43rd Turbomachinery
30th Pump SYMPOSIA
GEORGE R. BROWN CONVENTION CENTER
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VERTICAL INTEGRAL THRUST BEARING UPGRADE

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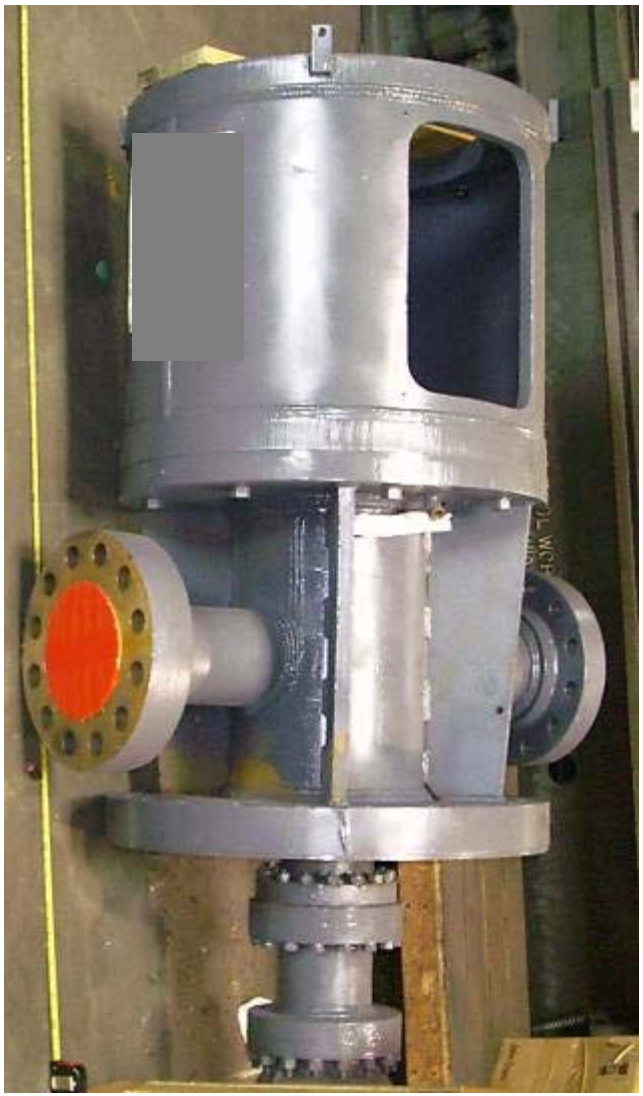
Outline

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



Service

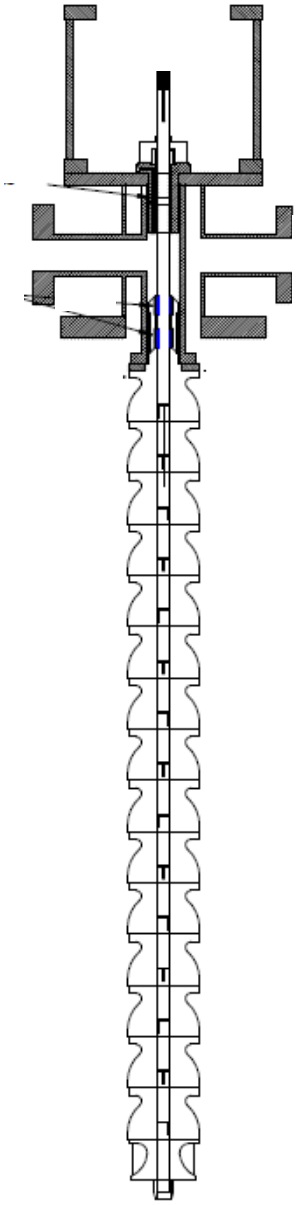
- Produced Water
- 2 pump in series
- Pump 1
  - Suction: 220 PSIG
  - Discharge: 1720 PSIG
- Pump 2
  - Suction: 1720 PSIG
  - Discharge: 3320 PSIG
- Temp 130°F

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.

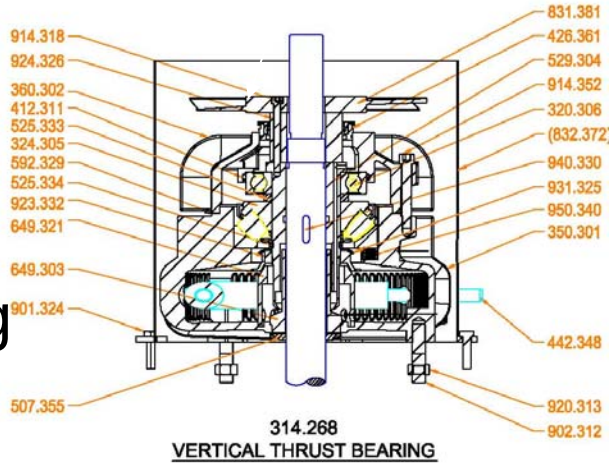


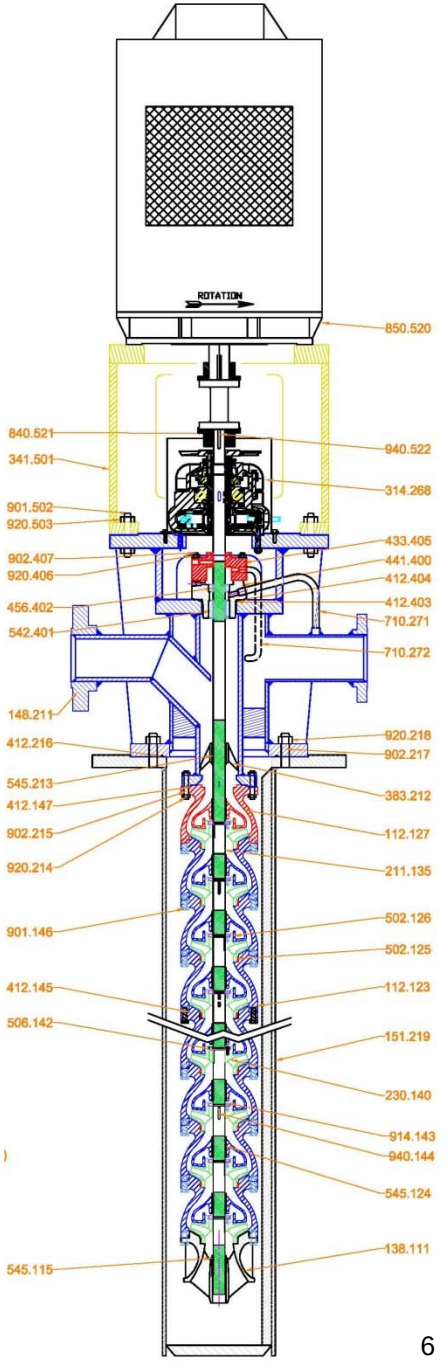
Upgrade

- 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)

Upgraded pump shown





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 alterative solution.
- Customer wanted options when upgrading other pumps in the field.

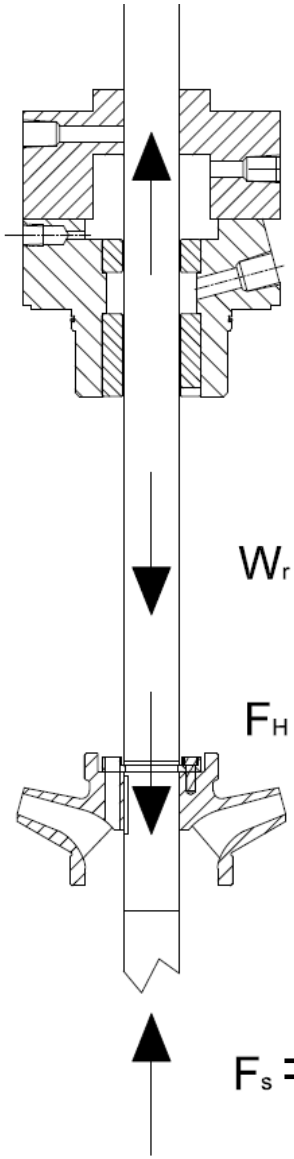
Solution

- 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 2nd pump in series can handle up-thrust from first pump during start up.

Free Body Diagram



F_{slv} = Force on sleeve

W_r =Rotor Weight

F_H = Hydraulic force

F_s = 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:

$$F_H = k * H_{stg} * N_{stg} * SG$$

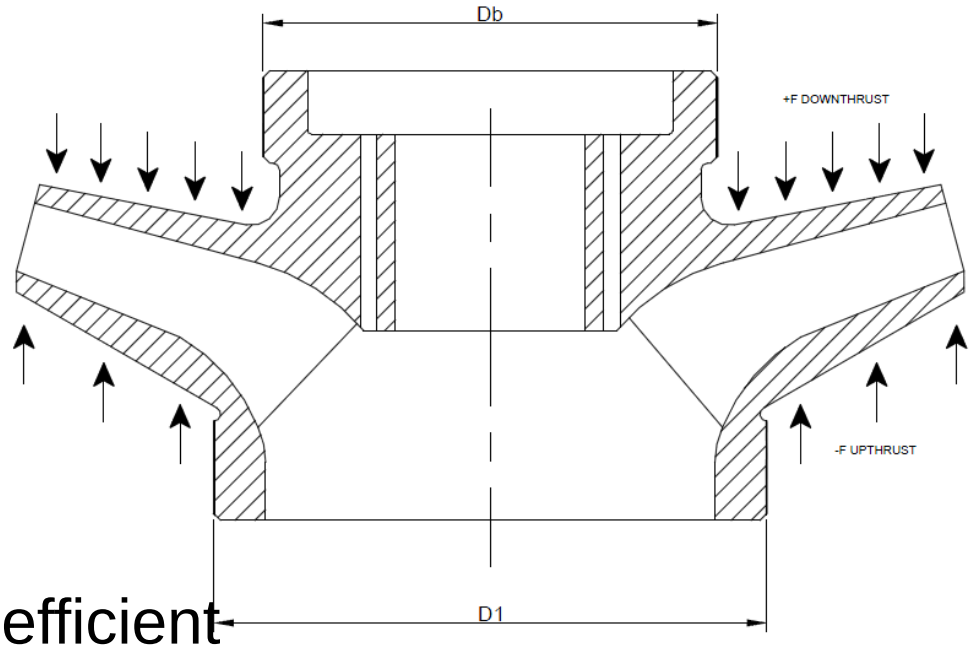
Thrust factor:

$$k = \frac{SG}{2.31} * C * A_b$$

C= experimental thrust coefficient

Unbalanced area:

$$A_b = \frac{\pi}{4} * (D_1^2 - D_b^2)$$



W_r = Rotor weight

F_{slv} = Force shaft sleeve

F_s = Force on end of shaft

Calculation – thrust

	Pump 1 @ Design	Pump 2 @ Design	Pump 1 @ Min. Flow	Pump 2 @ Min. Flow
FH	+6060	+6060	+8333	+8333
Wr	+305	+305	+305	+305
Fslv	-6	-48	-6	-64
Fs	-447	-3379	-447	-4478
Fn (Total)	**5912 lbf**	**2938 lbf**	**8184 lbf**	**4095 lbf**

+ 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.

$$F_{static} = W_r - F_s$$

$$F_{start} = W_r - F_s - 30\%^* F_H$$

Up thrust on Second Pump	
F(static)	-3495 lbf
F(start)	-5313 lbf

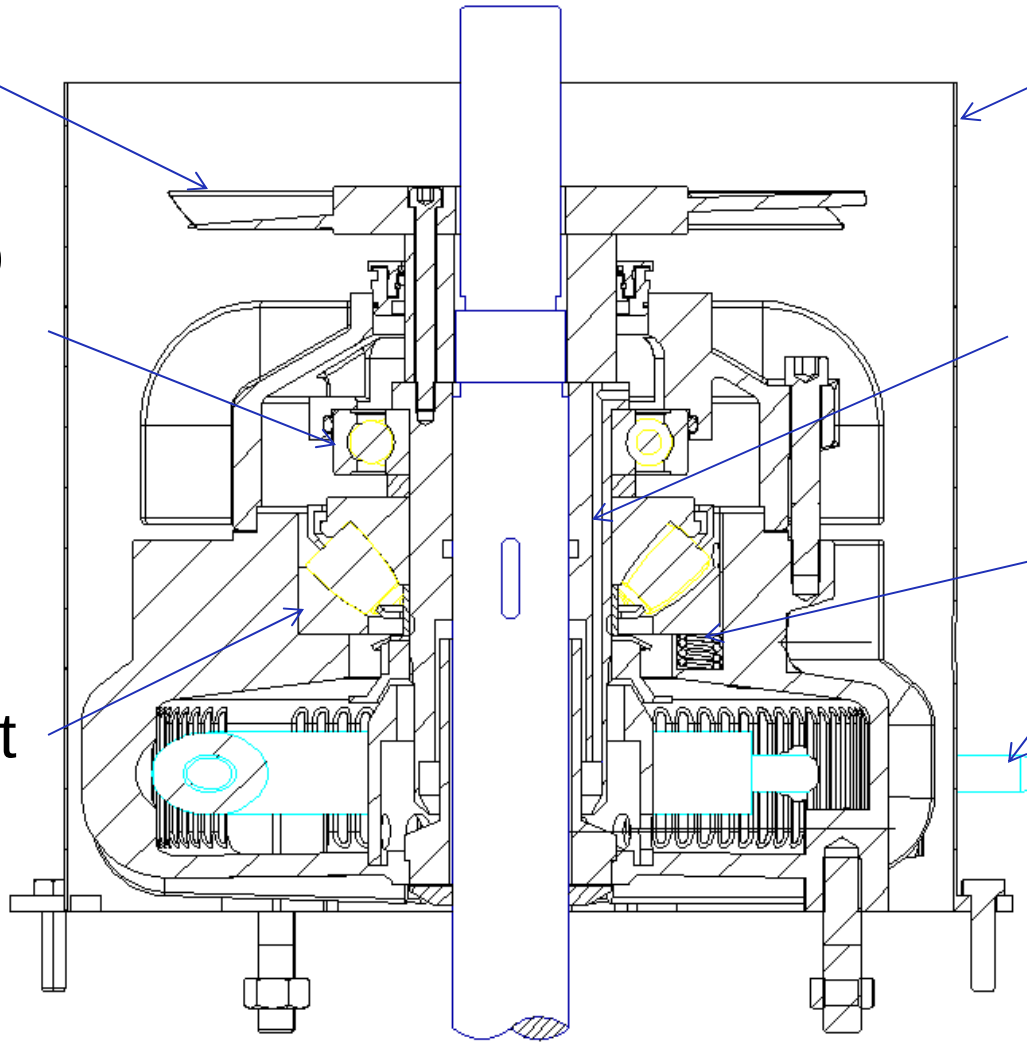
Calculations – Bearing life

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

- Where:
- C = 165000 lbf - bearing load rating
- n = 3560 rpm

	Pump 1	Pump 2
P @ Design	5912 lbf	2938 lbf
P@ Min Flow	8184 lbf	4095 lbf
L10 @ Design	308,794 hr	3,176,403 hr
L10 @ Min Flow	132,996 hr	1,337,127 hr

X-Section of Bearing

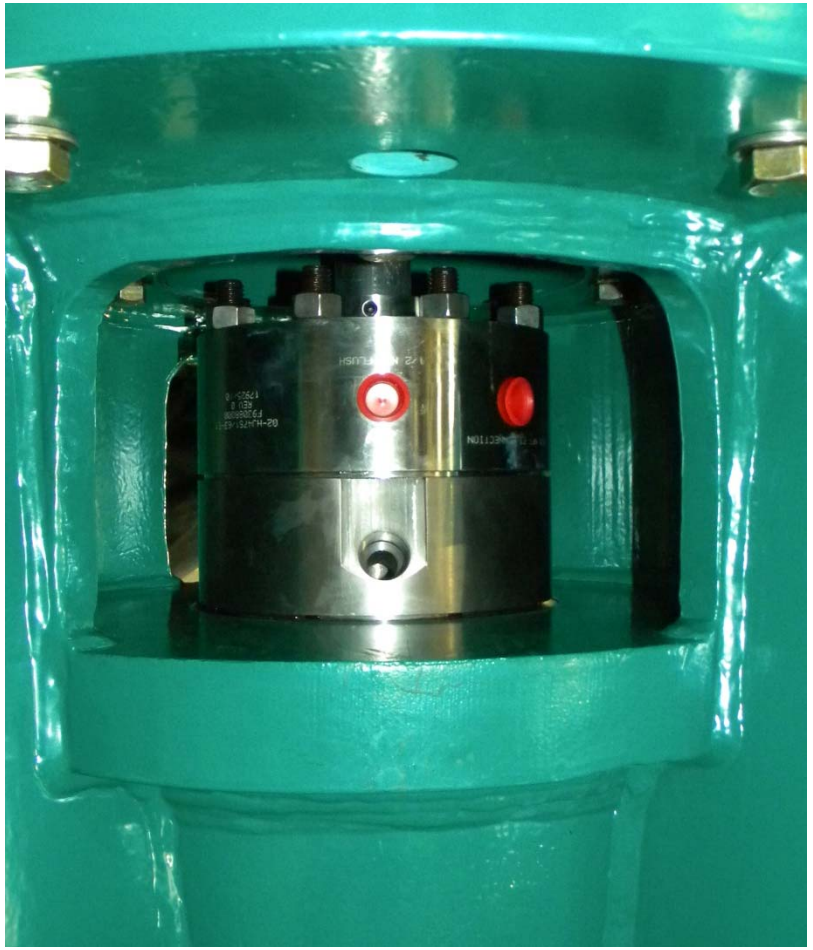


Design changes

- 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?