included
  - HVOF wear parts
  - Duplex material
  - Mechanical seal.
  - Material and flush plan.
  - Integral thrust bearing (discussed later)
Problem

- The end user was going to purchase a new motor for these pumps. Due to the thrust load the motor was expensive and had a long lead time. The end user wanted an alternative solution.
- Customer wanted options when upgrading other pumps in the field.
Solution

- The engineering team offered a vertical thrust bearing as a possible solution.
Selection Process

- Calculate Pump down thrust.
  - Thrust and speed determine bearing thrust pot
- Select a Bearing to fit housing foot print.
- Verify Bearing life will meet requirements
- Verify 2\textsuperscript{nd} pump in series can handle up-thrust from first pump during start up.
Free Body Diagram

\[ F_{slv} = \text{Force on sleeve} \]

\[ w_r = \text{Rotor Weight} \]

\[ F_H = \text{Hydraulic force} \]

\[ F_s = \text{Force on shaft end} \]
Calculations – Thrust

Net Pump assembly axial thrust:
\[ F_n = F_H + W_r - F_{slv} - F_S \]

- Net Bowl hydraulic thrust:
\[ F_H = k \times H_{stg} \times N_{stg} \times SG \]

Thrust factor:
\[ k = \frac{SG}{2.31} \times C \times A_b \]

C = experimental thrust coefficient

Unbalanced area:
\[ A_b = \frac{\pi}{4} \times (D_1^2 - D_b^2) \]

\[ W_r = \text{Rotor weight} \]
\[ F_{slv} = \text{Force shaft sleeve} \]
\[ F_S = \text{Force on end of shaft} \]
## Calculation – thrust

<table>
<thead>
<tr>
<th></th>
<th>Pump 1 @ Design</th>
<th>Pump 2 @ Design</th>
<th>Pump 1 @ Min. Flow</th>
<th>Pump 2 @ Min. Flow</th>
</tr>
</thead>
<tbody>
<tr>
<td>FH</td>
<td>+6060</td>
<td>+6060</td>
<td>+8333</td>
<td>+8333</td>
</tr>
<tr>
<td>Wr</td>
<td>+305</td>
<td>+305</td>
<td>+305</td>
<td>+305</td>
</tr>
<tr>
<td>Fslv</td>
<td>-6</td>
<td>-48</td>
<td>-6</td>
<td>-64</td>
</tr>
<tr>
<td>Fs</td>
<td>-447</td>
<td>-3379</td>
<td>-447</td>
<td>-4478</td>
</tr>
<tr>
<td>Fn (Total)</td>
<td>5912 lbf</td>
<td>2938 lbf</td>
<td>8184 lbf</td>
<td>4095 lbf</td>
</tr>
</tbody>
</table>

+ Down direction
- Up direction
Calculations continue.

The radial bearing is used to take the momentary up-thrust during and before start-up. The bearing selected is a radial deep groove bearing with a static load rating of 14400 lbf. According to the bearing manufacture the allowable axial load is 0.5*14400 lbf.

\[ F_{static} = W_r - F_s \]
\[ F_{start} = W_r - F_s - 30\% \cdot F_H \]

<table>
<thead>
<tr>
<th>Up thrust on Second Pump</th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>F(static)</td>
<td>-3495 lbf</td>
</tr>
<tr>
<td>F(start)</td>
<td>-5313 lbf</td>
</tr>
</tbody>
</table>
Calculations – Bearing life

\[ L_{10mhD} = a_1 \cdot a_{skf} \cdot \frac{1000000}{60 \cdot n} \cdot \left( \frac{C}{.93 \cdot P_d} \right)^{\frac{10}{3}} \]

- Where:
  - \( C = 165000 \text{ lbf} \) - bearing load rating
  - \( n = 3560 \text{ rpm} \)

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<tr>
<td>P @ Design</td>
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<td>2938 lbf</td>
</tr>
<tr>
<td>P@ Min Flow</td>
<td>8184 lbf</td>
<td>4095 lbf</td>
</tr>
<tr>
<td>L10 @ Design</td>
<td>308,794 hr</td>
<td>3,176,403 hr</td>
</tr>
<tr>
<td>L10 @ Min Flow</td>
<td>132,996 hr</td>
<td>1,337,127 hr</td>
</tr>
</tbody>
</table>
X-Section of Bearing

- Fan
- Radial Deep Groove Bearing
- Spherical Roller Thrust bearing
- Shroud
- Shaft Sleeve/oil carrier
- Spring
- Cooling tube
Design changes

- New shaft Design
- New Coupling
- Modifications to the Head and Motor Stand.
Benefits and Disadvantages

- Lowers motor cost.
- Easier bearing maintenance.
  - Can leave pump motor in place.

- Bearing housing has to be removed to service Mechanical seal.
Lessons Learned

- Allow more room for design changes to accommodate a more maintenance friendly Discharge head. – Mechanics would of like more room for installing the Mechanical seal.
- The large start-up up-thrust was limited by the radial bearing.
Current Status

- Pumps are running.
Questions?