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GT2017-63385

STATIC LOAD PERFORMANCE OF A WATER LUBRICATED HYDROSTATIC THRUST BEARING

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Justification

- **Electrical submersible pumps (ESPs) are vulnerable to increases in synchronous vibration amplitude caused by modest changes in axial position.**

- **Rotating equipment relies on thrust bearings as a primary means of axial load support and rotor position**.

- **Axial loads in turbomachinery are speed and pressure dependent, their prediction is largely empirical.**

- **Thrust bearing design relies on validated models benchmarked to test data.**

Brief Literature Review

ASME J. of Trib., 122(1) **ASME J. of Trib., 124(1)**

San Andrés (2000, 2002)

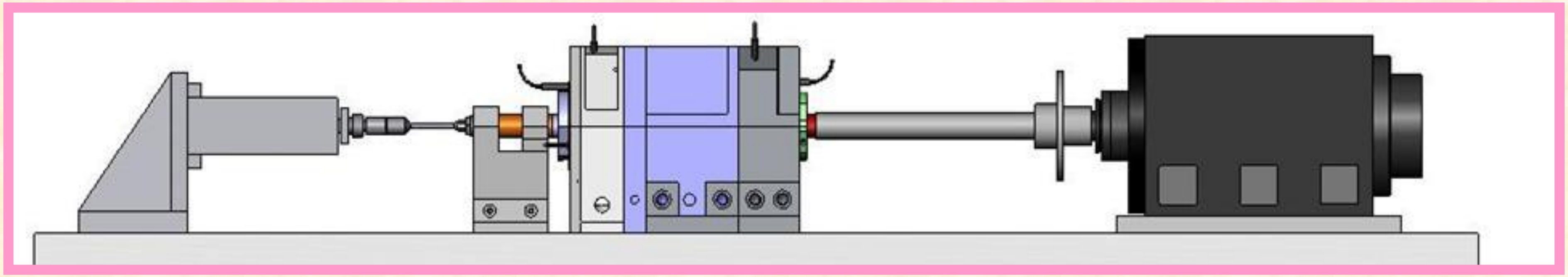
Bulk flow analysis to predict the performance of a multi-recess, orifice-compensated, angled injection hybrid thrust bearing operating with angular misalignment. Application: cryogenic turbopumps.

2008 JANNAF-120 Paper → GT2016-56349 paper (S&D Best Paper Award)

San Andrés, Phillips, and Childs (2008 → 2016)

Test rig to measure thrust bearing performance and validate predictive tool for operation at high rotor speed (17.5 krpm). Water at high pressure (1.72 MPa) supplies bearings. Flow rate measurements show onset of fluid starvation at high rotor speed and large axial clearance (small load). Measurements of axial clearance, recess pressure and flow rate correlate well with predictions.

Description of test rig



Shaker load and stinger

Axial load shaft & test thrust bearing

Rotor & Radial bearings

Coupling

Drive motor

Test Rig Features

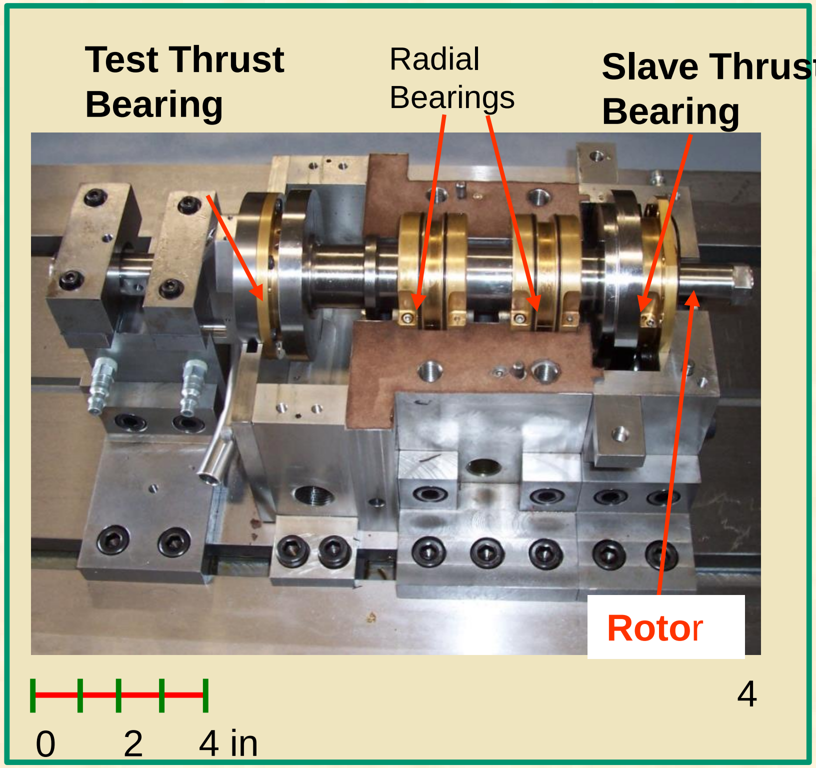
Test Fluid: **WATER**

0-25 krpm,

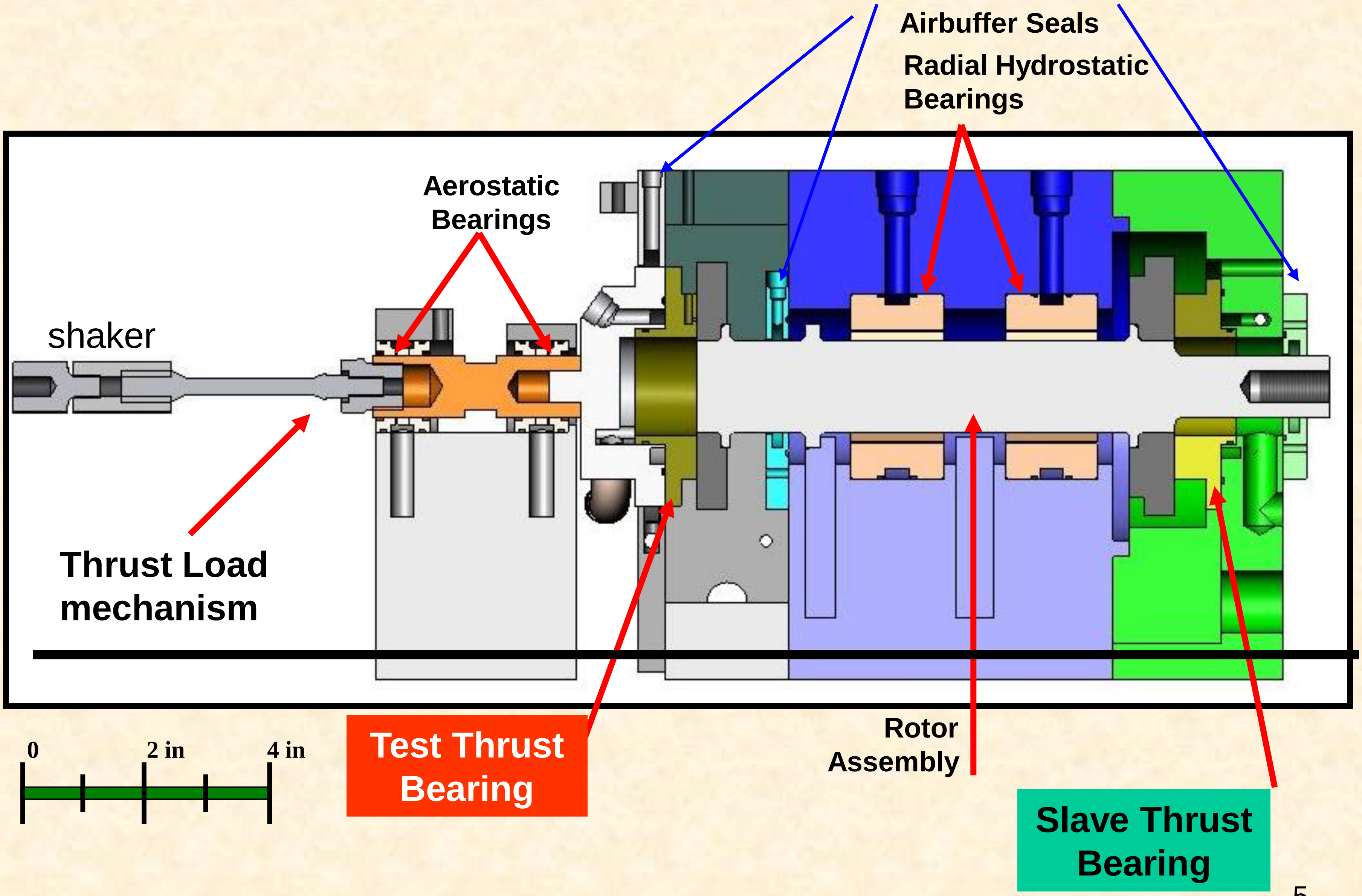
(3.4 to 17 bar) 50-250 psi supply pressure,

Range of static + dynamic axial load: 1000 lbf,

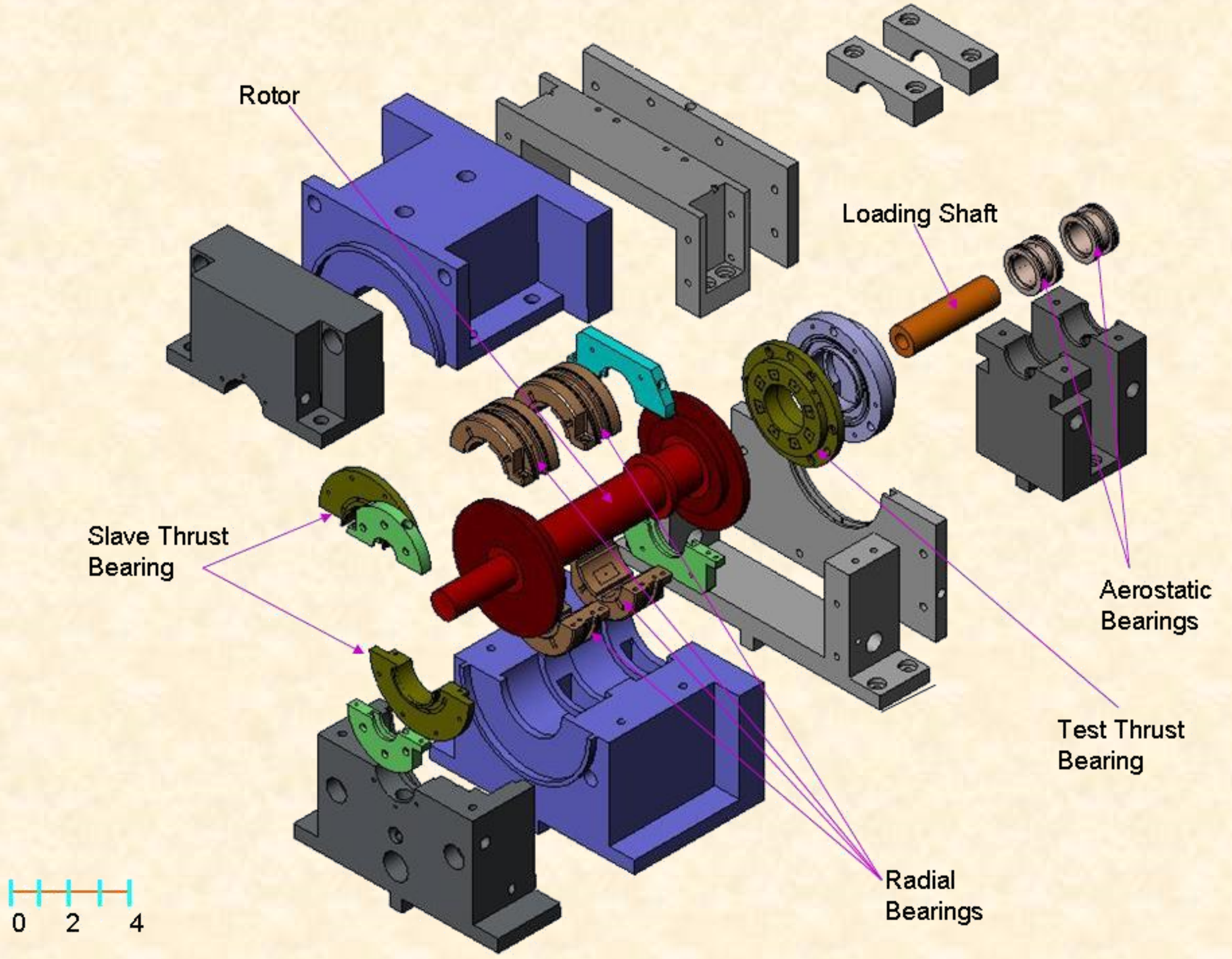
frequency range: 0-600 Hz



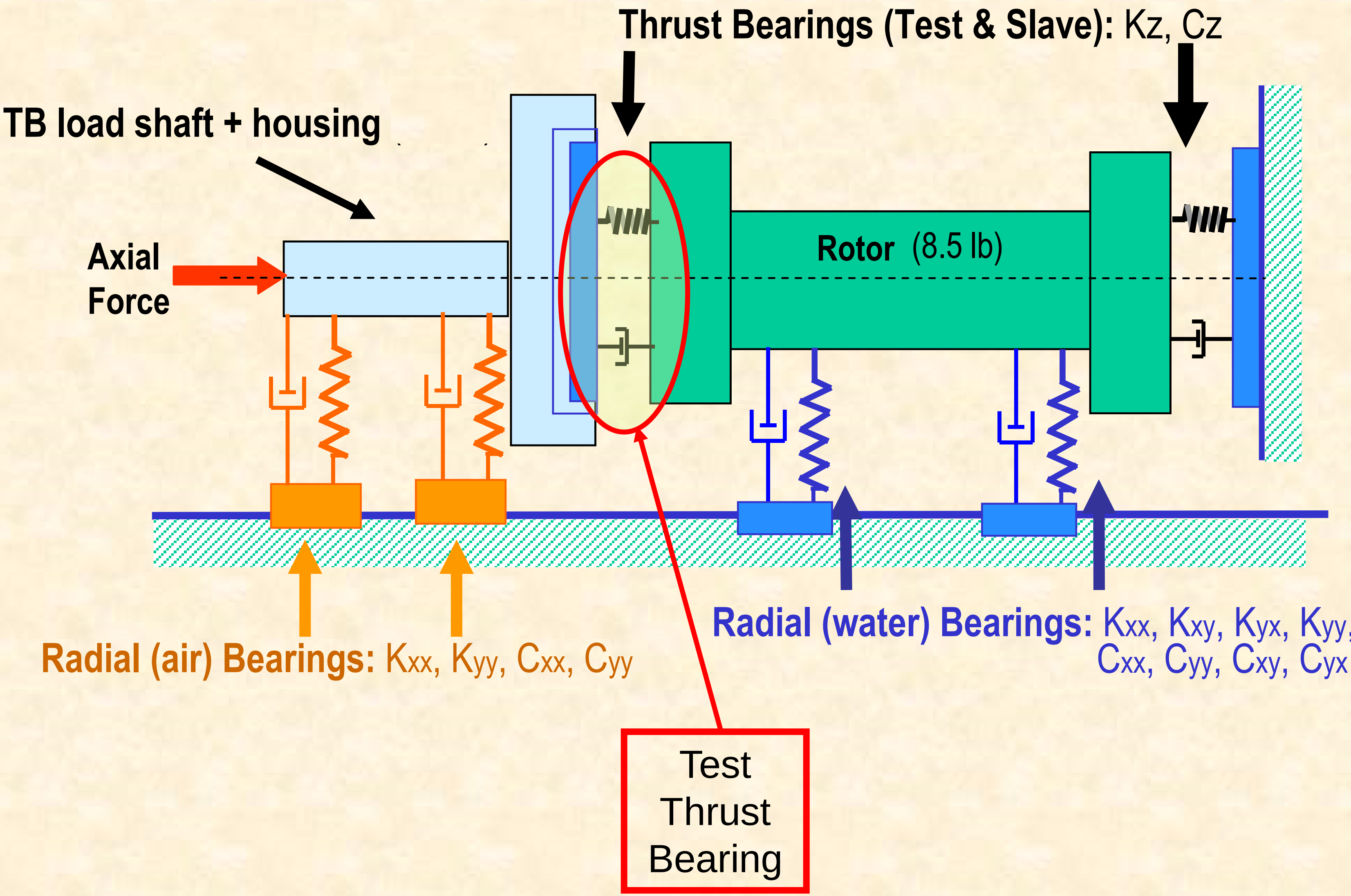
Hybrid Thrust Bearing Rig – Cross Section



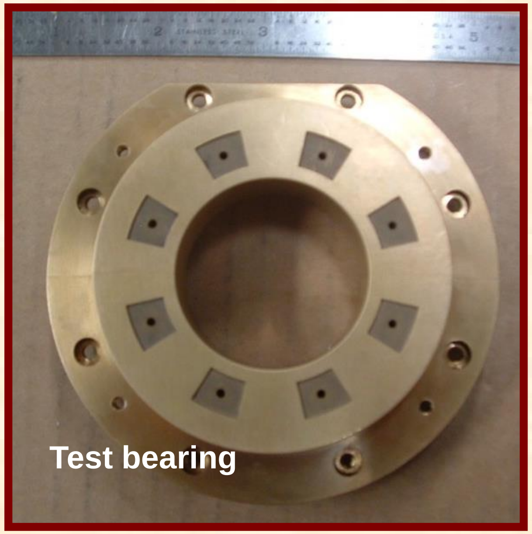
Exploded View of Thrust Bearing Test Rig



Schematic Thrust Bearing Test Rig



Thrust bearings: Test & Slave

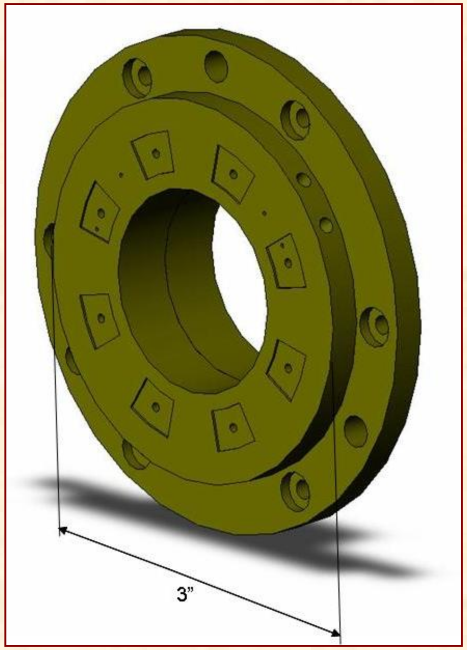


Thrust bearings

Material 660 Bearing bronze
Inner diameter: 1.60 inch
Outer diameter: 3.00 inch
Axial clearance 0.5-5.5 mil

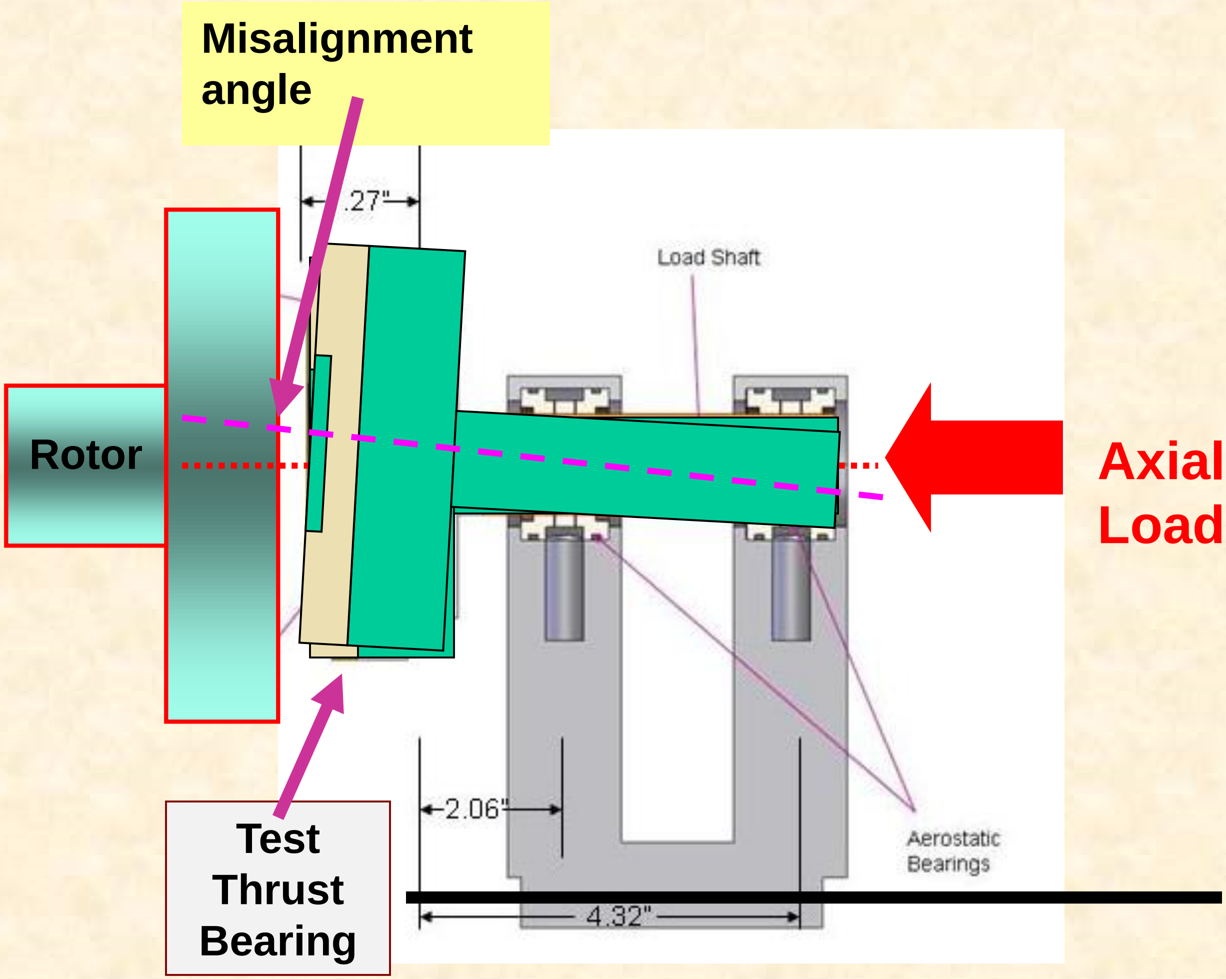
EIGHT (8) Pockets:
Mean Diameter: 2.16 inch
radial length: 0.32 inch
Arc length: 20 degrees
Depth: 0.020 inch
Pocket/wetted area ratio = **19%**

Orifice size: 0.071 inch
Axial injection at r=1.08 inch



Orifice discharge coefficients determined empirically from test data (~0.62)

Loading action and thrust face misalignment

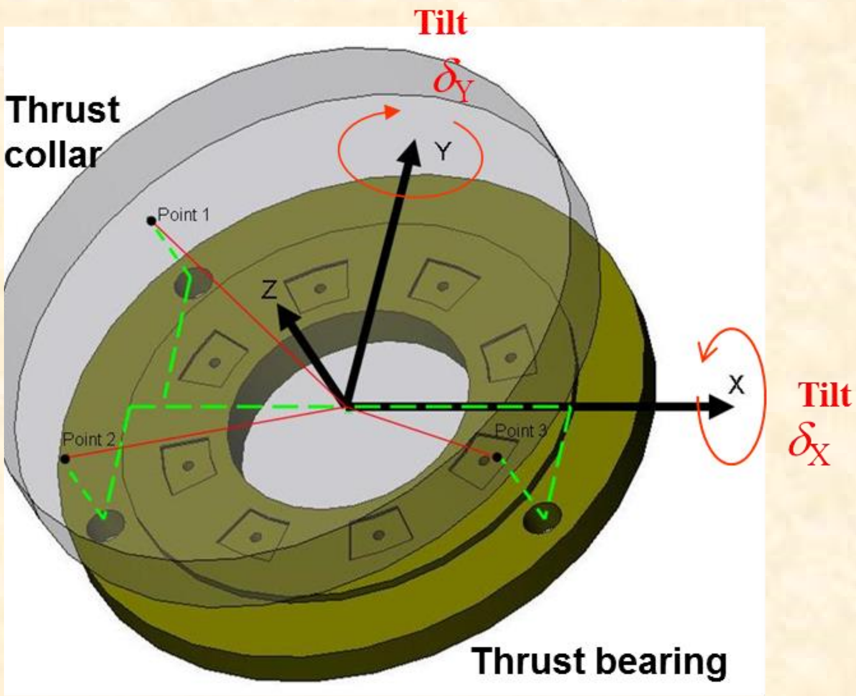


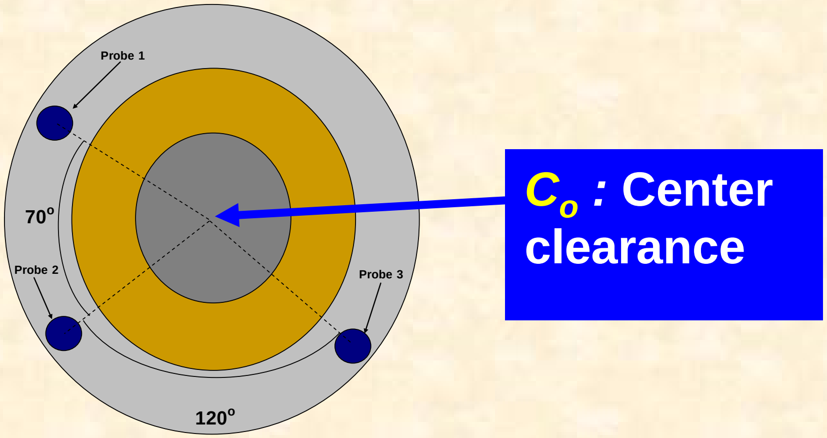
Chronic thrust bearing face misalignment. Worsens with shaft rotation.

TB clearance (*c*) and tilt angles (δ)

Axial clearance measured at three angular locations

→ estimate center clearance and tilts (rotations).



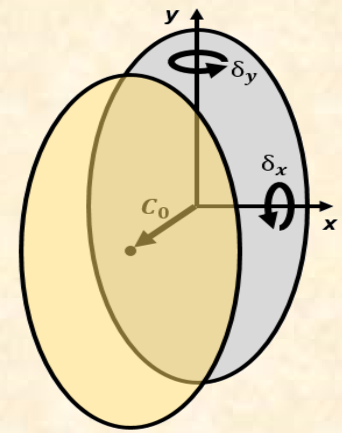


$$c_i = c_o + \underline{R}\cos(\theta_i)\,\delta_Y + \underline{R}\sin(\theta_i)\,\delta_X,\ \mathrm{i}=1,2,3$$

Measurements

$C(x,y) = C_0 + \delta_x y + \delta_y x$

Controlled Inputs	Measured Outcomes
Water at T= 24C-31C	Axial Clearance (C_0)
Supply Pressure (P_S)	Tilt about x-axis (δ_x)
Axial Load *(W)*	Tilt about y-axis (δ_y)
Rotor Speed (*N*)	Supply Flow Rate (Q_S)
	Flow Rate through Inner Diameter (Q_{ID})
	Recess or Pocket Pressure (P_R)

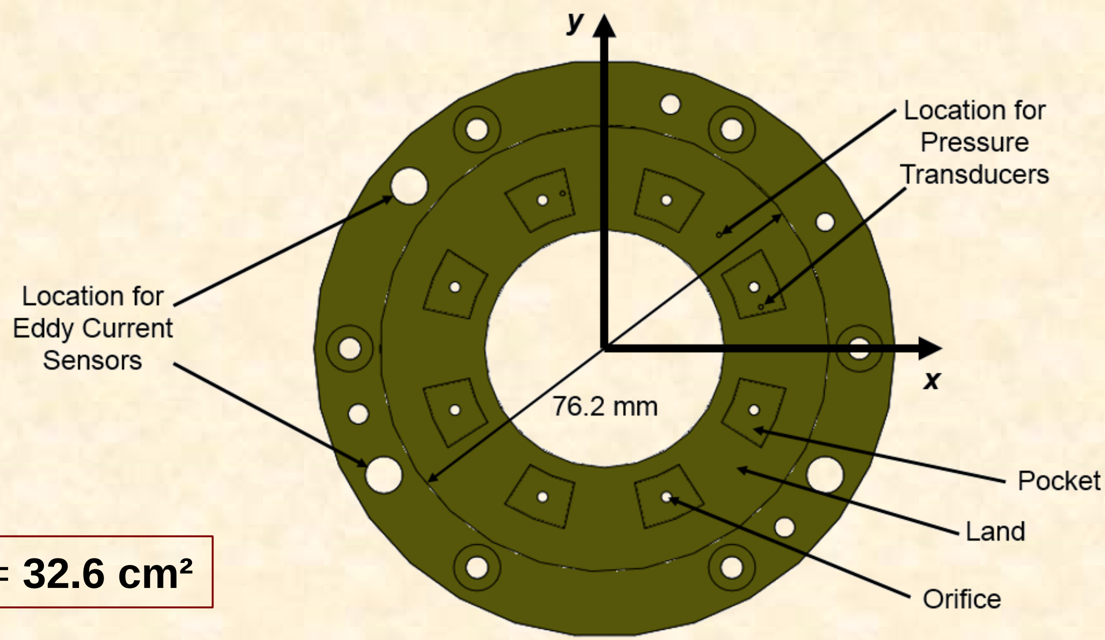


Note: Unless stated otherwise:

- **Error bars in axial clearance (C_o) and tilt angles indicate maximum and minimum magnitudes.**
- **Error bars in other measured parameters indicate their uncertainty.**

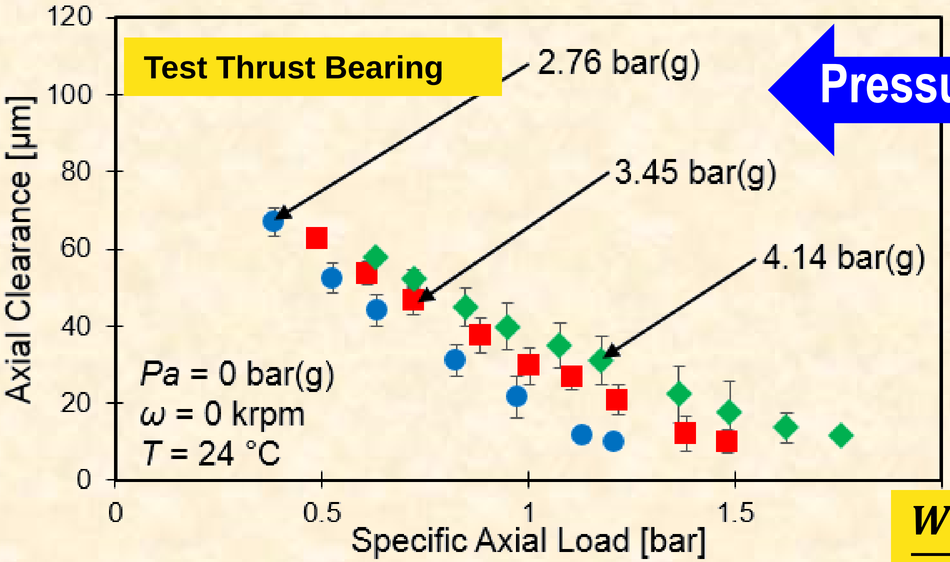
$$A = \frac{\pi}{4}\left(D_{out}^2 - D_{in}^2\right)$$

Bearing area = **32.6 cm²**



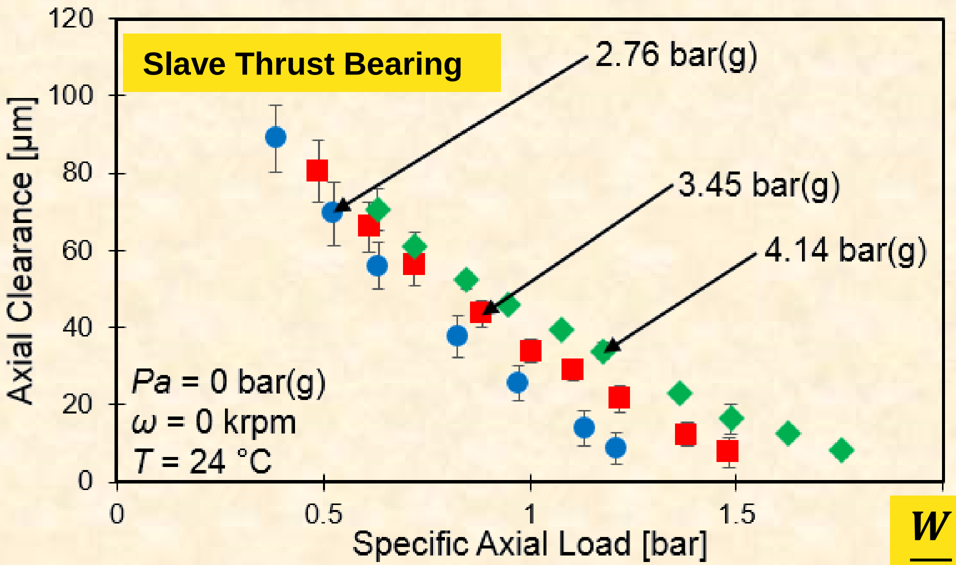
Tests without shaft speed

TBs Clearance vs. load (0 rpm)



$\frac{W}{A}$

Pressure supply varies



$\frac{W}{A}$

Findings

Axial clearance increases as water supply pressure increases.

Clearance decreases as applied load increases.

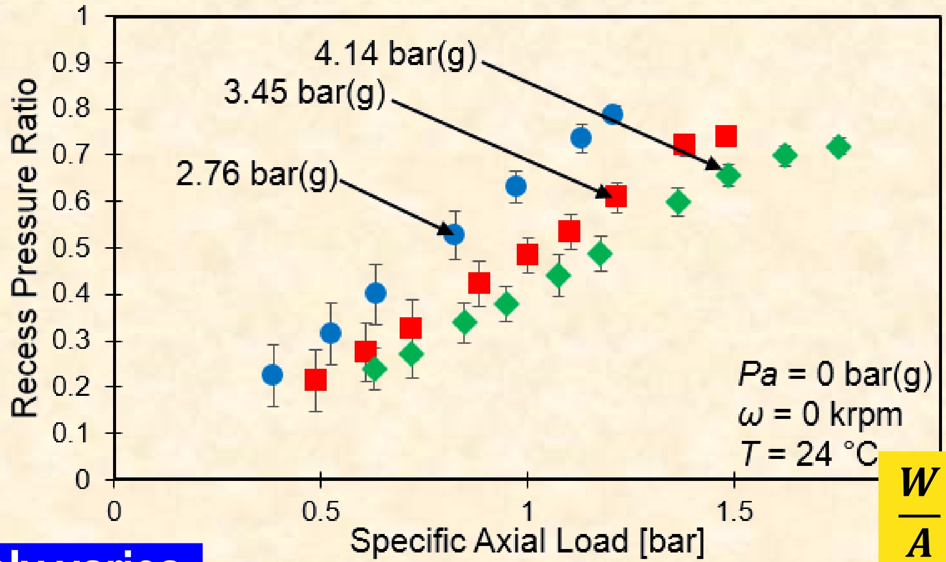
Load per unit area is only a fraction of water supply pressure.

Slave bearing operates with larger clearance because of larger orifice diameters.

TBs Pocket pressure vs clearance & load

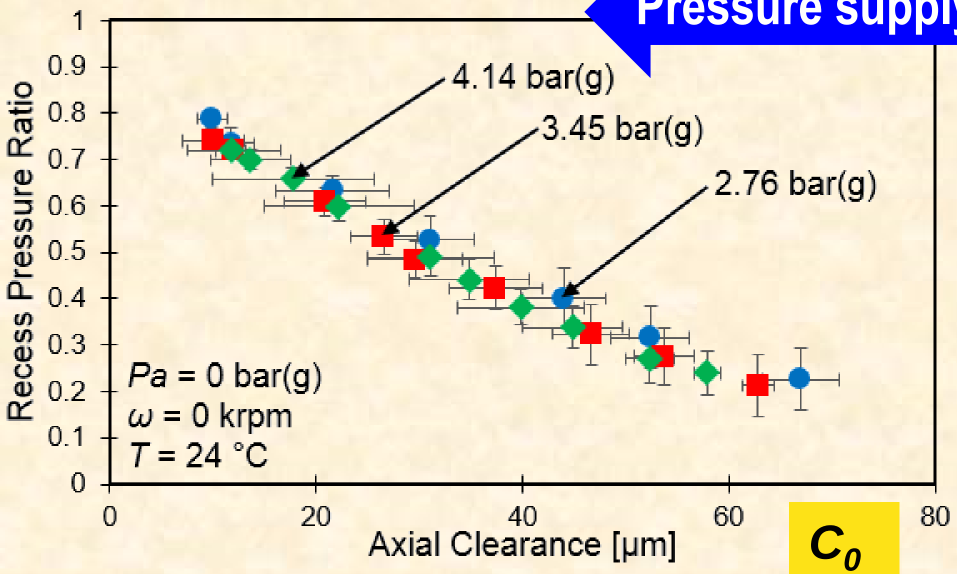
Test Thrust Bearing **0 rpm**

$\frac{P_R - P_a}{P_S - P_a}$ = Recess or pocket pressure ratio



$\frac{W}{A}$

Pressure supply varies



C_0

Findings:

Flow rate decreases as supply decreases and as clearance decreases (load increases).

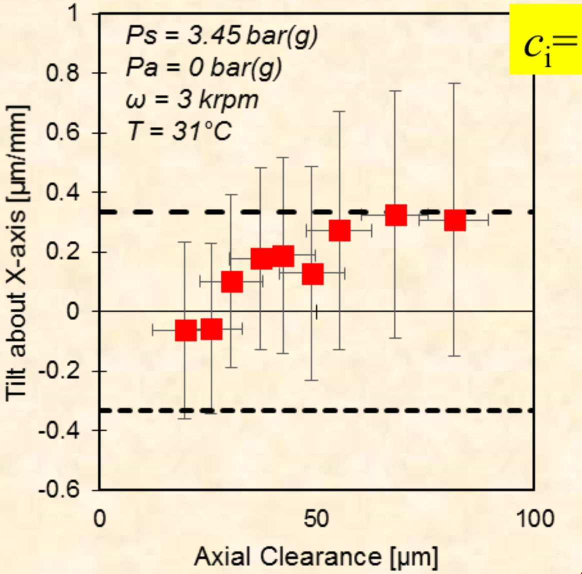
As flow rate decreases, pressure in a pocket increases, $P_R \rightarrow P_S$

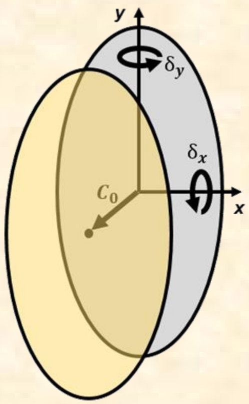
Tests with shaft speed

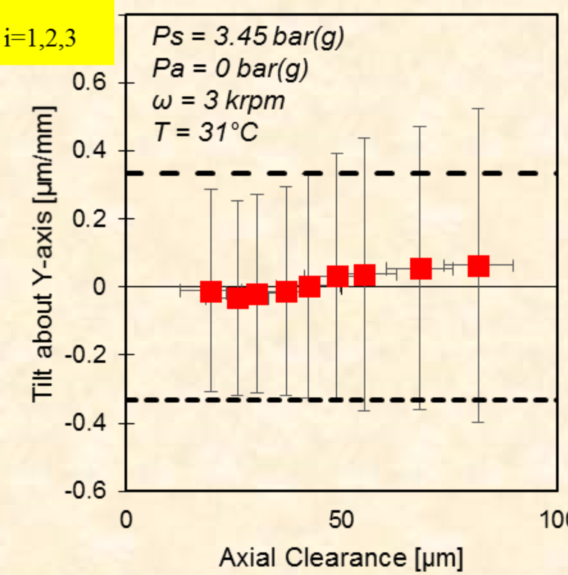
3 krpm (surface speed OD = 16 m/s)

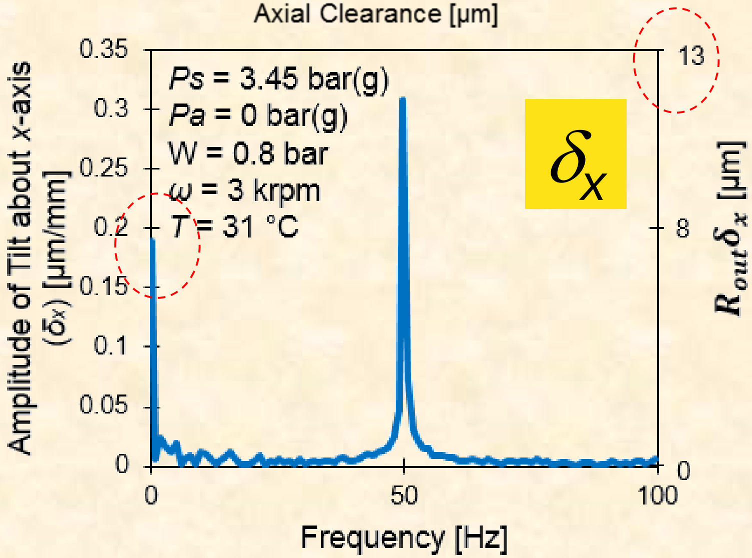
TB tilts (static & dynamic) at 3 krpm

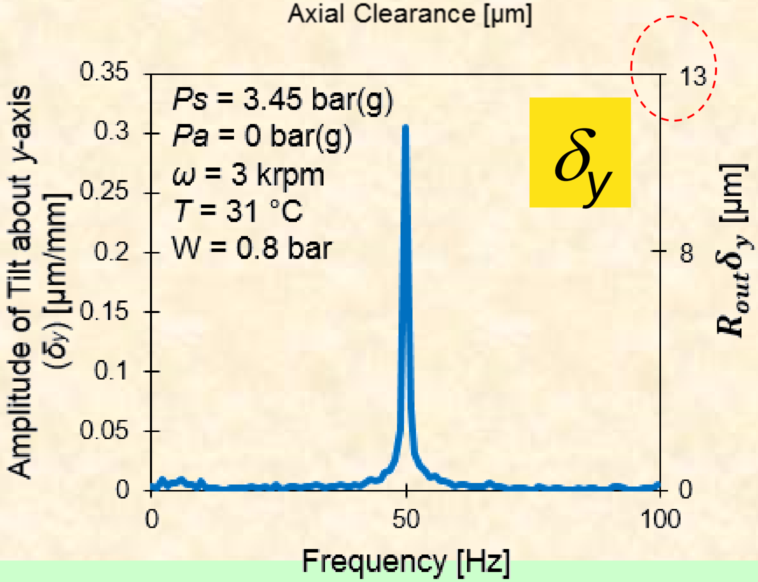
$c_i = c_o + \underline{R}\cos(\theta_i)\ \delta_Y + \underline{R}\sin(\theta_i)\ \delta_X$, i=1,2,3









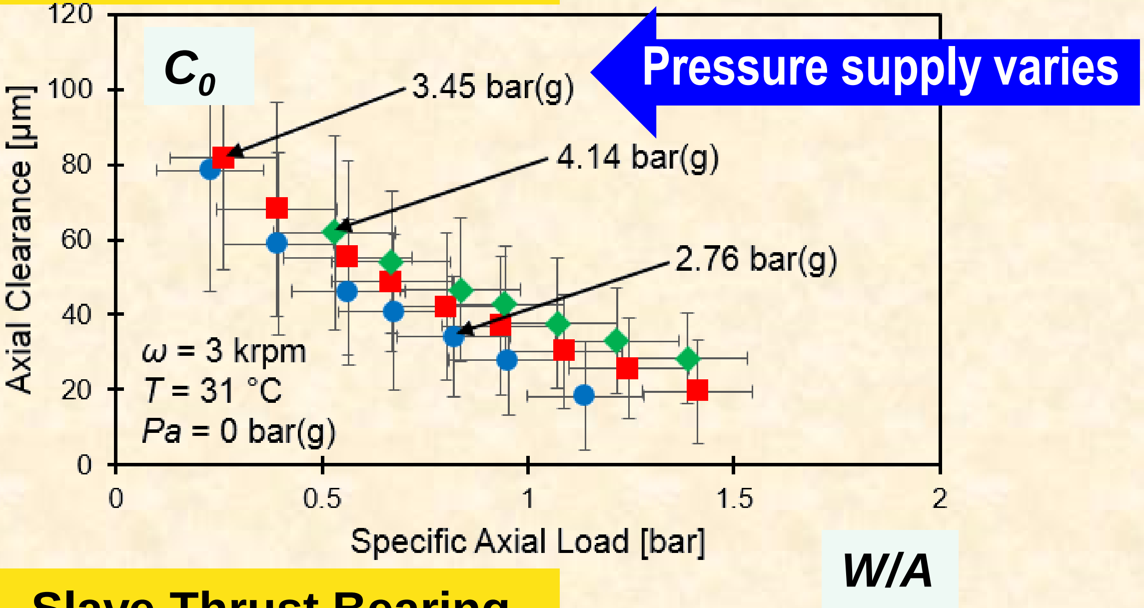


NOTE: Thrust collar tilts & wobbles with 1X frequency (50 Hz)

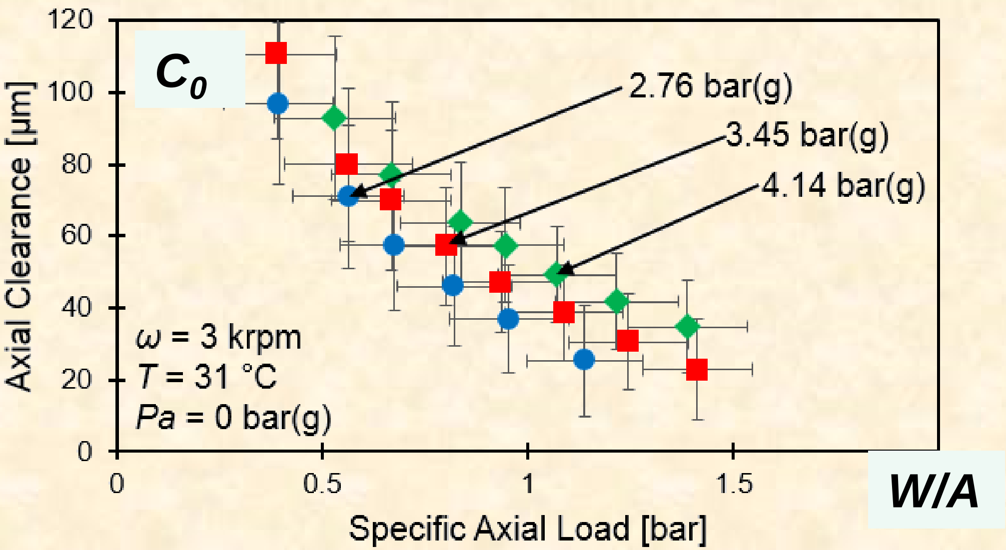
TBs axial clearance C_0 vs. load

3 krpm

Test Thrust Bearing



Slave Thrust Bearing



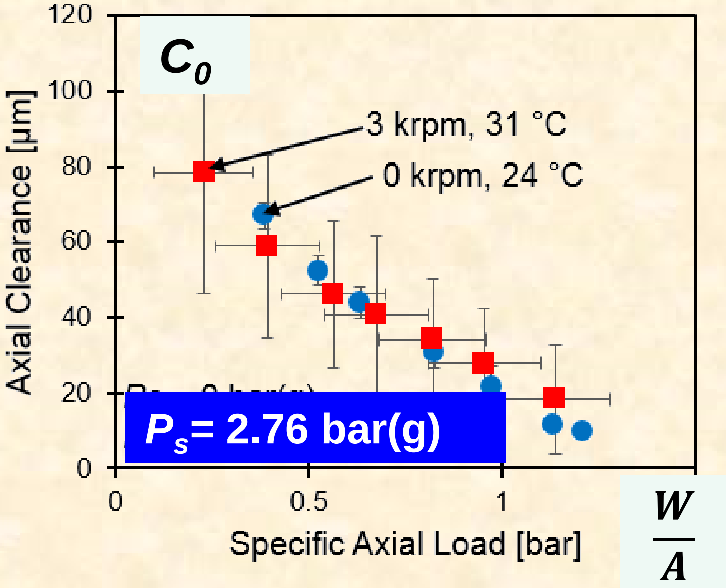
Findings

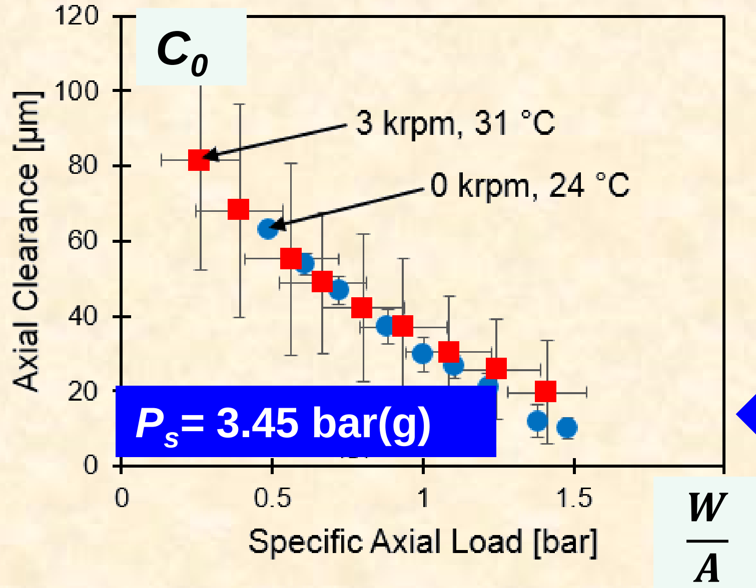
Clearance decreases as applied load increases and increases as water supply pressure increases.

Slave TB operates with a larger clearance than test TB because of its larger orifice diameter.

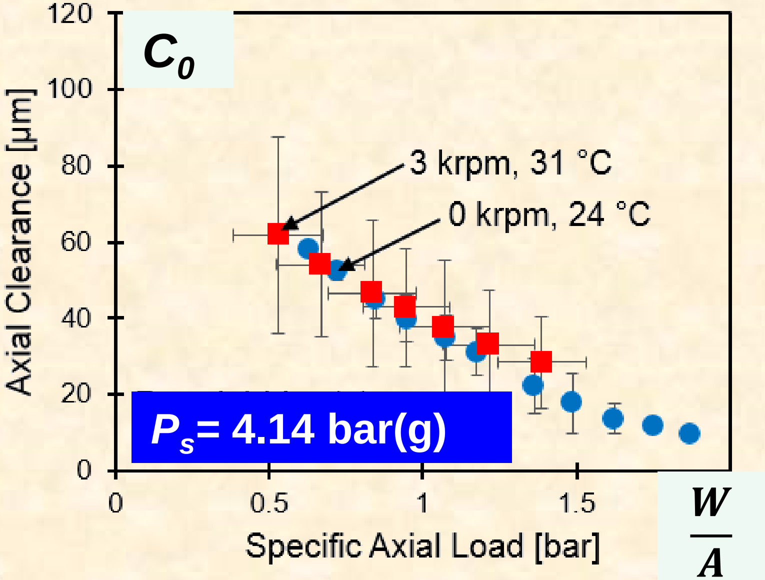
Note large *error bars* due to tilting of collar.

Compare TB performance: 0 krpm vs 3 krpm





Pressure supply varies

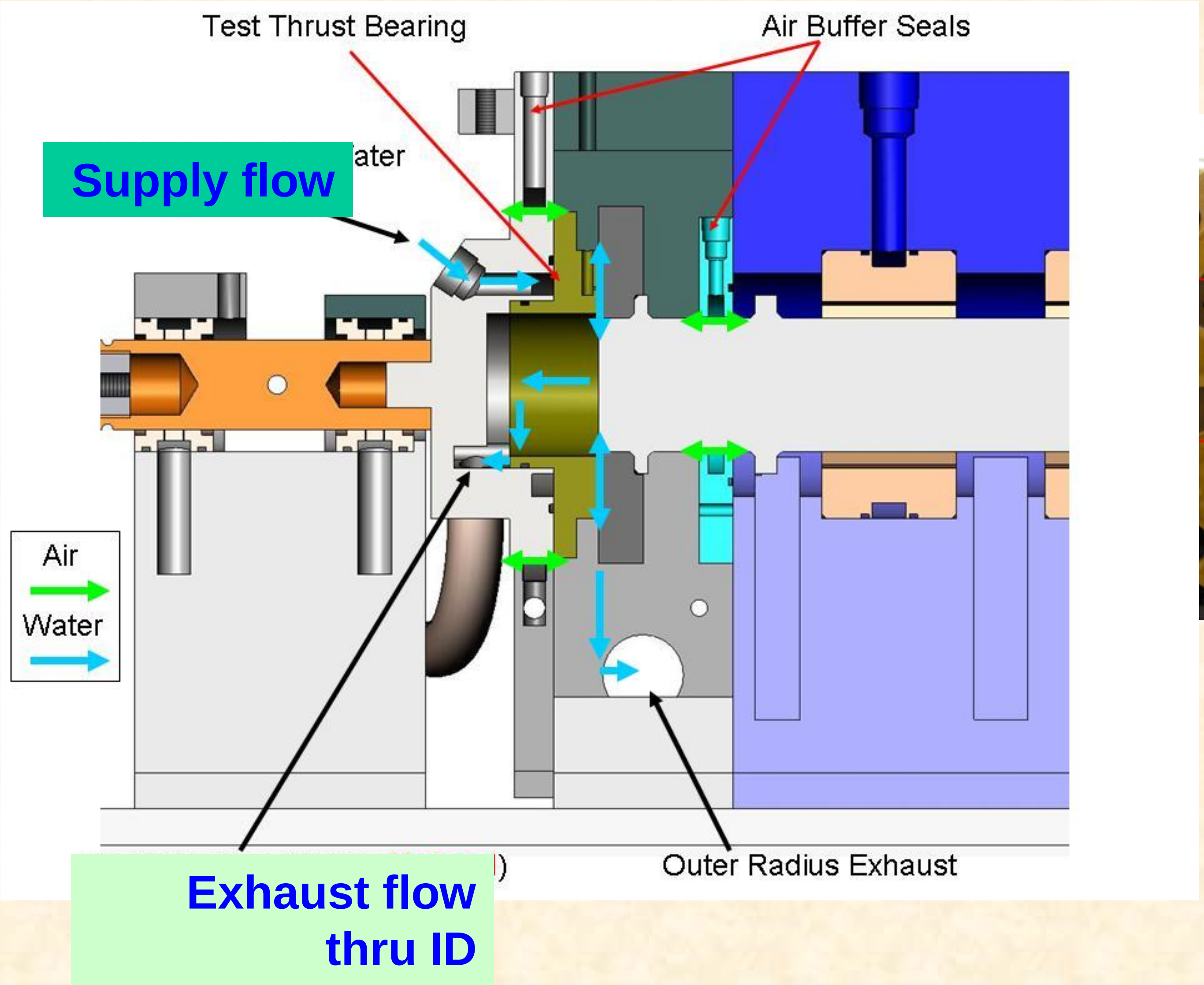


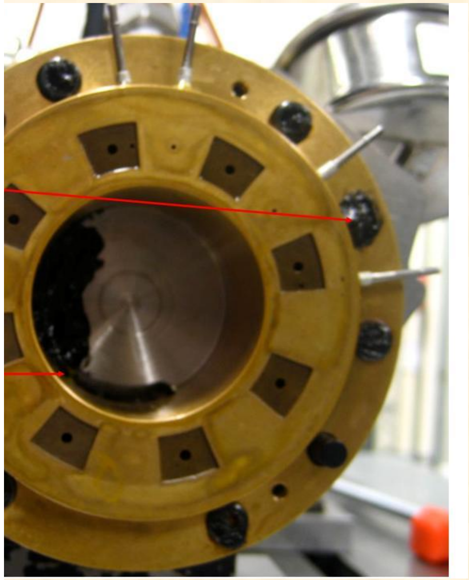
Findings

Shaft speed has not effect on axial clearance → thrust bearing operates mostly as a hydrostatic bearing.

Test Thrust Bearing

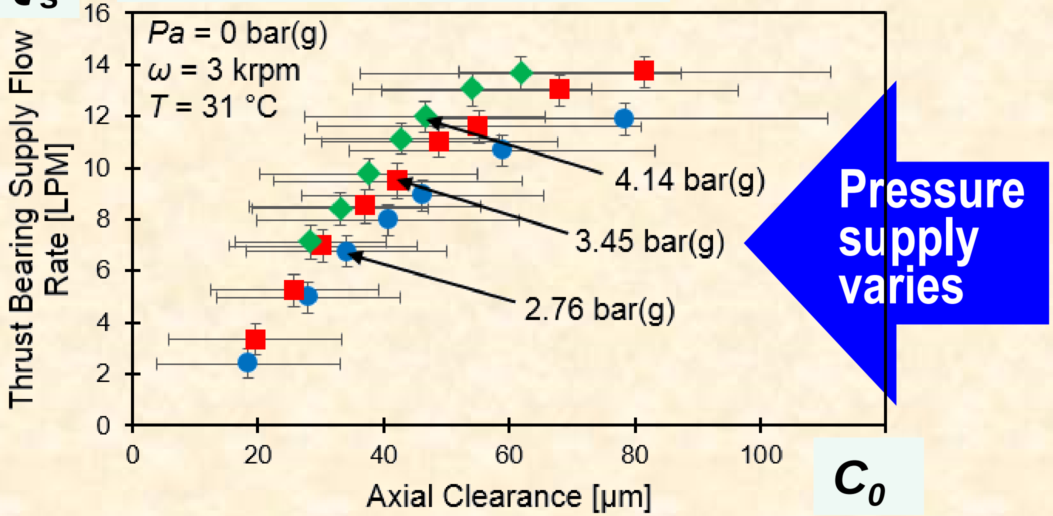
TB flow rates (supply and ID)





Test TB Bearing Flow rates vs axial clearance C_0

Q_s Supply flow



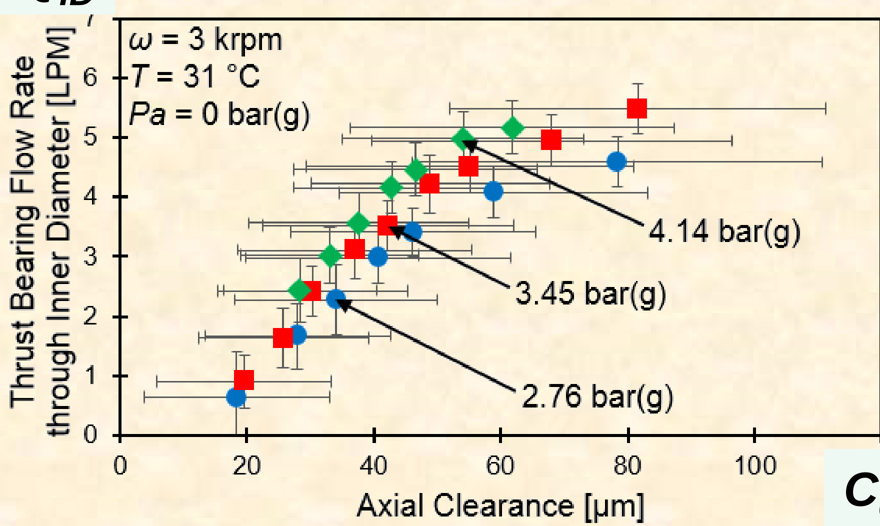
C_0

Findings

Flow rates increase as axial clearance increases (load decreases) and when supply pressure increases.

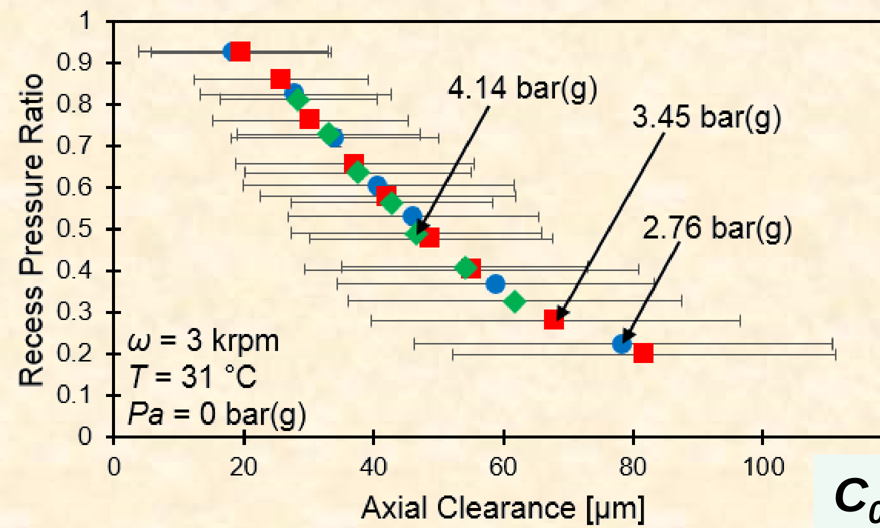
Recess pressure decreases as axial clearance increases.

Q_{ID} Outflow through bearing ID



C_0

$$\frac{P_R - P_a}{P_S - P_a} = \text{Recess Pressure Ratio}$$



C_0

Test TB ratio of flow rates (ID/supply)

Q_S = Supply Flow Rate

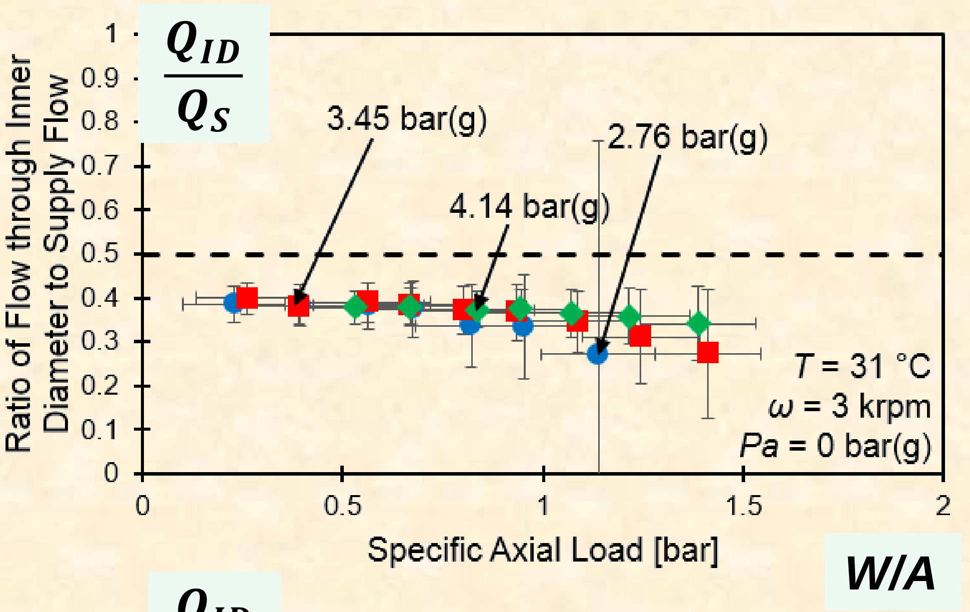
Q_{ID} = Flow Rate through Inner Diameter

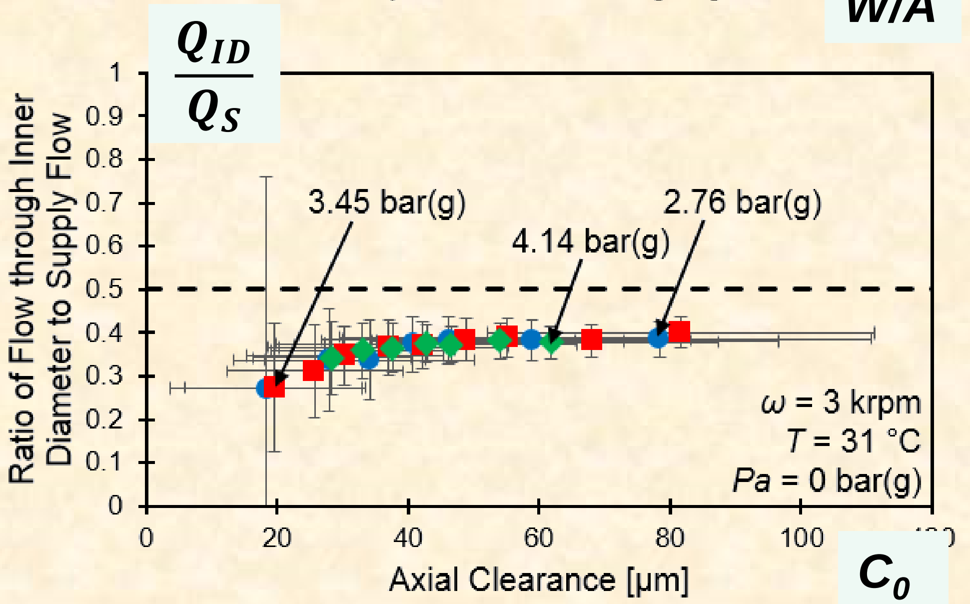
$\frac{Q_{ID}}{Q_S}$ = **Ratio of Flows**

Findings

Ratio of flows is fairly constant (40%→30%), decreasing as axial load increases (clearance decreases).

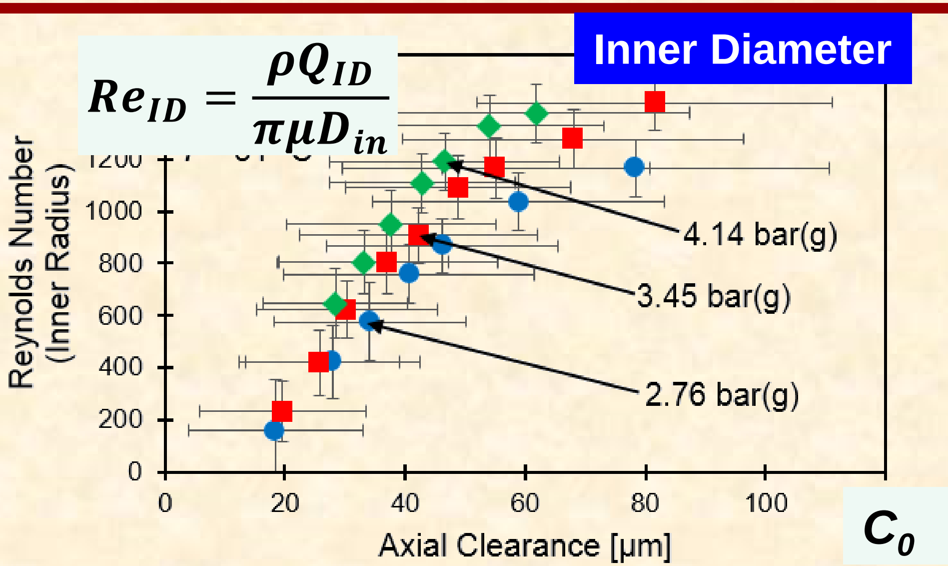
Shaft speed is too low too cause starvation of fluid bearing ID

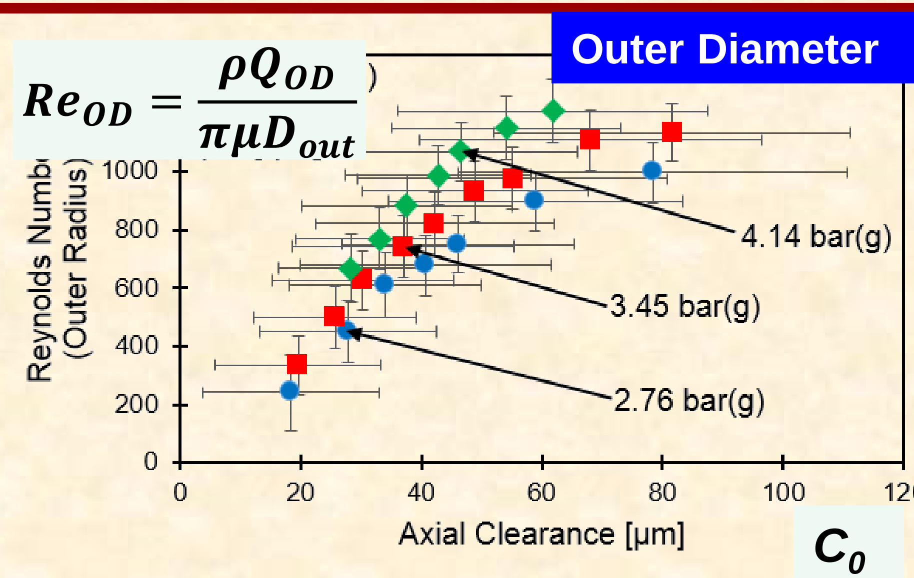




Test Thrust Bearing

Test TB Reynolds Numbers at 3 krpm



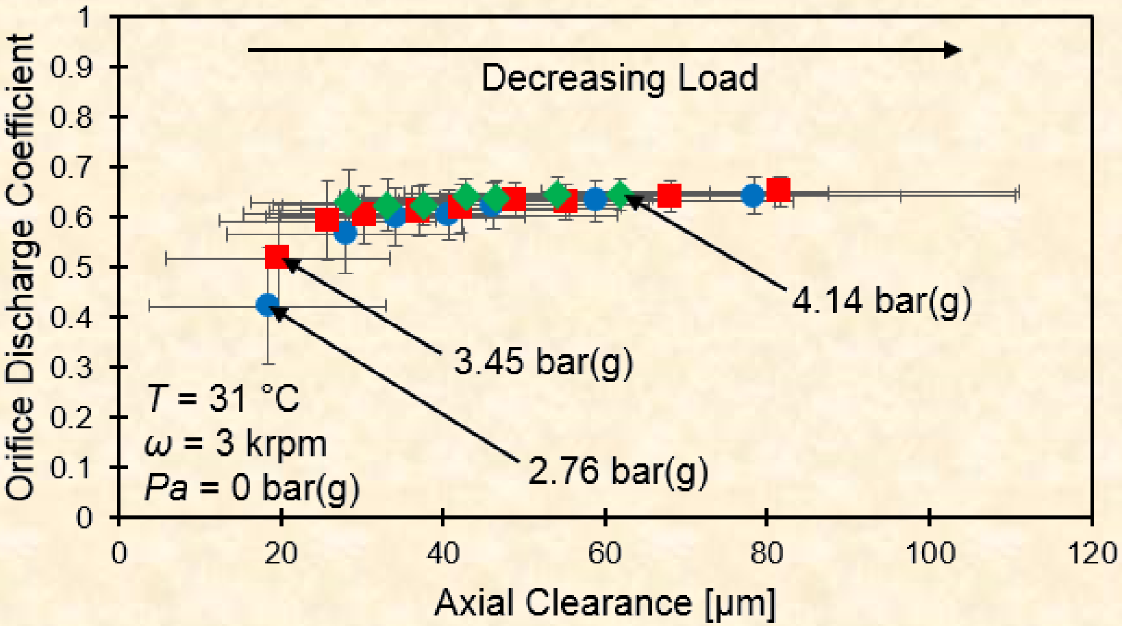


C_0	$Re_C = \frac{\rho}{\mu}\omega RC$ ID	OD
20 μm	160	300
80 μm	650	1220

Re_{ID} and Re_{OD} increase as supply pressure increases → more flow rate.

TB operates in two flow regimes: laminar → transition to turbulence.

Estimation of Orifice discharge coefficient



$$C_d = \frac{Q_O}{A_O\sqrt{\frac{2}{\varrho}(P_S - P_R)}}$$

C_d = Orifice Discharge Coefficient

Q_O = Flow Rate through Orifice

A_O = Area of Orifice

Supply Pressure (P_S)	C_d
2.76 bar(g)	**0.61** ± 0.07
3.45 bar(g)	**0.62** ± 0.05
4.14 bar(g)	**0.64** ± 0.02

Findings

C_d → ~0.62 at large clearance.

Important for prediction of TB performance

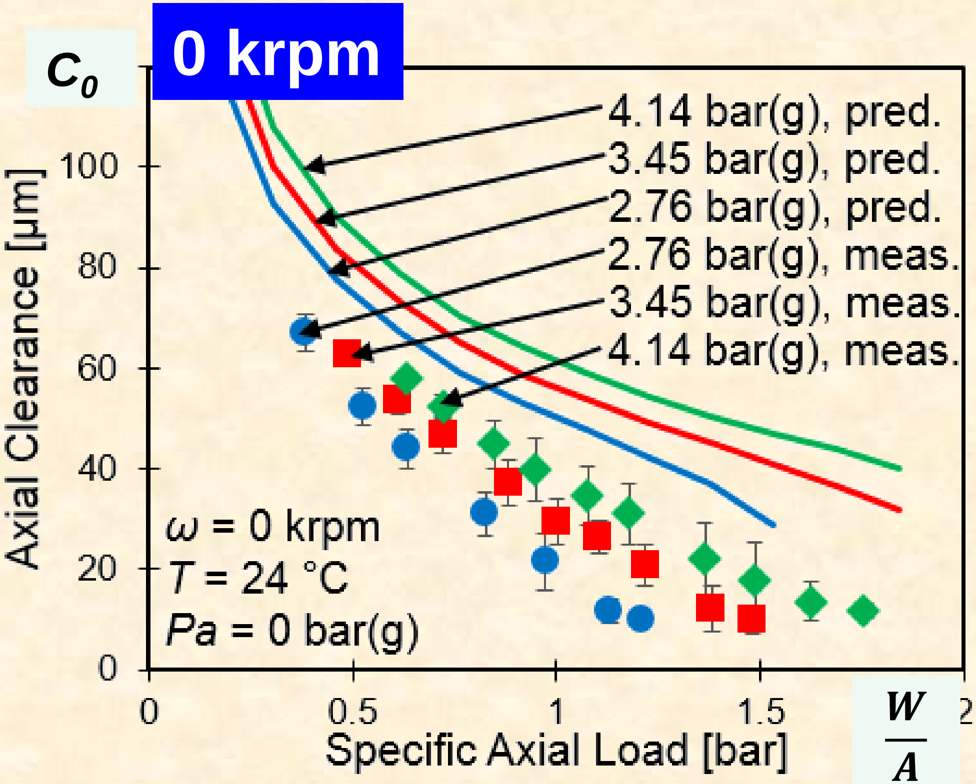
Test Thrust Bearing

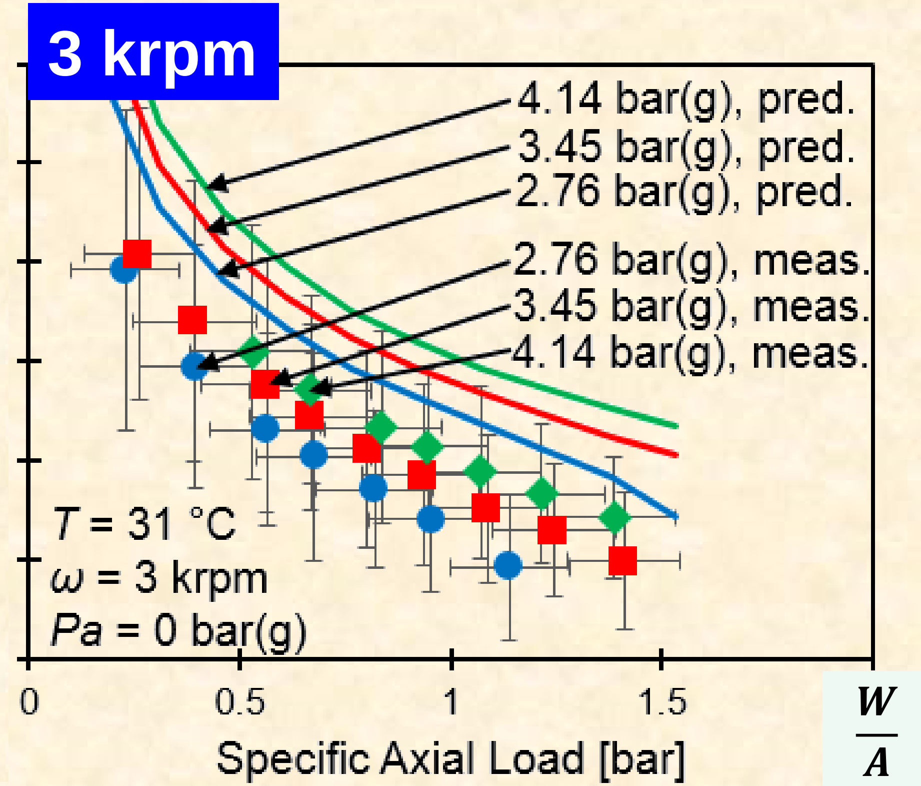
Hydrostatic/Hydrodynamic TB Model Validation

Tool XLHYDROTHRUST®

TB predictions vs. test data

Test Thrust Bearing





Findings

Similar results for slave TB.

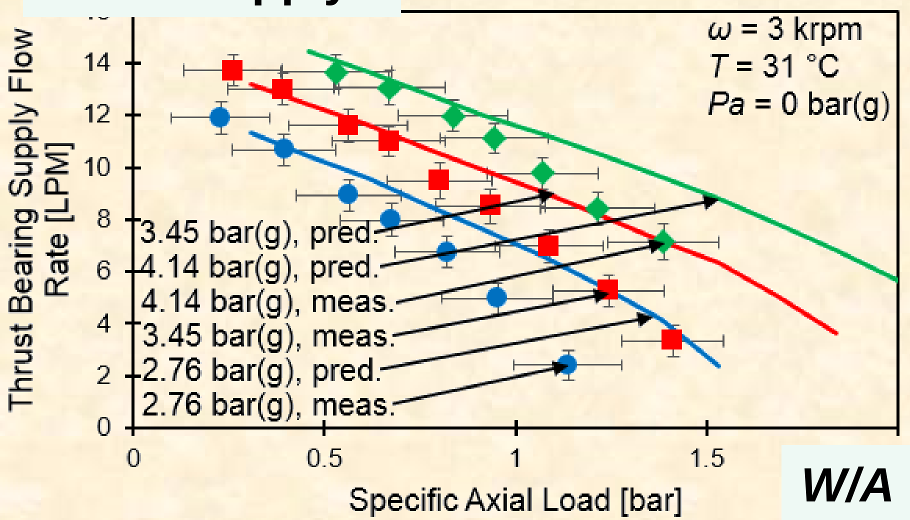
Predicted clearance is larger than test clearance. Worse correlation for highest load and operation with rotor speed.

GT2016-56349 shows better correlation test vs. prediction for high speed & high pressure TB.

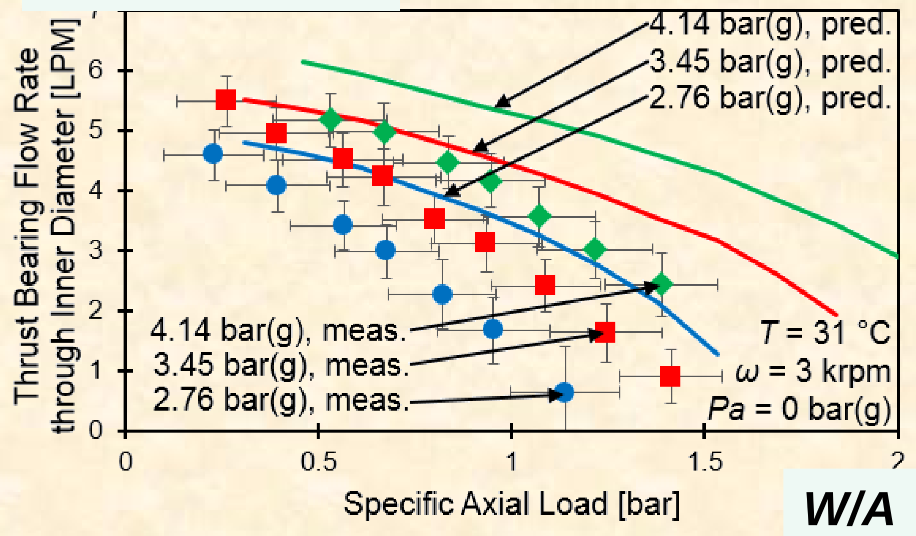
Test TB predictions vs. measurements

3 krpm

Flow supply

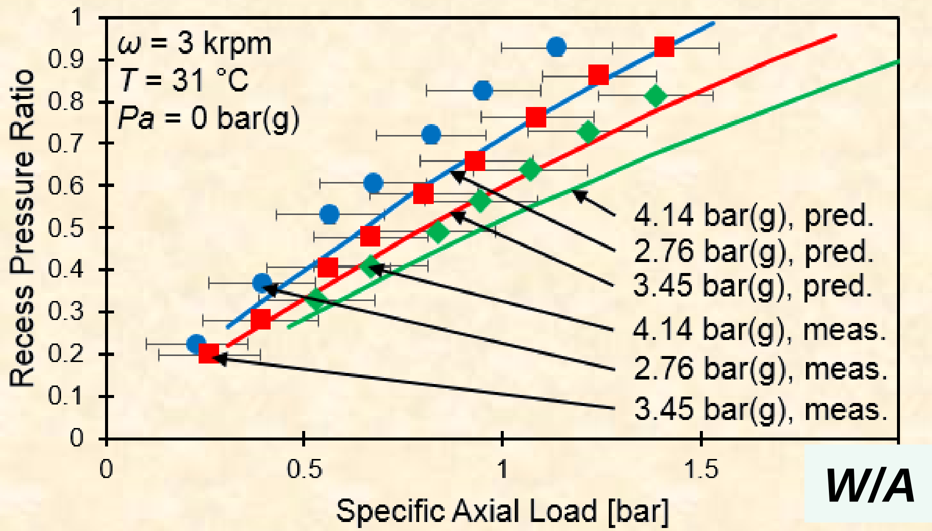


Flow thru ID



$$\frac{P_R - P_a}{P_S - P_a}$$

Recess pressure ratio



Findings

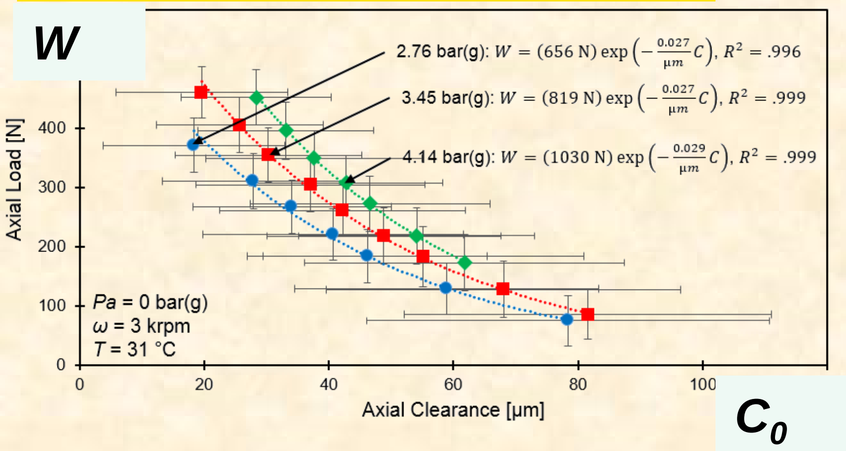
Predictions agree with test data for recess pressure and flow rate at low load (large clearance).

Predictions are poor for large loads (likely due to flow transition).

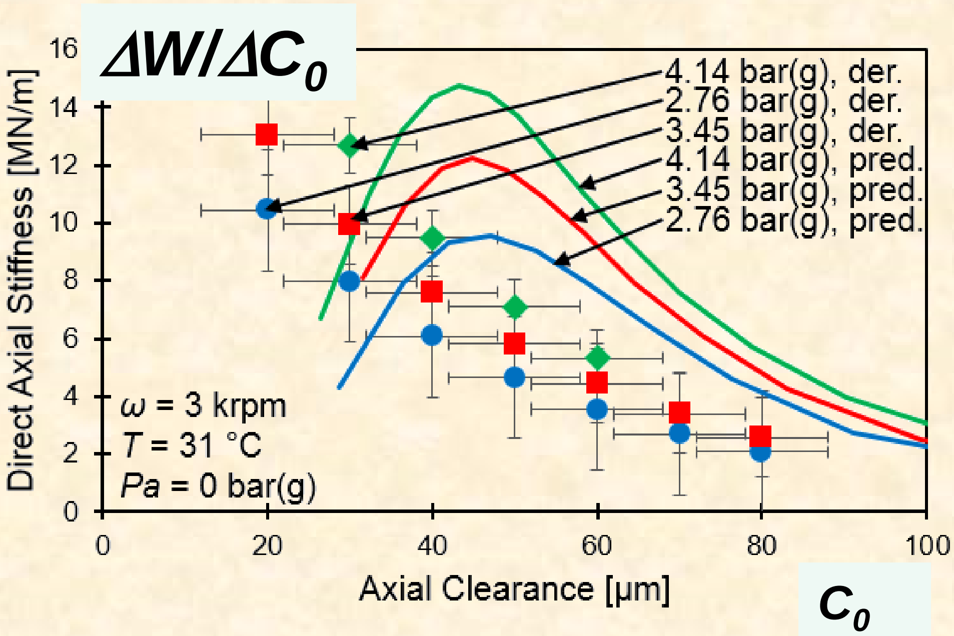
Test TB derived static (axial) stiffness

3 krpm

Exp. load vs Clearance



Estimated axial stiffness



Findings

Test derived stiffness *(K)* is of same magnitude as predicted one. Prediction delivers a *harder K* than test.

Conclusion

STATIC LOAD PERFORMANCE OF A WATER LUBRICATED HYDROSTATIC THRUST BEARING

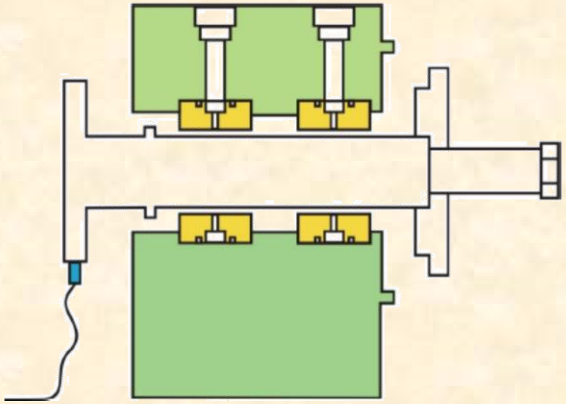
Conclusion

- TEST RIG operation obscured by severe thrust collar and bearing misalignment.
- Axial clearance and flow rate increase as water supply pressure increases and as the axial load decreases.
- Predictions accurate on the influence of applied load and supply pressure on the thrust bearing performance.
- Predictions of TB performance are poor for operation with large load (low clearance). Variation in tilts (misalignment) and flow regime operation may explain differences.
- A higher supply pressure into the radial bearings could mitigate the misalignment of rotor and thrust collar.

Acknowledgments

GT2017-63385

Thanks to Turbomachinery Research Consortium



Questions (?)

Learn more at **http://rotorlab.tamu.edu**

Past Work with Same Test Rig

TAMU M.S. Thesis

Forsberg (2006)

Designs and builds a test rig to test water-lubricated hybrid thrust bearings and performs tests without rotor speed. Water at high pressure (1.72 MPa) supplies thrust bearing. Flow rates through inner and outer diameter are different, which could cause fluid starvation.

TAMU M.S. Thesis

Ramirez (2008)

Performs tests on water-lubricated hybrid thrust bearing for operation with high supply pressure (1.72 Mpa) and high rotor speed (17.5 krpm). Rotor speed does not have a large effect on the thrust bearing performance. Flow measurements show onset of starvation at inner diameter.

TAMU M.S. Thesis

Esser (2010)

Performs tests on water-lubricated thrust bearings with different orifice diameters for operation with high supply pressure (1.72 Mpa) and high rotor speed (17.5 krpm). Larger orifice diameters mitigate fluid starvation at the inner side. Larger orifices provide larger clearance at the cost of a larger flow rate.

Measurements of thrust bearing static load performance correlate well with predictions in each work.

San Andres, 2002, ASME J. of Trib., 124(1)

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