Proceedings of ASME Turbo Expo 2016: Turbine Technical Conference and Exposition, June 13-17, 2016, Seoul, South Korea

Paper GT2016-56349

A Water Lubricated Hybrid Thrust Bearing: Measurements and predictions of Static Load Performance

Luis San Andrés
Mast-Childs Chair
Professor

Stephen Phillips
Research Engineer

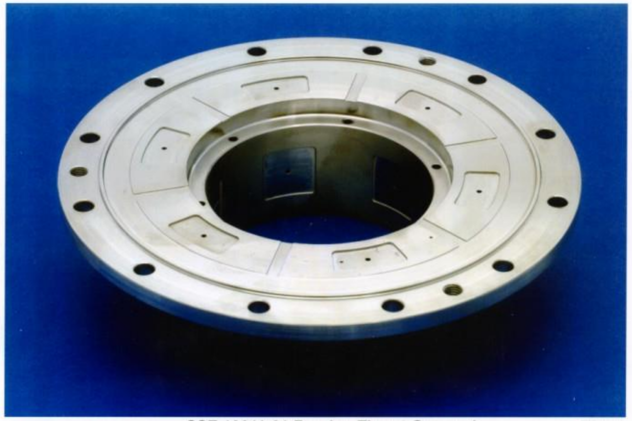
Dara Childs
L.T. Jordan Chair
Professor

TURBOMACHINERY LABORATORY
TEXAS A&M ENGINEERING EXPERIMENT STATION

Texas A&M University
College Station, TX 77843-3123

Supported by USRL Upper Stage Technology Program

Hybrid bearings for cryogenic turbopumps



Low cost primary power cryogenic turbo-pumps (TP) are compact, operate at high speeds, and require of externally pressurized fluid film bearings to support radial and thrust loads.

Hybrid thrust & radial bearings enable smaller and lighter turbopumps with no DN life limitations

Large stiffness (accuracy of positioning) and damping force coefficients allow for unshrouded impellers with increased TP efficiency

Hybrid Bearings: Model Validation

Radial hybrid bearings: Tool XLHYDROJET®

Tests at TAMU (1992-1996) with water (1000 psi (70 bar) max, 25 krpm max).
* +20 bearings x 3 clearances & 2 pocket depths, different pocket shapes, macro-roughness (surface textured) bearings, angled injection.

Gas Honeycomb seals

Water Lomakin Bearings (Snecma-SEP, 2000-2002)

Oil tilting and flexure pivot journal bearings (TRC, 2002 – 2015)

Thrust hybrid bearings: Tool XLHYDROTHRUST®

Until 2008: NONE available for high speed, high pressure (turbulent flow) bearings

Concerns: centrifugal and advection fluid inertia cause severe fluid starvation in bearing

USET program OBJECTIVE

USET: Upper Stage Technology Program (AFRL)

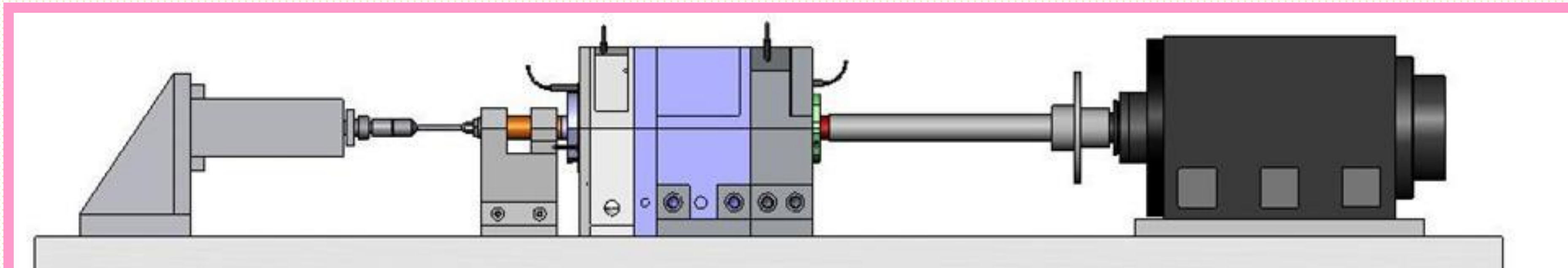
Objective USET (Upper Stage Technology Program)

To verify that predictive methods and tools used to design and manufacture cryogenic turbopumps are valid and accurate.

Tasks (Outline)

- **Design & construction of thrust bearing test rig.**
- **Operation and troubleshooting of test apparatus.**
- **Measurements of axial clearance, load, pocket pressures, and flow rates in a water hybrid thrust bearing.**
- **Prediction of performance from tool XLHYDROTHRUST and comparisons to test data.**

Description of test rig



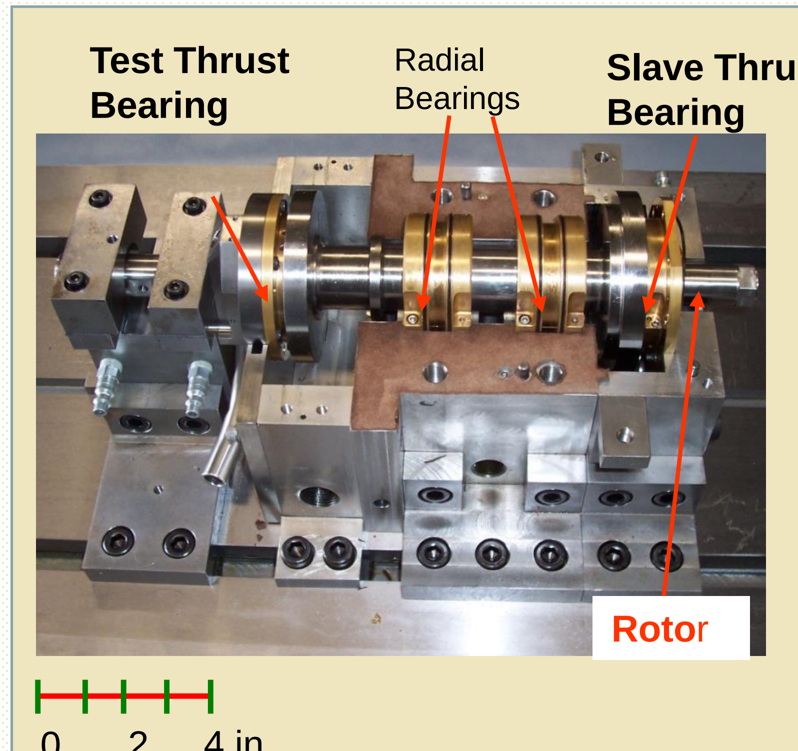
Shaker load and stinger

Axial load shaft & test thrust bearing

Rotor & Radial bearings

Coupling

Drive motor



Test Rig Features

Test Fluid: **WATER**

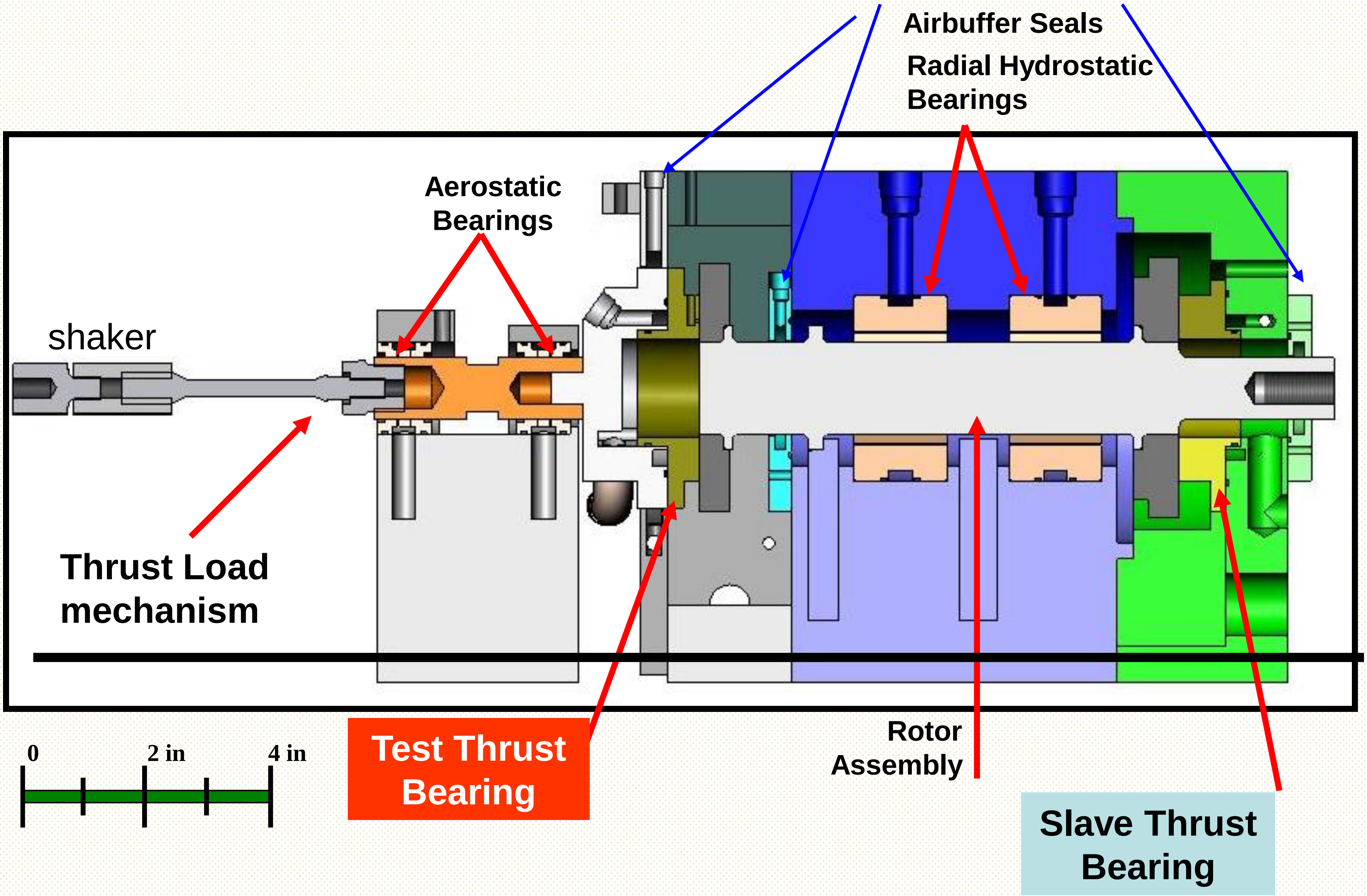
0-25 krpm,

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

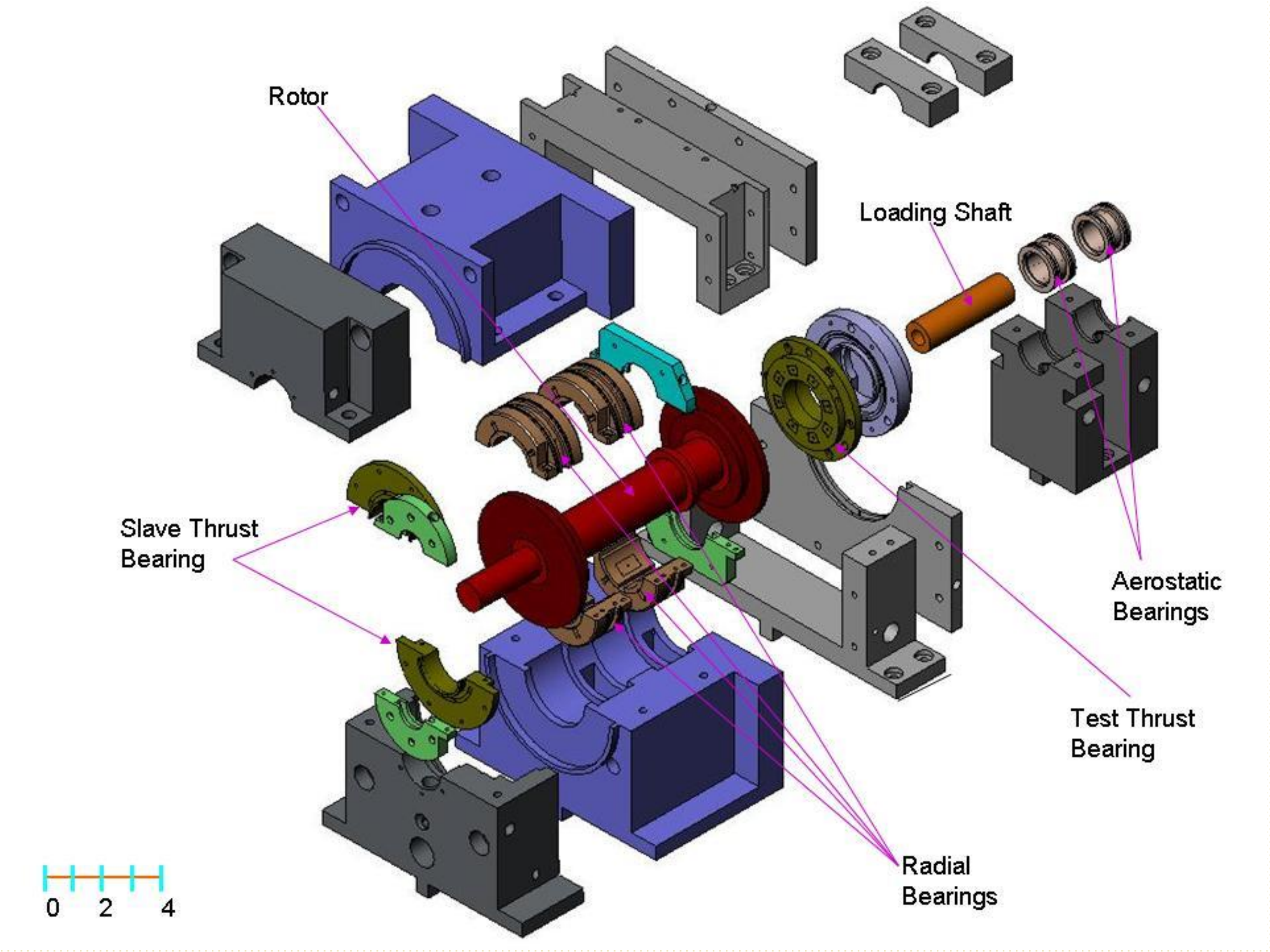
Range of static + dynamic axial load: 1000 lbf,

frequency range: 0-600 Hz

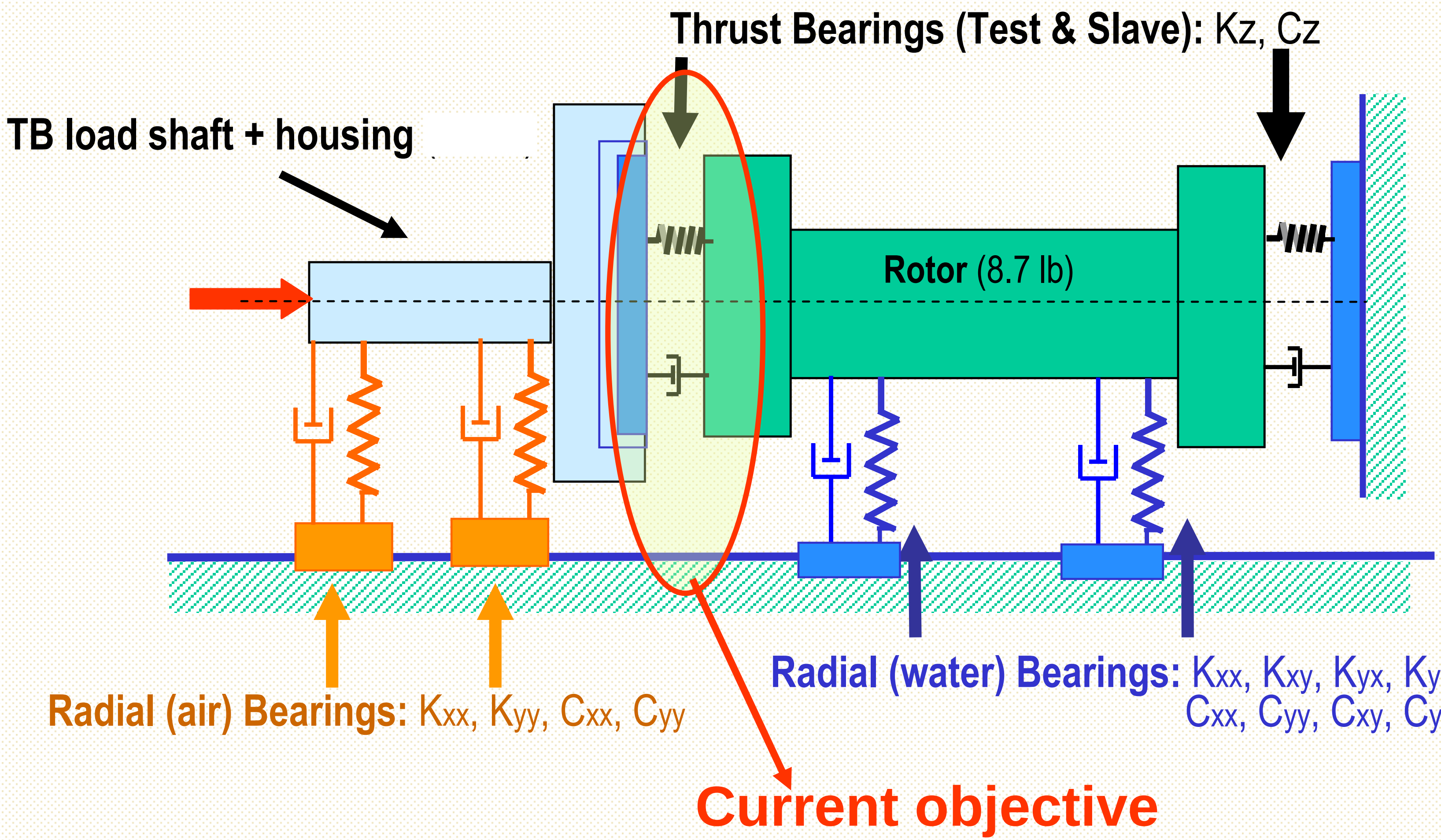
Hybrid Thrust Bearing Rig – Cross Section



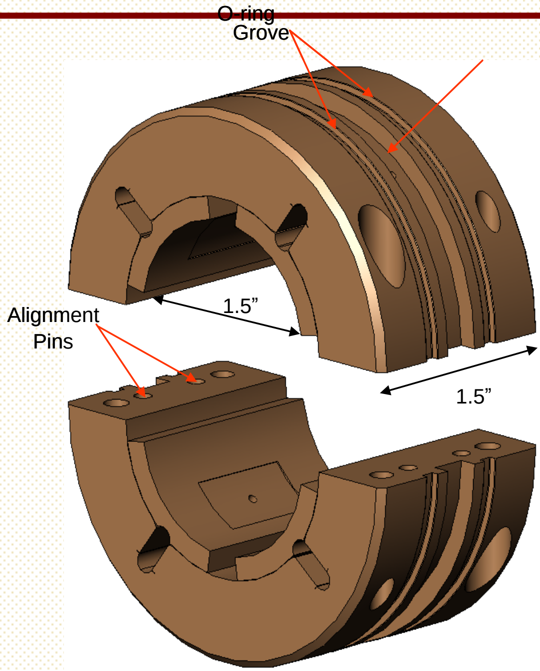
Hybrid Thrust Bearing Test Rig – Exploded View



Schematic representation of test rig: thrust and radial bearings as mechanical elements with stiffness and damping coeffs.

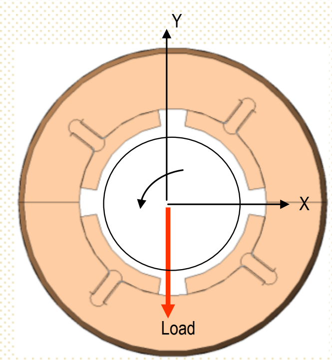


Radial Support: Flexure Pivot Tilting Pad Hybrid Bearings

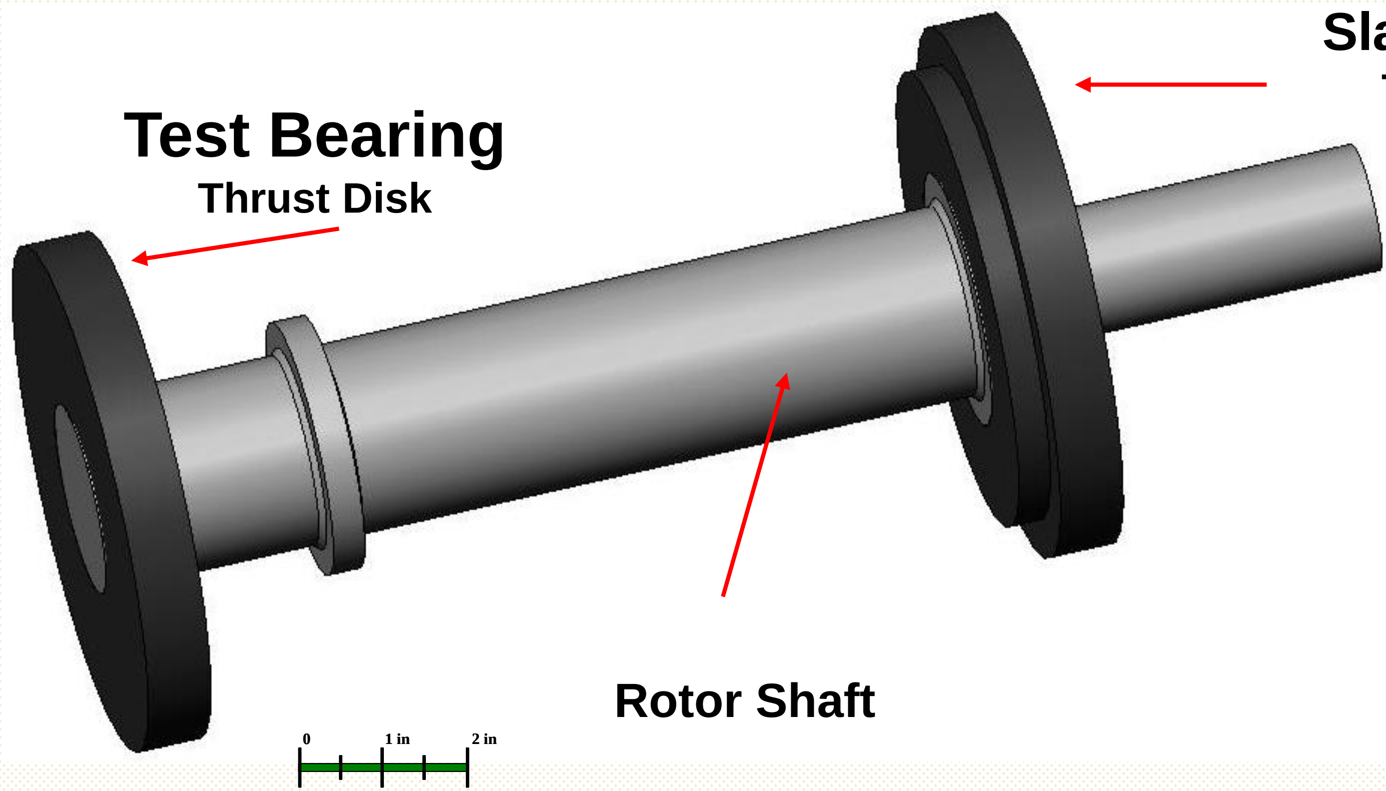


Nominal	**Radial clearance**	**3 mil**
Bearing	**Inner Diameter**	**1.5 inch**
	Outer Diameter	**3.0 inch**
	Length	**1.5 inch**
Pads	**Number of pads**	**4**
	Arc length	**72 °**
	Pivot offset (dim)	**60% arc length**
	Preload (dim)	**0.20**
	Flexure stiffness	**1,770 lb_f-in/rad**
Pockets	**Axial length**	**0.50 inch**
one per pad	**Arc length**	**24 °**
	Depth	**20 mil**
	Mean Diameter	**2.16 inch**
	Pocket/wetted area ratio	**0.11**
	Inlet coefficient from pocket to land	**0.20**
Orifices	**Diameter**	**0.067 inch (1.7mm)**
one per pocket	**radial injection**	**50% of pocket length**

Material: 330
Bearing Bronze
Wire-EDM manufacturing
Modeled using XLHYDROJET



Test Rotor: shaft & thrust collar disks



Materials

Shaft: 304 SS

Disks:718 Inconel

Slave Thrust Disk:

Width: 0.75 inch

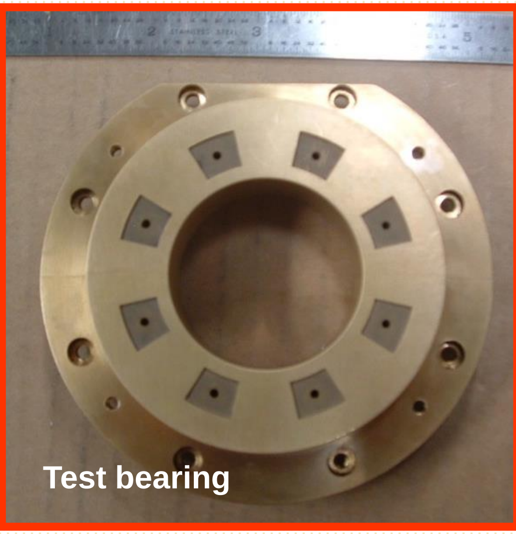
OD: 4.20 inch

Test Thrust Disk:

Width: 0.50 inch

OD: 4.00 inch

Thrust Hybrid 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:

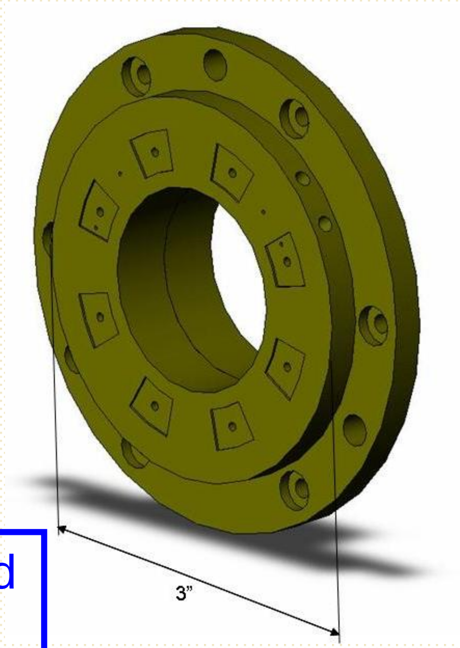
Mean Diameter:	2.16 inch
radial length:	0.32 inch
Arc length:	20 degrees
Depth:	0.020 inch

Pocket/wetted area ratio = 19%

Orifice size: 0.071 inch

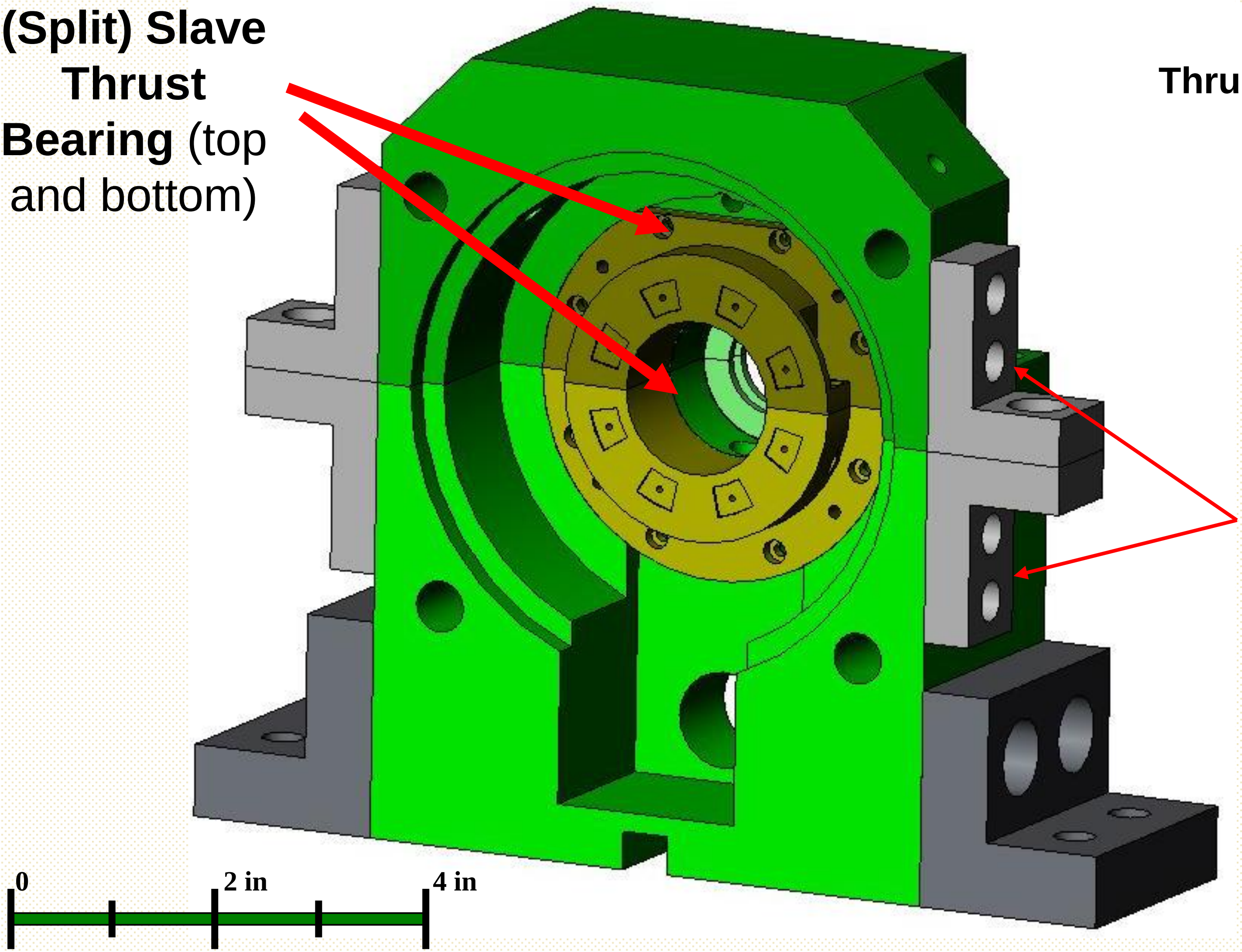
Axial injection at r=1.08 inch

Orifice discharge coefficients determined empirically from test data (~0.60)



Slave Thrust Bearing Housing : Assembly

(Split) Slave Thrust Bearing (top and bottom)



Materials

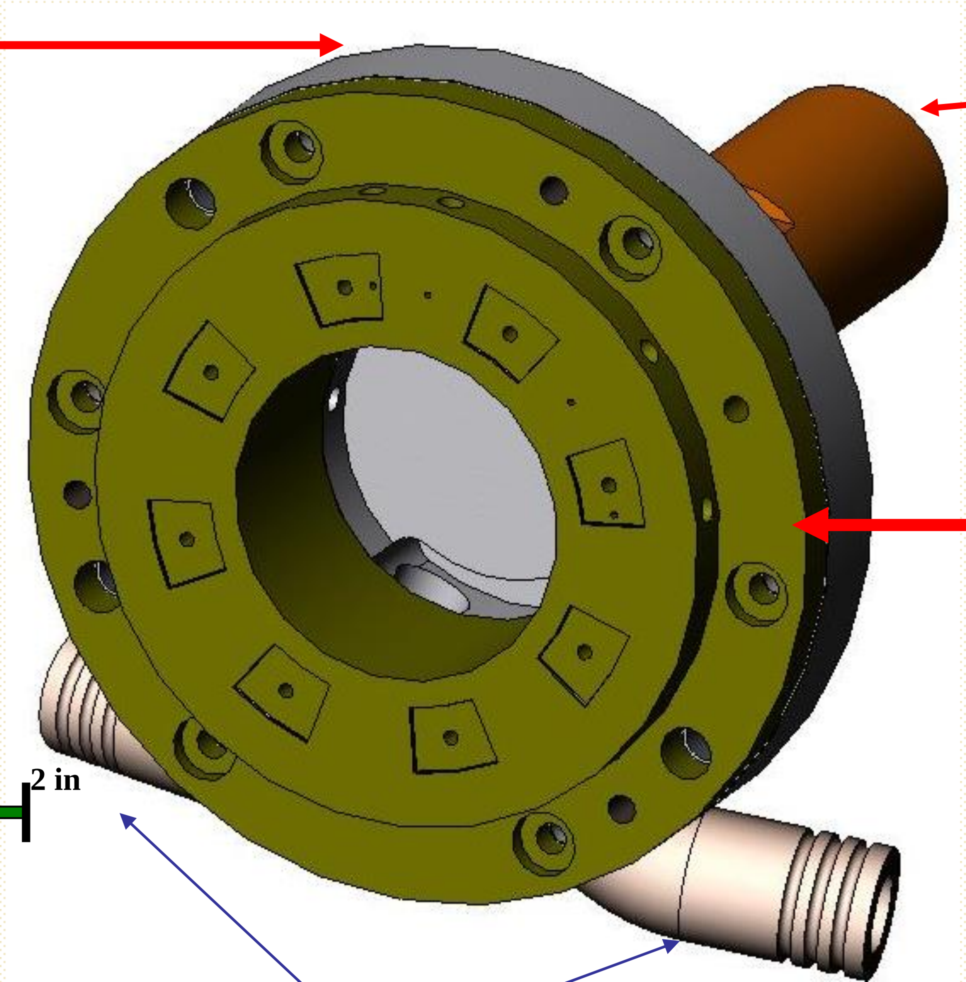
Thrust bearing: 660 Bearing Bronze

Housing: 304 SS

Brackets: AISI 4140

Housing (top and bottom)

TEST THRUST BEARING and load support

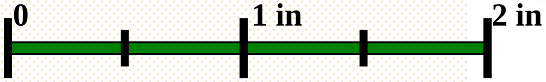


<u>**Materials**</u>

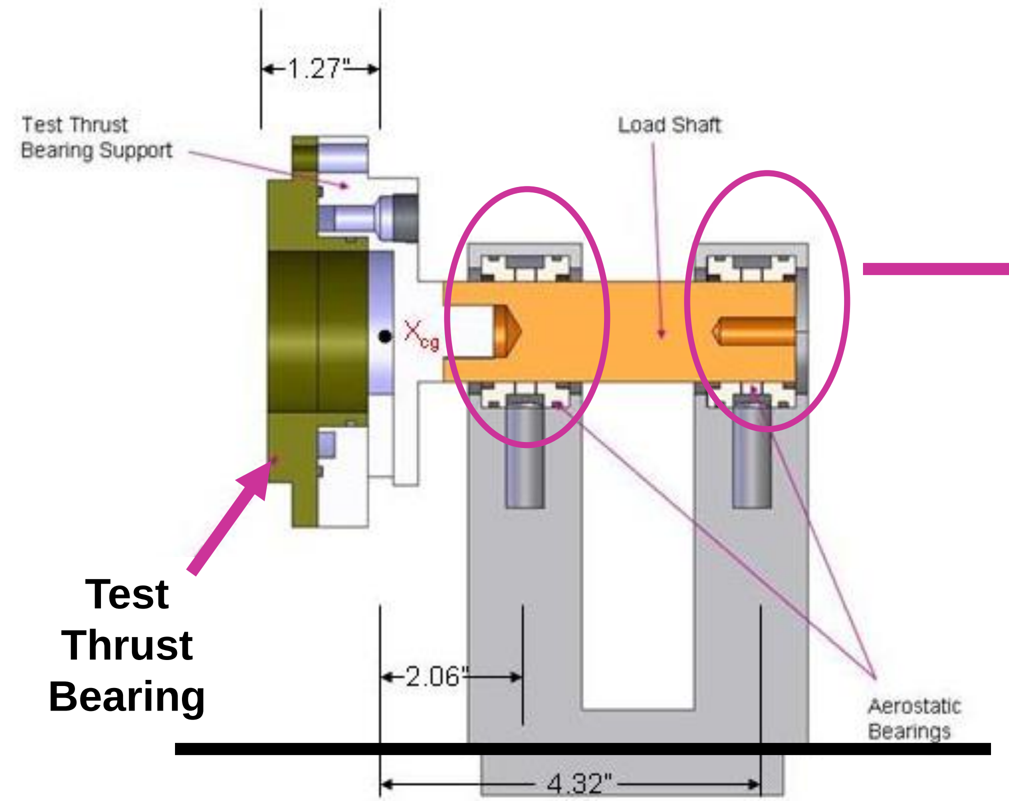
Thrust bearing: 660 Bearing Bronze

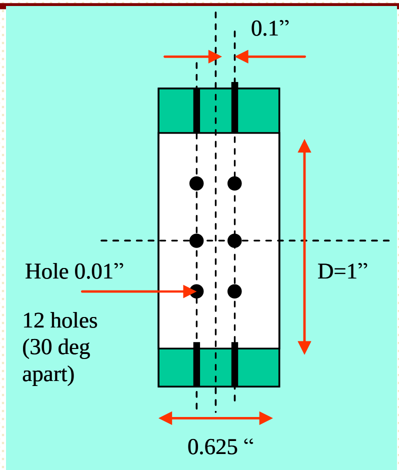
Holder: 6262-T9 Marine Aluminum

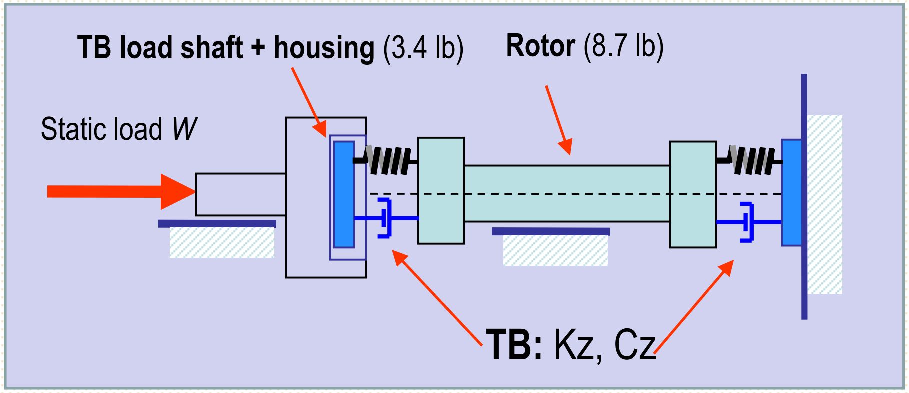
Load Shaft: 718 Inconel



Gas Bearings Support Axial Load Shaft







Gas Bearings:

Diameter: 1.00 inch

Radial clearance: 0.50 mil

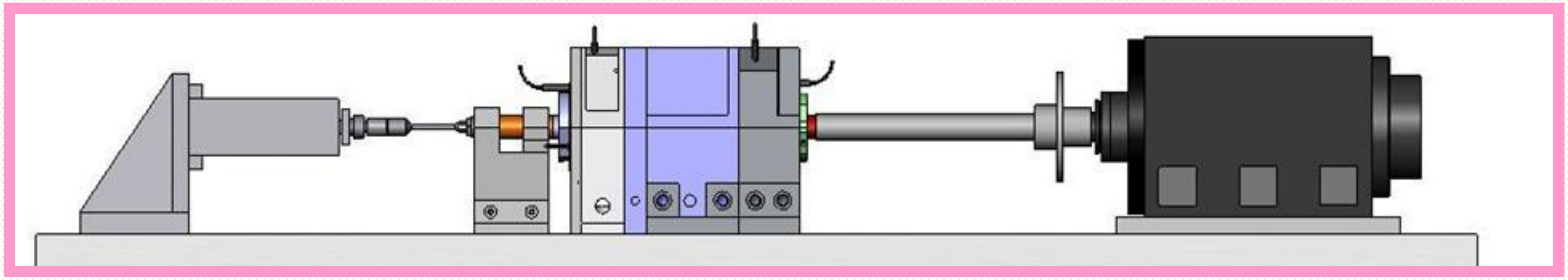
Length: 0.625 inch

2 rows x 6 orifices (60 deg apart)

Orifice size: 0.010 inch

660 Bronze Bearing

Test Rig Operation



STATIC LOAD TESTS

Thrust and radial bearings lubricated with water at 91F (~32 C)

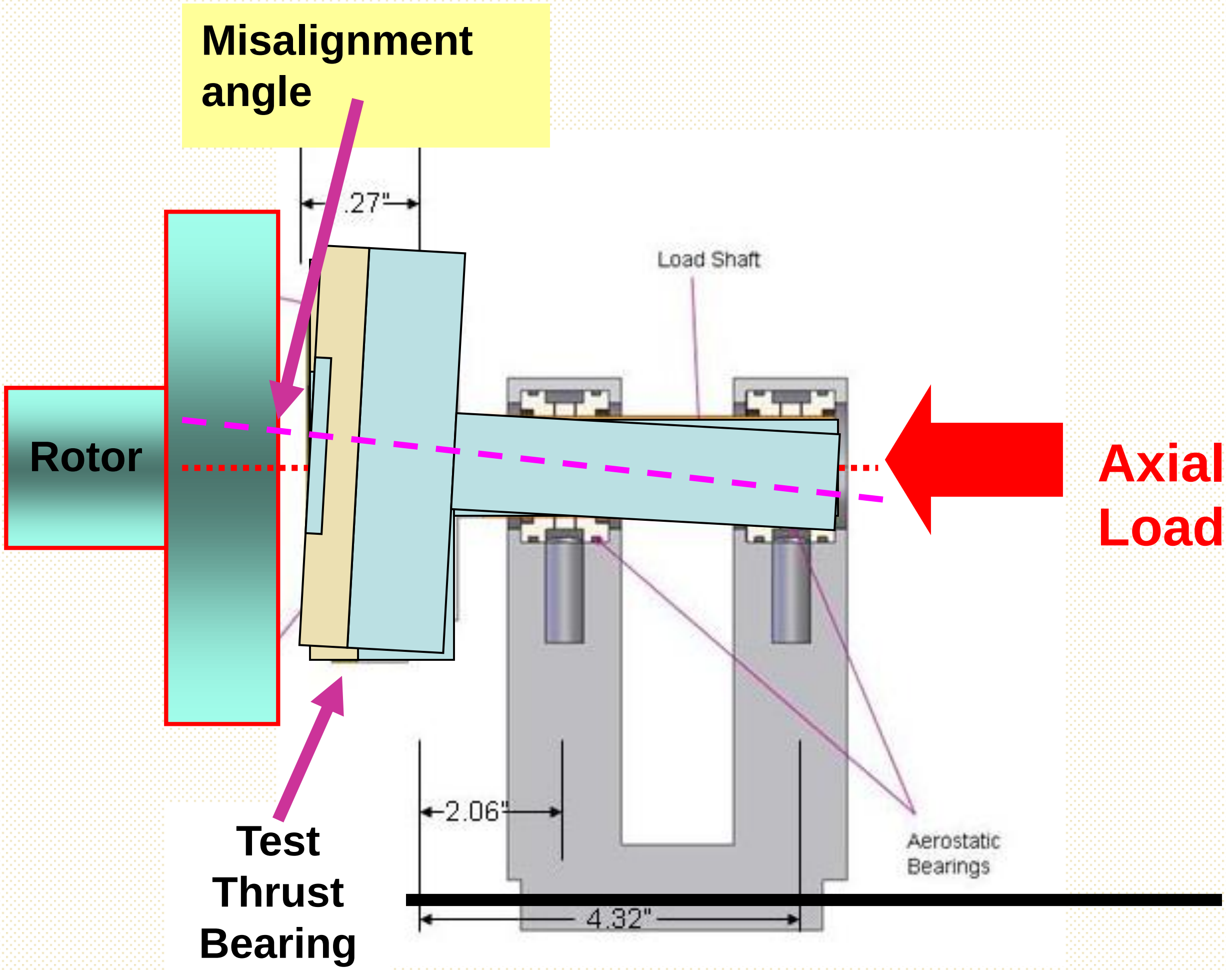
- Open water lines and SET supply pressure into radial bearings (100 psig)
- Supply pressure to gas bearings supporting axial load arm (100 psig)
- **SET supply pressure into thrust bearings:** 50 psi (3.4 bar) → 250 psig (17.2 bar)

- SET rotor speed (max 25 krpm): **7.5, 10.5, <u>17.5 krpm</u>**

Circ. flow Reynolds number $Re_c = ½ (\nu W c_o D_{OD}) = 18{,}040$ to 4,370.
Radial flow Reynolds number $Re_r = \nu Q_{OD}/(\pi D_{OD}) = 1{,}145$

- **Shaker applies axial load** – increasing & decreasing
- **MEASURE** rotor axial displacements (clearances) at test & slave bearings, **RECORD** flow rates (in & out), pocket pressures, supply and discharge P&T's

Loading action and thrust face misalignment



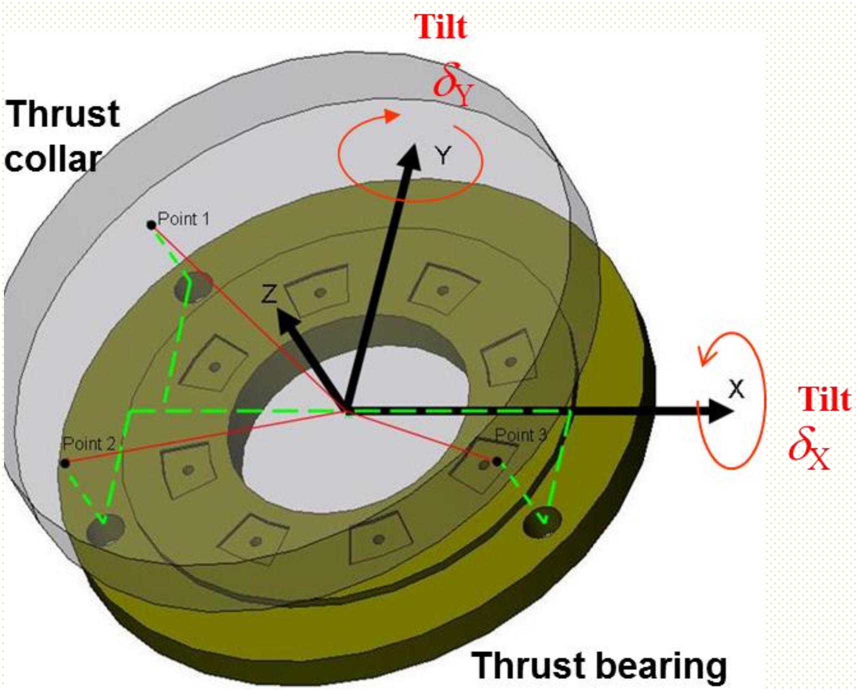
Chronic thrust bearing face misalignment minimized with careful alignment of load shaft support with shims.

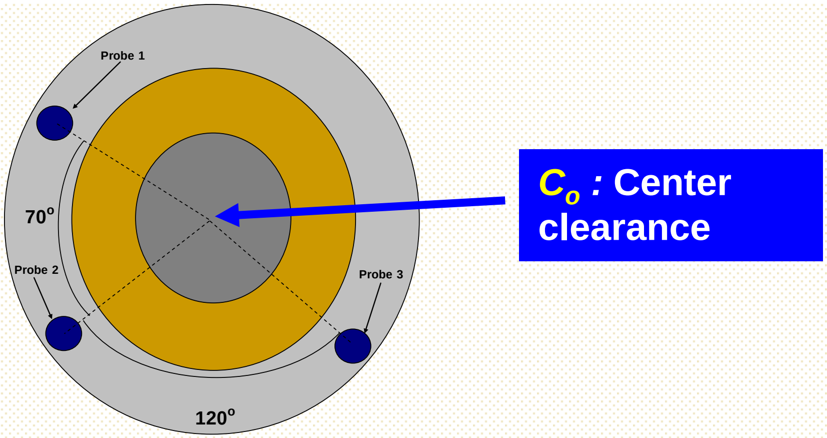
Measurements fully assess clearance variations with load.

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

Axial clearance measured at three angular locations

→ estimate center clearance and tilts (rotations).

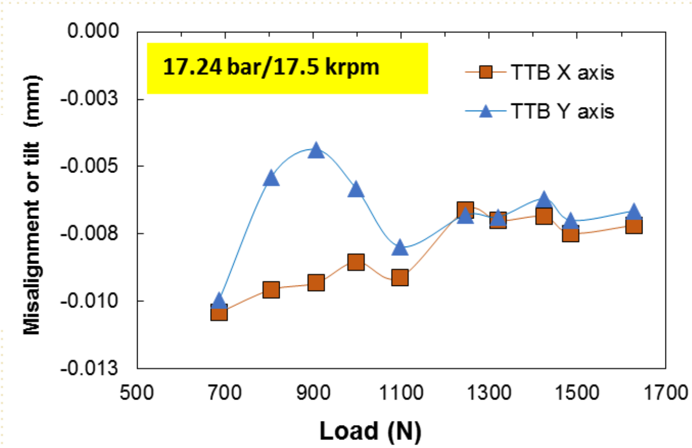




$$c_i = c_o + \underline{R}\cos(\theta_i)\ \delta_Y + \underline{R}\sin(\theta_i)\ \delta_X,\ \mathrm{i=1,2,3}$$

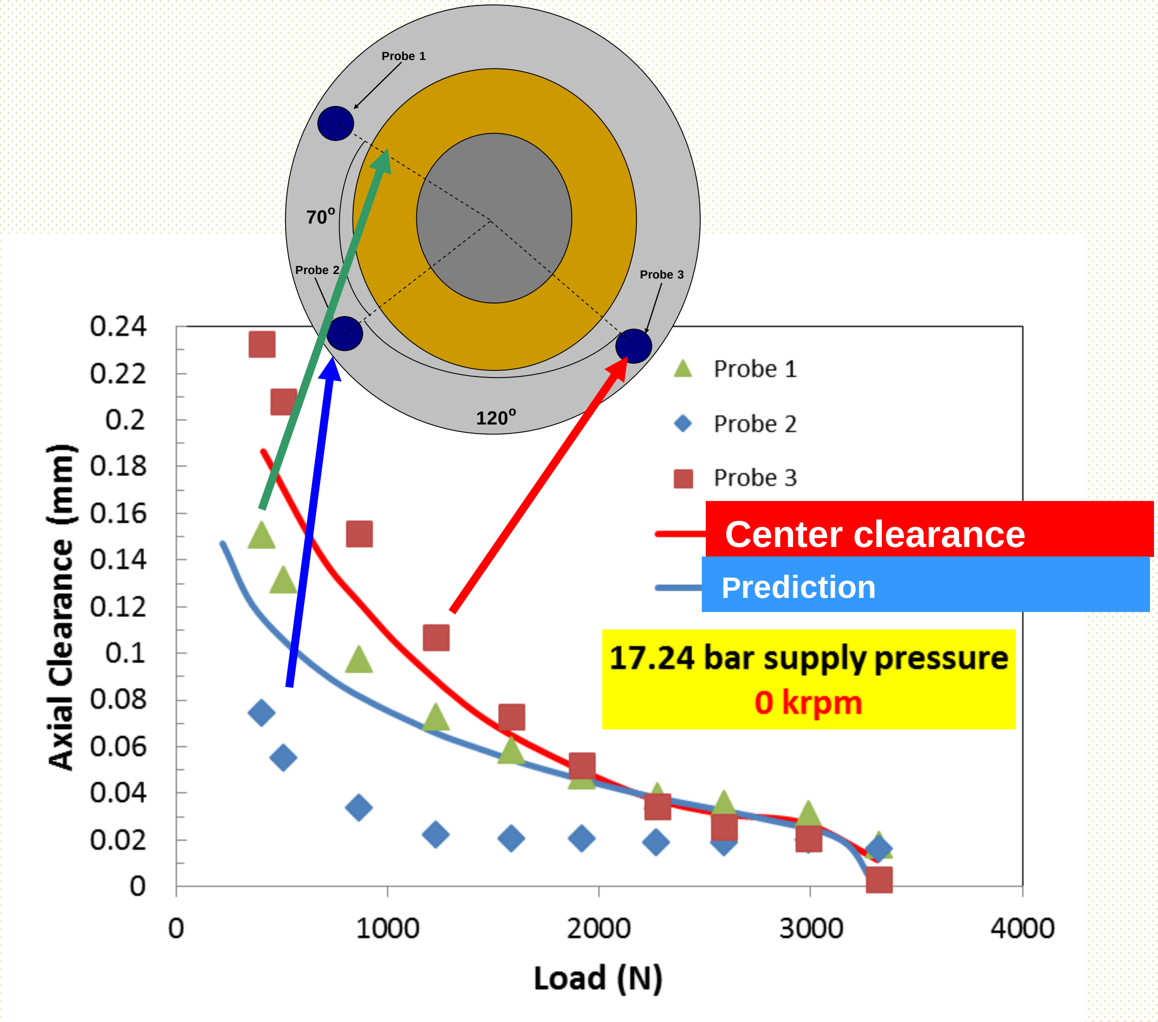
Example of tilts:

$\underline{R}$ x δ



TB clearances vs. axial load - 0 rpm

- No shaft rotation. Water supply pressure= 250 psig (17.2 bar)

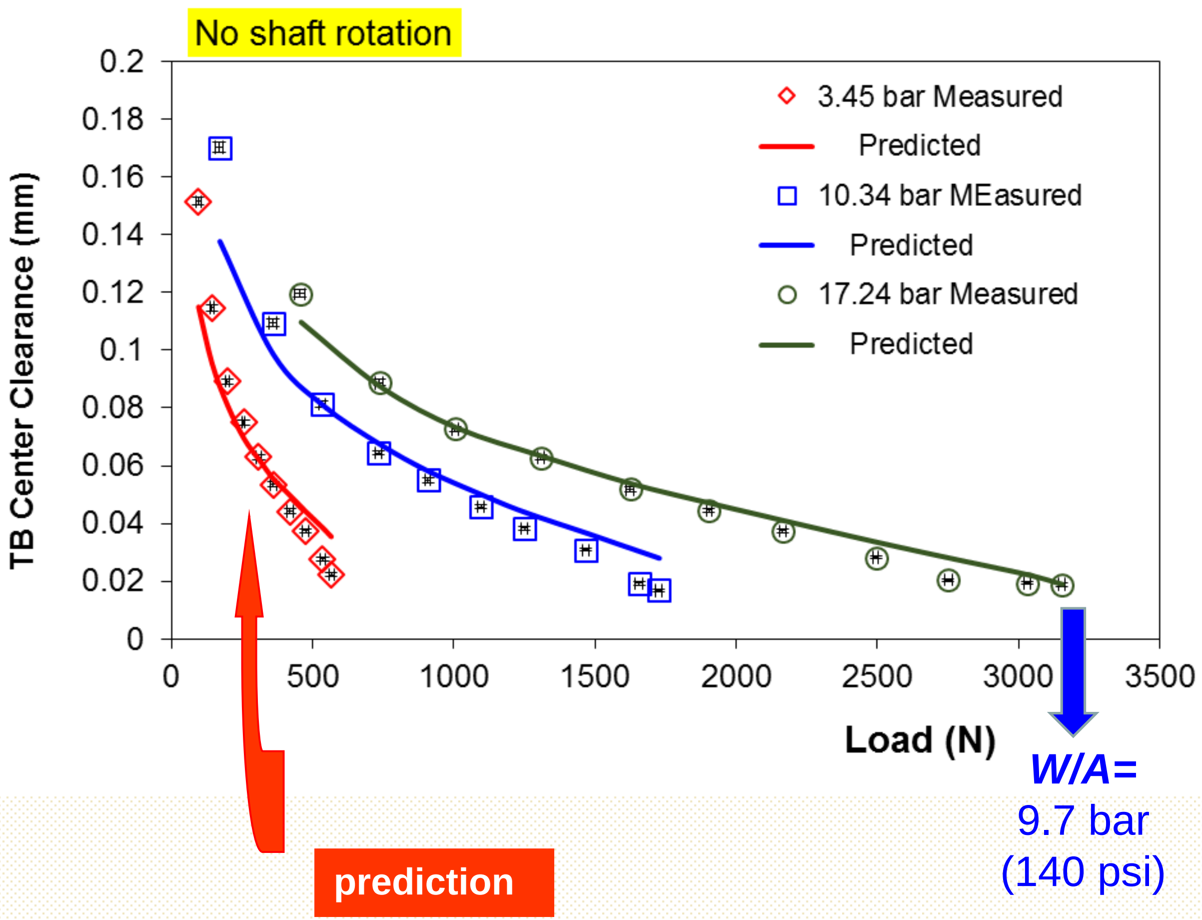


Test data shows significant collar tilts; worse at smallest load. Predictions agree well with center clearance at largest load.

TEST RESULTS & PREDICTIONS

TB clearance vs. axial load - 0 rpm

Water at 93°F (34°C) and supply pressure at 3.42, 10.34 & 17.2 bar (250 psig)

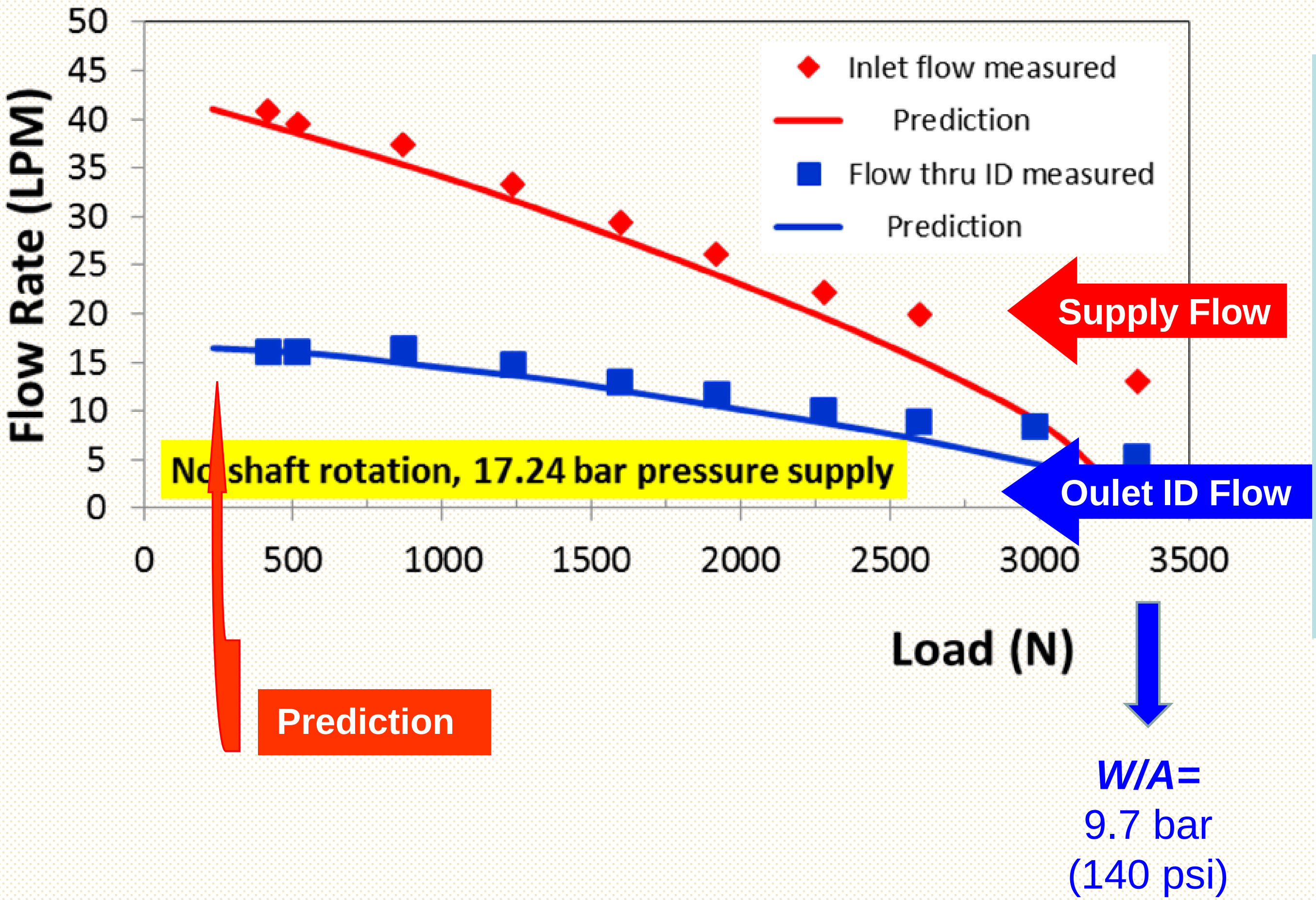


Axial clearance decreases exponentially with load. Predictions agree well with test data

TEST RESULTS & PREDICTIONS

TB flow rate (supply and ID) vs. axial load – 0 rpm

Water at 93ºF (34ºC) & supply pressure at 17.2 bar (250 psig). No shaft rotation



Flow rate decreases as load increases since axial clearance becomes small.

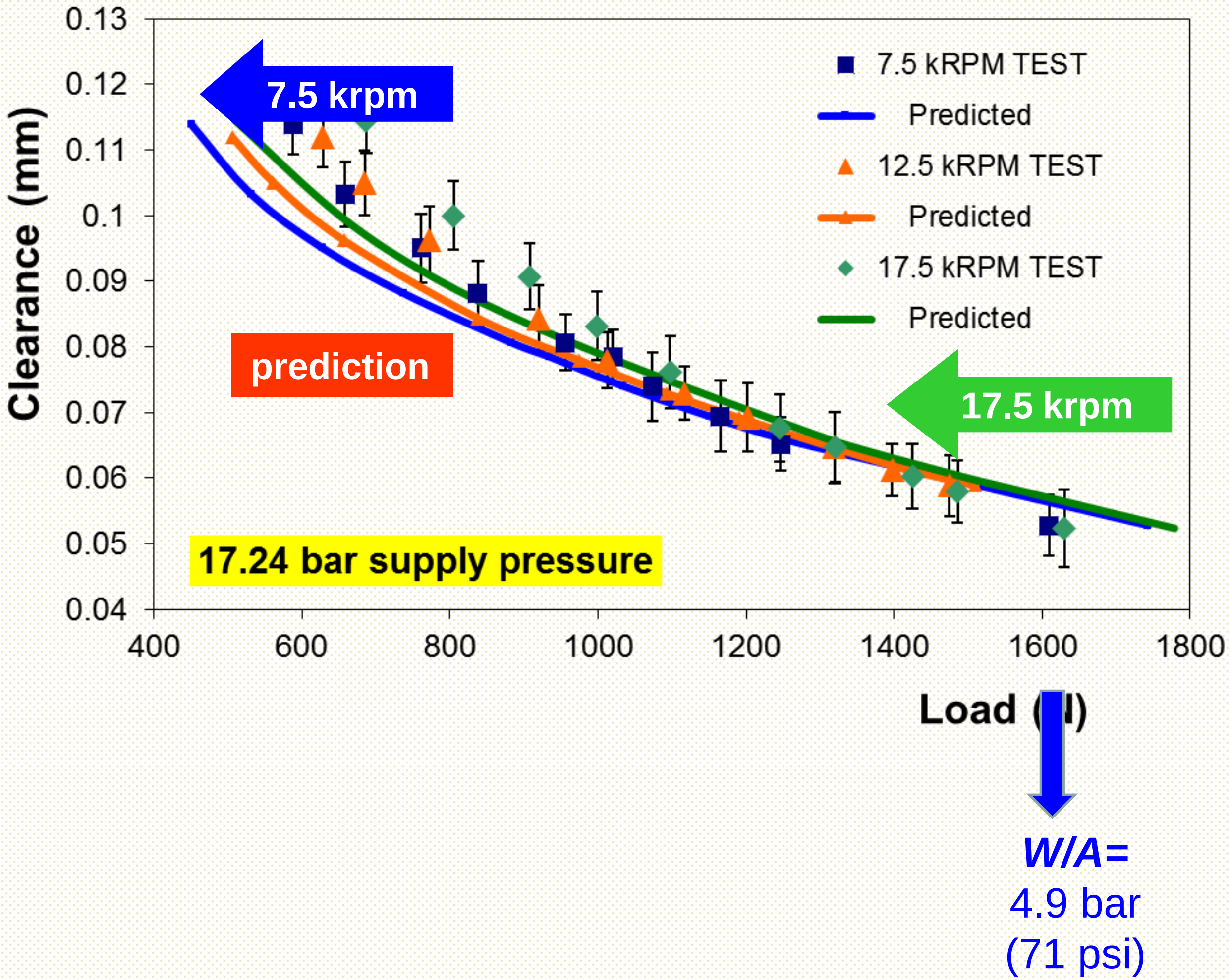
ID flow is NOT 50% of supplied flow

Predictions agree well with test data

TEST RESULTS & PREDICTIONS

TB clearance vs. axial load: 7.5, 12.5 & 17.5 krpm

Water at 93°F (34°C) and supply pressure at 17.2 bar (250 psig)



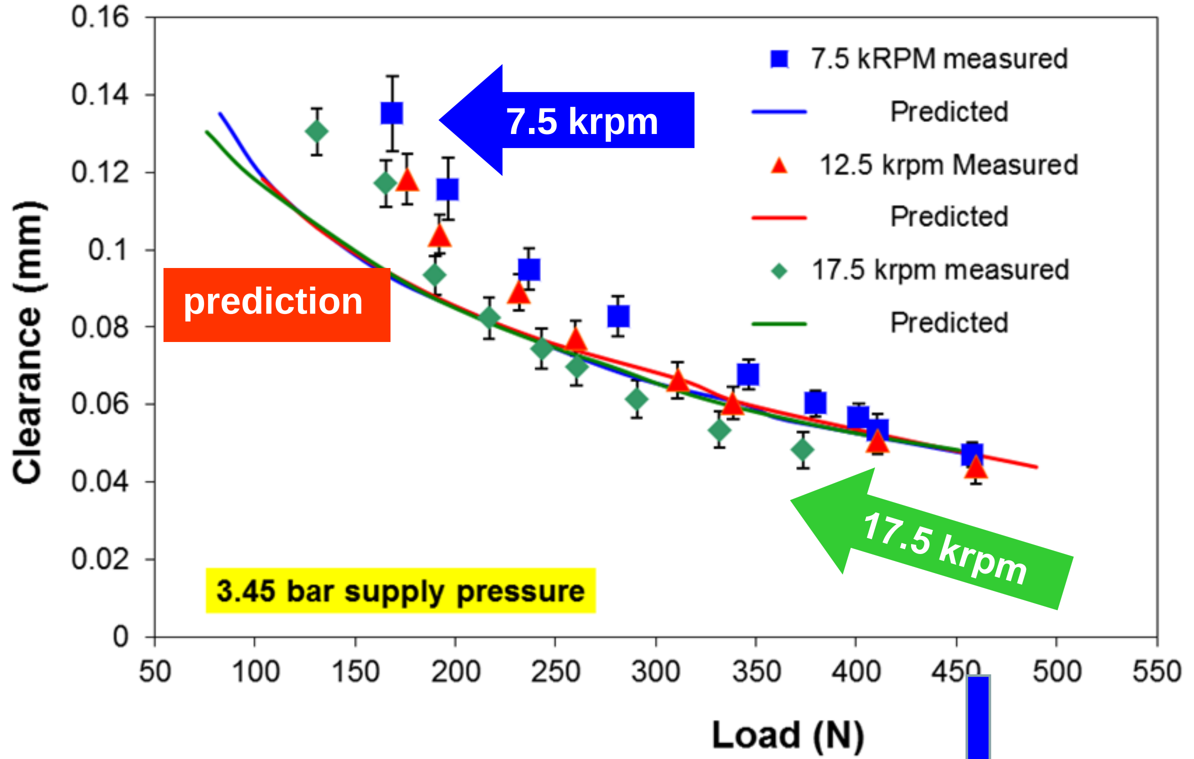
Axial clearances is not a strong functions of rotor speed – hydrostatic effect mainly

Predictions agree well with test data; best at highest load (1.6 kN)

TEST RESULTS & PREDICTIONS – high pressure

TB clearance vs. axial load: 7.5, 12.5 & 17.5 krpm

Water at 93ºF (34ºC) and supply pressure at 3.45 bar (50 psig)



***W/A*=**
1.4 bar
(20 psi)

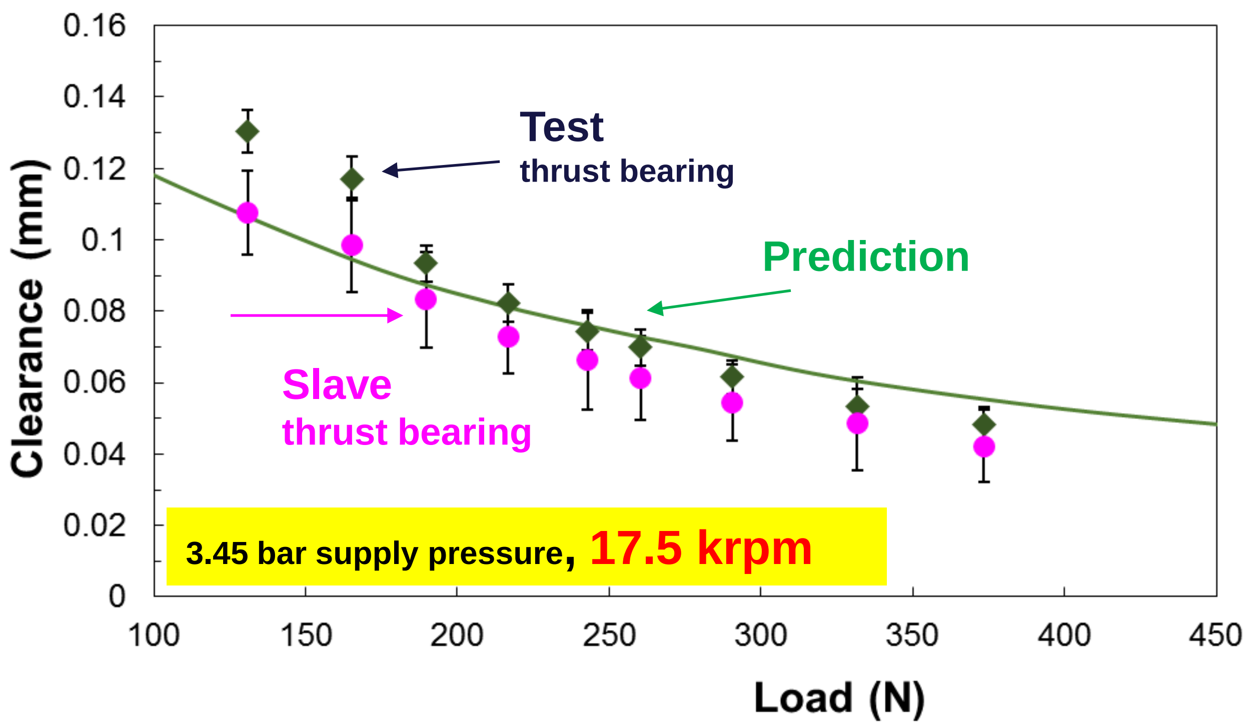
Tests show axial clearance is a function of rotor speed – hydrodynamic effect.

Predictions agree with tests at highest load (0.46 kN)

TEST RESULTS & PREDICTIONS – low pressure

Compare test and slave thrust bearings

Water at 93°F (34°C) and supply pressure at 3.45 bar (50 psig)

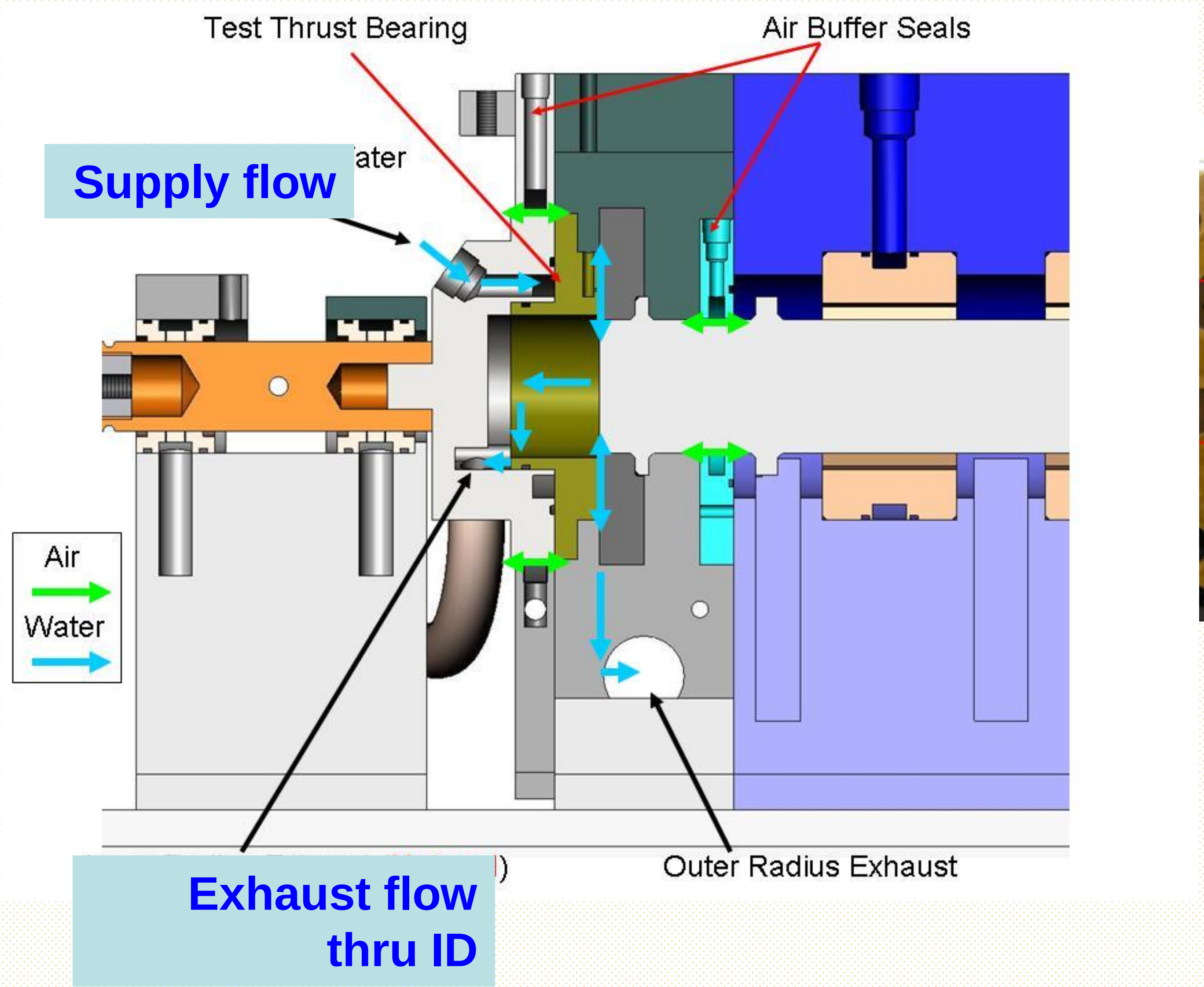


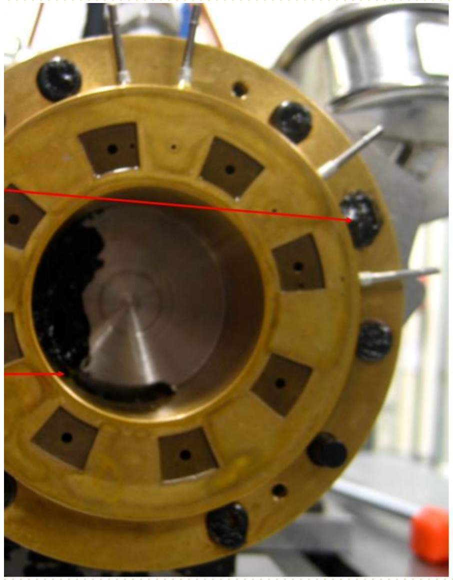
Slave TB has different orifice diameter → gives lesser clearance.

Both TBs perform similarly.

TEST RESULTS & PREDICTIONS

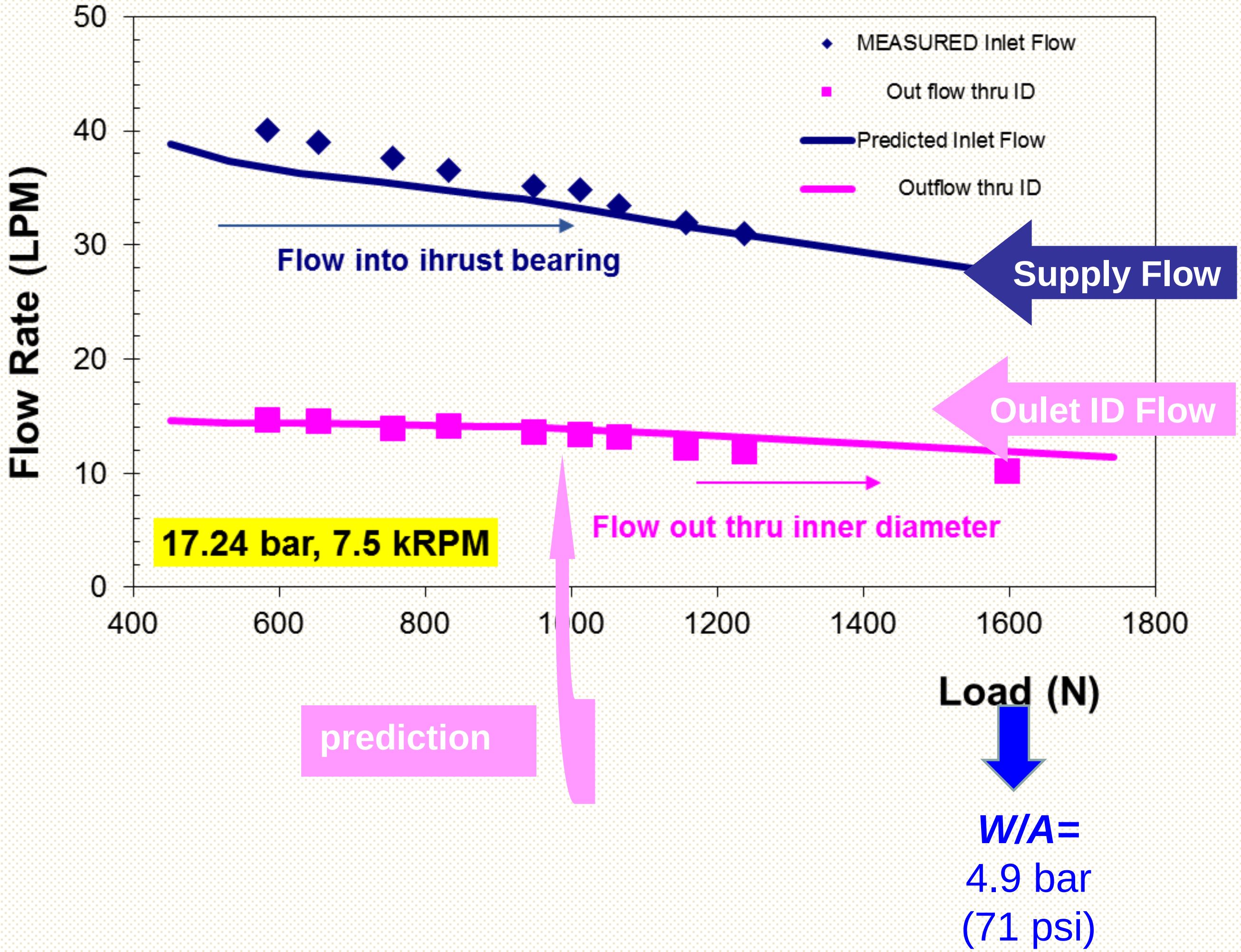
TB flow rates (supply and ID)





TB flow rate (supply and ID) vs. load – 7.5 krpm

shaft speed = 7.5 krpm. Water supply pressure= 250 psig (17.2 bar)

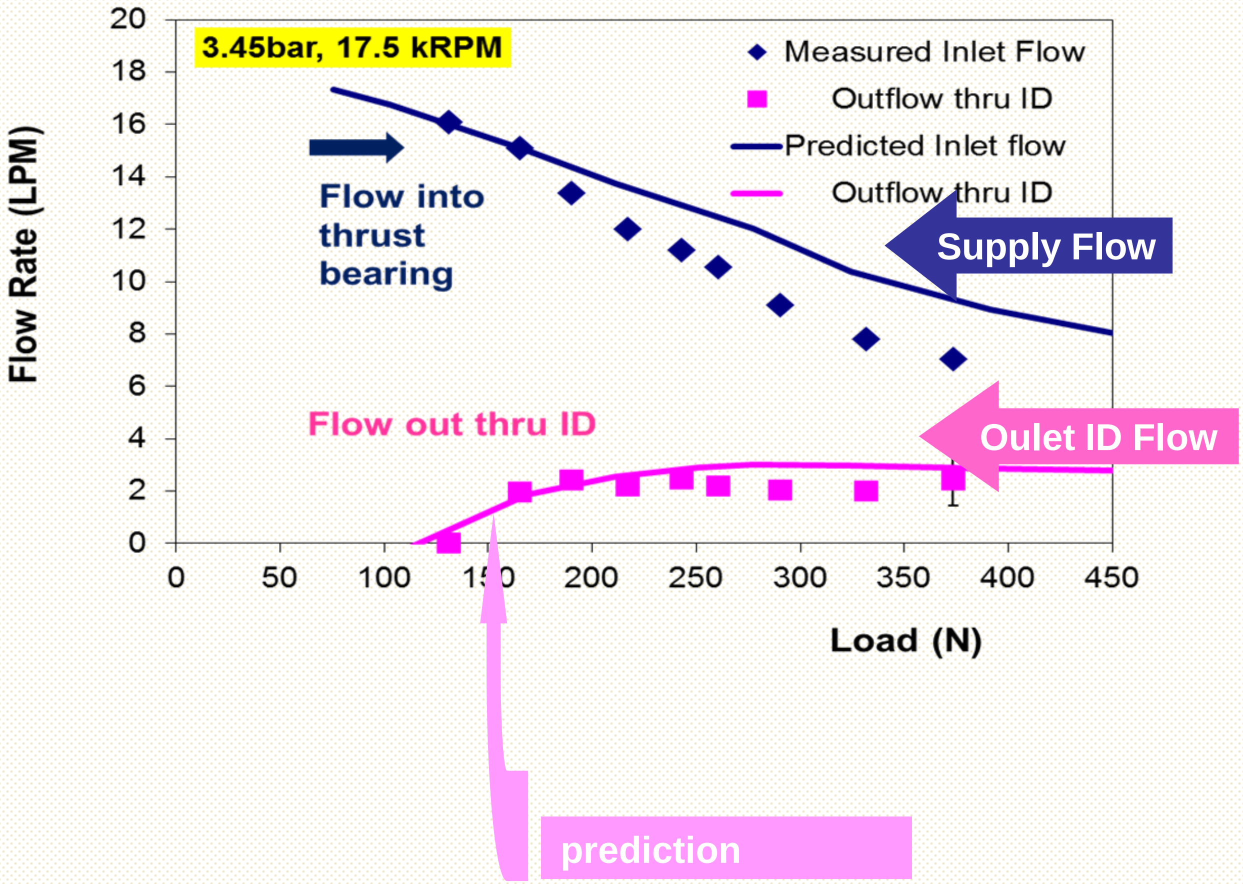


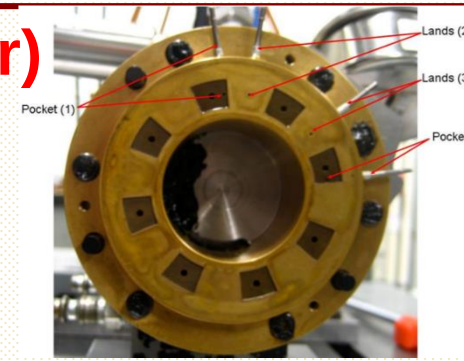
Flow rate decreases with load and rotor speed.
ID flow is less than 50% of supplied flow.
Predictions match well with test data

TEST RESULTS & PREDICTIONS – high pressure

TB flow rate (supply & ID) vs. load – 17.5 krpm

shaft speed = 17.5 krpm. Supply pressure= **50 psig (3.4 bar)**





At low supply pressure and high rotor speed, inner side of bearing starves!

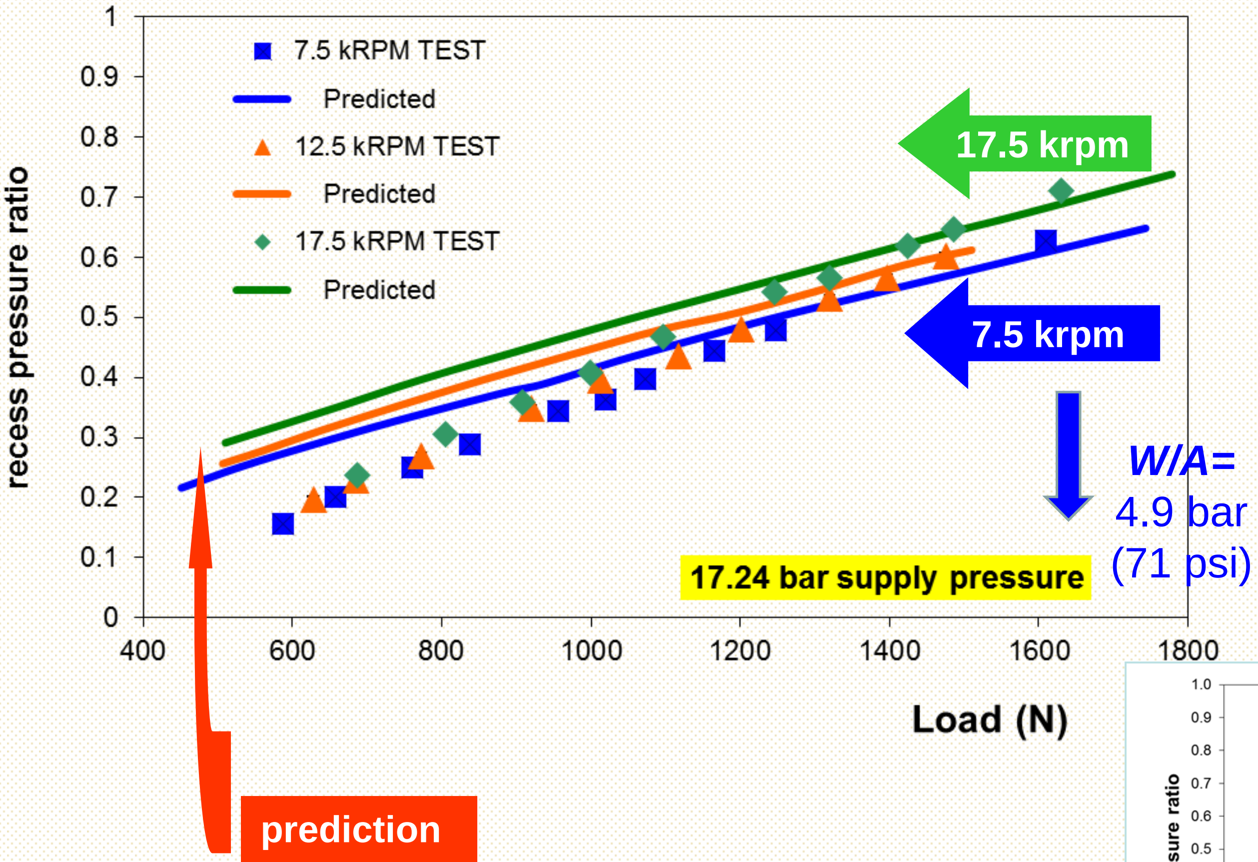
ID flow << 0.5 x supply flow.

Predictions agree with test data - demonstrate importance of centrifugal flow effects.

TEST RESULTS & PREDICTIONS – low pressure

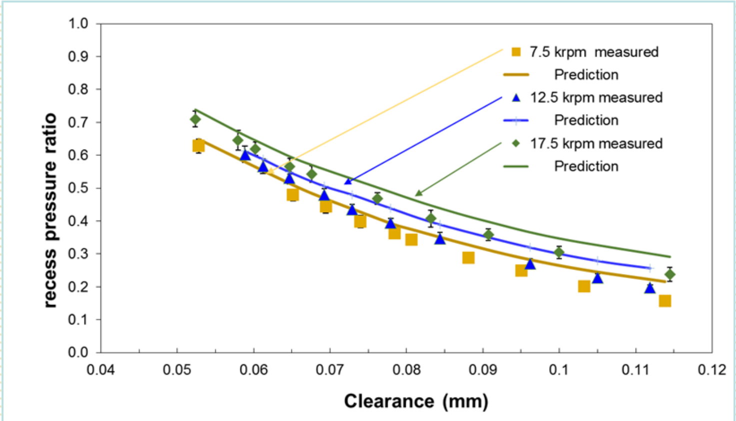
Recess pressure vs. axial load: 7.5, 12.5 & 17.5 krpm

Water at 93ºF (34ºC) and supply pressure at 17.2 bar (250 psig)



Pocket pressure approaches supply pressure as load increases.

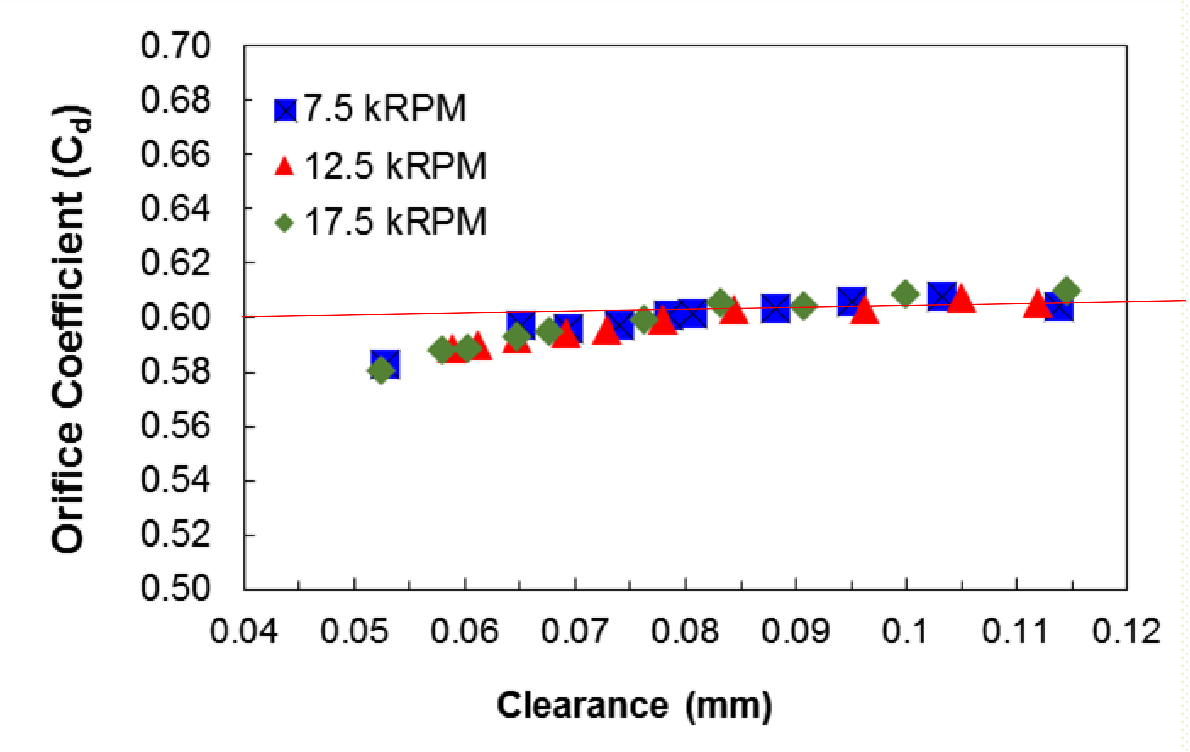
Predictions agree at highest load (1.6 kN)



TEST RESULTS & PREDICTIONS

Orifice discharge coef. vs. clearance: 7.5, 12.5 & 17.5 krpm

Water at 93°F (34°C) and supply pressure at 17.2 bar (250 psig)



C_d varies from 0.58 for small clearance (large load) to 0.60 for the largest clearance (lowest load).

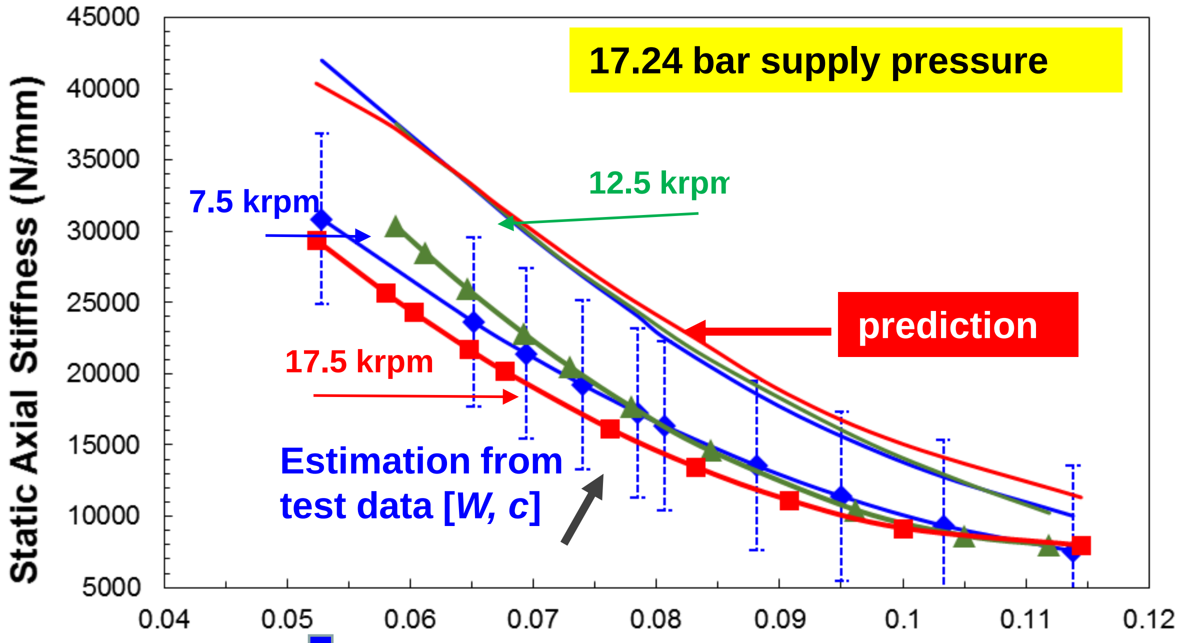
C_d used for prediction of bearing performance.

Derived from test results

$$C_d = \frac{Q_o}{A_o} \frac{1}{\sqrt{\frac{2}{\rho}\left(P_S - P_R\right)}}$$

TB stiffness K_z vs. clearance: 7.5, 12.5 & 17.5 krpm

Water at 93°F (34°C) and supply pressure at 17.2 bar (250 psig)



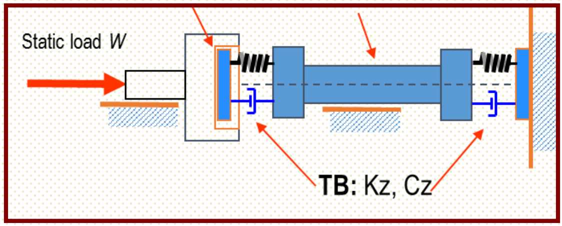
W/A= 4.9 bar (71 psi)

Static load range: 1600 N → 600 N

Test static *K* derived from (curve fit) of load vs clearance.

Predictions over estimate stiffness. Worse at highest load (1.6 kN) : smallest clearance

Dynamic force coefficients not obtained- program lost funding.



TEST RESULTS & PREDICTIONS

Conclusion

Measurements of hybrid thrust bearing static load performance obtained with water at 50 to 250 psig (3.4 to 17.2 bar) supply pressure and rotor speed to 17.5 krpm.

Circ. flow Reynolds number $Re_c = \frac{1}{2} (\nu W c_o D_{OD}) = 18{,}040$ to $4{,}370$.
Radial flow Reynolds number $Re_r = \nu Q_{OD} / (\pi D_{OD}) = 1{,}145$

Chronic TB face misalignment issues minimized. Predictive tool accounts for effect. The measurements show:

- **Centrifugal flow effects due to rotation cause fluid starvation on the inner side of hybrid thrust bearing. Effect is worst at lowest pressure and highest rotor speed.**

- **Predictive tool reproduces recorded bearing static performance, i.e. operating clearance decreasing exponentially with applied load and lubricant starvation.**

- **Research products satisfy:**

a) verification of hybrid thrust bearing performance,
b) experimentally validation of predictive tool.

Acknowledgement

Paper GT2016-56349

Thanks AFRL-USET Program & Northrop Grumman Space Technology

Mr. Alan Sutton (AFRL),

Mr. Chuck Papesh and Mr. Gordon Dressler (Northrop Grumman)

Mr. Tim Miller and Mr. Dave Lowe (Barber-Nichols Inc.)

Graduate MEEN students: Mr. Michael Forsberg, Mr. Fernando Ramirez

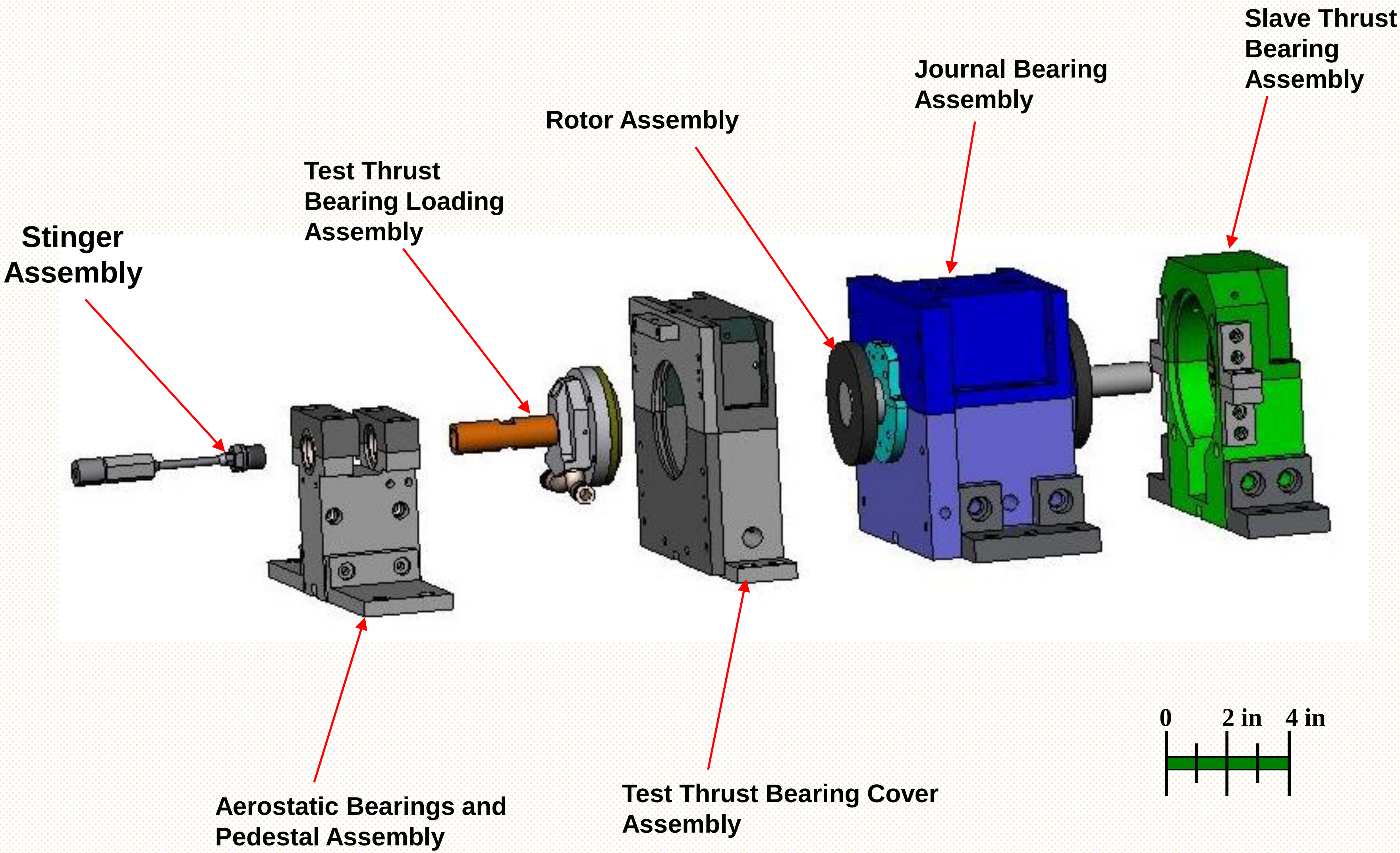
Undergraduate MEEN students: Ms. April Acosta, Mr. Scott Wilson

Questions (?)

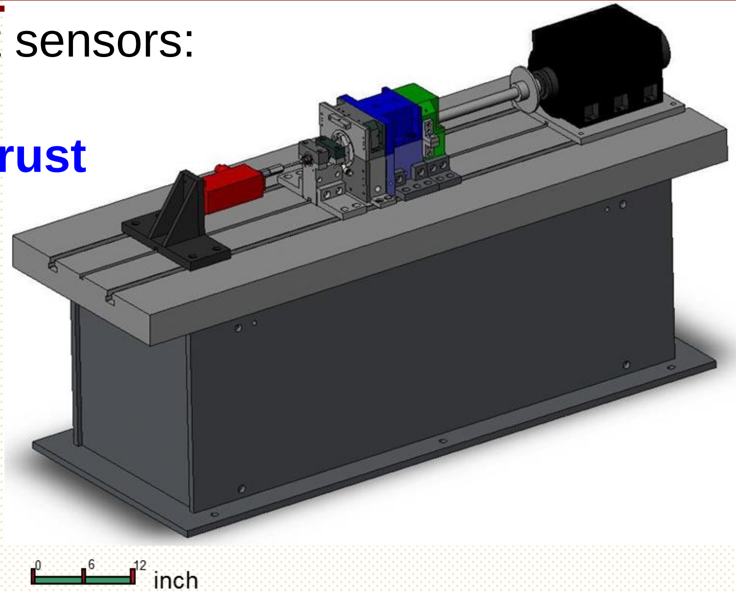
Learn more at http://rotorlab.tamu.edu

Backup slides

Test Rig – Components Isometric View



Test Rig Instrumentation



Rotor lateral radial motions: 2 x 2 (X,Y) eddy current sensors:

Rotor collar axial and tilt motions at test & slave thrust bearings 2 x 3 (120 deg) (Z) eddy current sensors:

Thrust Force with strain gauge **load cell** & stinger connected to shaker.
Torque mechanism not active.

Turbine flow meters in supply lines to water radial bearings, and **test thrust bearing INLET and OUTLET at ID**

Three (3) strain-gauge pressure sensors for measurement of **(2) pocket and (1) land pressures in test thrust bearing.**

Thermocouples: **water inlet & outlet of test thrust bearing**
Pressure gauges: **supply and discharge of test thrust bearing**

Tachometer: **rotor speed**

LABVIEW® DAQ system and control

Bearing & seals: TYPES

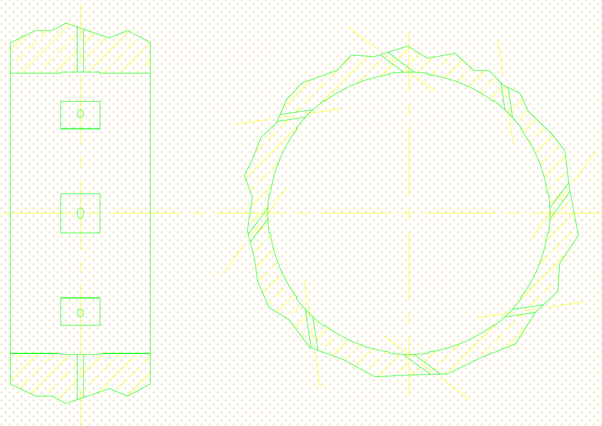
The predictive tools include **full fluid inertia, turbulence flow and thermohydrodynamic models for high-speed, high-pressure, hot/cold cryogenic and process fluid operating conditions.** Cryogenic fluids: O2, H2, N2 (liquid or gas)

HYDROJET - models

hydrostatic/hydrodynamic radial bearings, angled injection, roughened surfaces

Honeycomb seals and annular damper seals

tilting and flexure pivot journal bearings, simple foil bearings,



HYDROTHRUST - models

hydrostatic/hydrodynamic thrust bearings
Inner pressurized face seals with angular misalignment

2002/5: Excel® graphical user interfaces linking Hydro codes (Fortran DOS applications) to modern Windows based rotordynamics analysis software.

Hybrid Bearings: Bulk Flow Models

At Texas A&M Turbomachinery Laboratory: **Hydrojet® & Hydrothrust®**

Equations for flow in film lands of bearing	Mass conservation, Bulk-Flow momentum in circumferential and axial directions (2D), Energy transport for mean flow temperature Various surface temperature models Fluid inertia effects at entrance and exit flow regions.
Equations for flow in pockets of hydrostatic bearing	Global mass conservation: orifice inlet flow, flow from recess towards or from film lands, and rate of accumulation of fluid within pocket volume, Global momentum in circumferential direction due to angled injection. Global energy transport with adiabatic heat flow surfaces
Flow conditions	**Laminar, laminar to turbulent transition and fully developed turbulent bulk-flow model.** Turbulent flow closure model: Moody's friction factor including surface roughness. Fluid with variable properties *f*(P,T)

INPUT Hydrostatic Bearings: XLHYDROTHRUST®

Physical variable - INPUT	Practice	Notes
Fluid TYPE	**Cryogenic:** LH2, LO2, LN2	**MIPROPS:** 32-term state equations for cryogens
Geometry: Length, Diameter ID & OD, axial learance and tilt angles	**L, D: Constrained by TP dimensions**	Clearance most important for HIGH stiffness
Geometry: pocket area & depth,	**Keep pocket area/land ratio ~ 0.2**	Pocket depth/Clearance ~ 5 to 10 to avoid pneumatic hammer.
orifice diameter	**Sized to obtain pocket pressure ratio ~ 0.60**	Empirical value – experimentally validated.
Shaft speed	**Low-to-high**	Shaft rotation induces large centrifugal flow effects
Supply and discharge pressures	**Max load ~ 0.25 Pressure drop x L x D**	**Hydrostatic bearings have limited load capacity**

Predictions THRUST Hybrid Bearings

Physical variable - prediction	Experimental validation	Notes
Bearing flow rate & pocket pressures	yes	As shat speed increases, flow through inner diameter decreases. Potential for bearing starvation and bearing collapse
Drag torque & power and temperature rise	NO	
load capacity (fluid film forces and restoring moments),	Yes	
9 complex impedance force coefficients due to axial shaft motions and collar axis rotations	NO Static AXIAL stiffness only	Damping coefficients are large in test facility

Learn more:

Forsberg, M., "Comparison Between Predictions and Experimental Measurements for an Eight Pocket Annular Hydrostatic Thrust Bearing," M.S. thesis, Texas A&M University, College Station, TX, May 2008.

Ramirez, F., "Comparison Between Predictions and Measurements of Performance Characteristics for an Eight Pocket Hybrid (Combination Hydrostatic/Hydrodynamic) Thrust Bearing," M.S. thesis, Texas A&M University, College Station, TX, December 2008

Funding for HB Tool development

Rocketdyne (1988-1991),
Pratt & Whitney (1991-92),
NASA GRC (1993-1996),
NASA MSFC (1998/99-2001/2)
Norhtop Grumman (2005-2007) - (USET Program)

All US turbo pump manufacturers and NASA, including SNECMA-SEP, use Hydrojet® and Hydrothrust® to model cryogenic fluid film bearings and seals. Other industries and Universities have benefited from technology.

USET Program

CLIN 4.2.1.3.2 (a) Non-linear forced response of fluid film bearing
CLIN 4.2.1.3.2 (b) Mixed flow regime – lift off response
CLIN 4.2.1.3.7 Experimental Study of Hydrostatic / Hydrodynamic Thrust Bearings