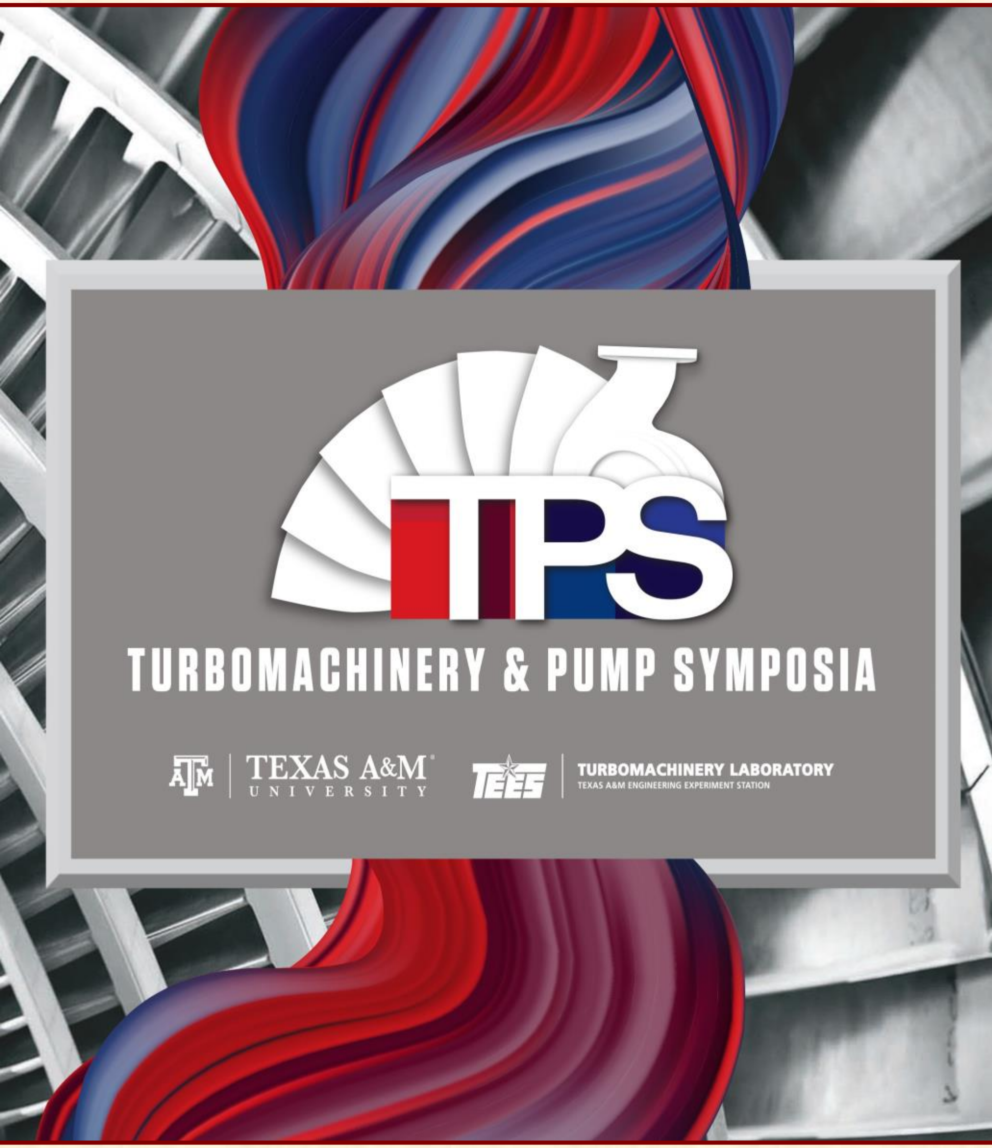

EFFECT OF REDUCED FLOW RATE ON THE STATIC AND DYNAMIC PERFORMANCE OF A TILTING PAD JOURNAL BEARING RUNNING IN BOTH FLOODED AND EVACUATED CONDITIONS

Funded by Turbomachinery Research Consortium

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Abstract

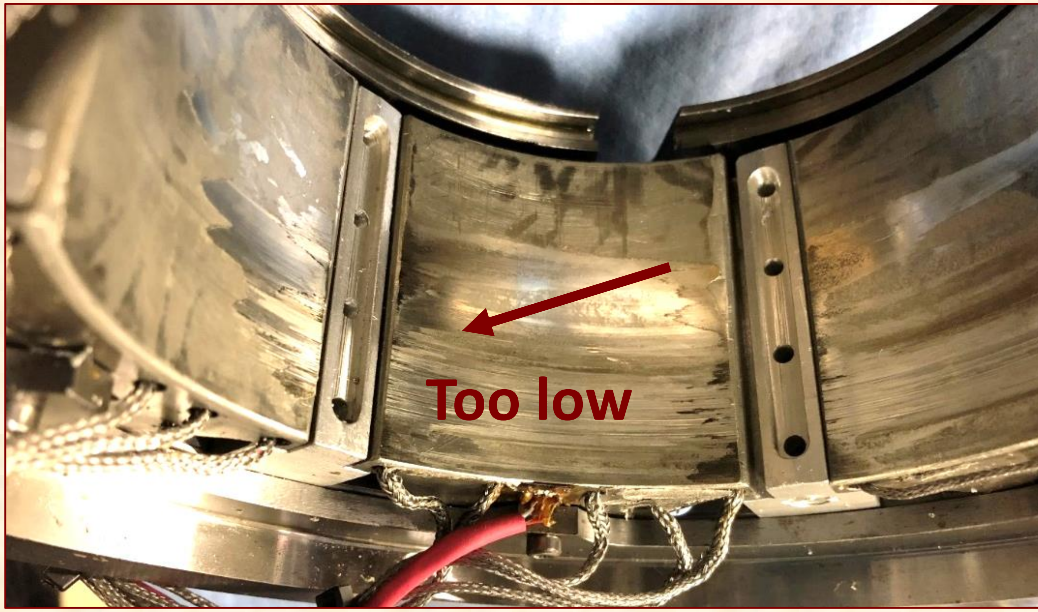
EFFECT OF REDUCED FLOW RATE ON THE STATIC AND DYNAMIC PERFORMANCE OF A TILTING PAD JOURNAL BEARING: FLOODED AND EVACUATED

The lecture presents measurements of the static and dynamic load performance in a tilting pad journal bearing running under flooded and evacuated conditions, and lubricated with flow rates ranging from a nominal rate to over flooded (150% nominal), and then to a starved flow (25% or lesser of nominal). A reduction in flow rate makes both bearings operate more eccentrically. The evacuated bearing operates at a larger eccentricity, which for the lowest flow rate (25% or so of nominal) does not align with the direction of the applied load, hence displaying a sizable attitude angle. Pad temperatures are similar for both bearing configurations, although the evacuated bearing is colder by a few Celsius degrees; and its oil exit temperature is much lower, in particular for an over flooded condition. Drag power losses derived from the oil exit temperatures show the evacuated bearing produces up to 40+% lesser power loss. The bearings direct stiffnesses increase with load and show little dependency on shaft speed. Direct damping coefficients reduce in magnitude as the supplied flowrate decreases. For sufficiently small flow rates, operation at 6 krpm and under a low load (0.345 MPa) produced SSV Hash.

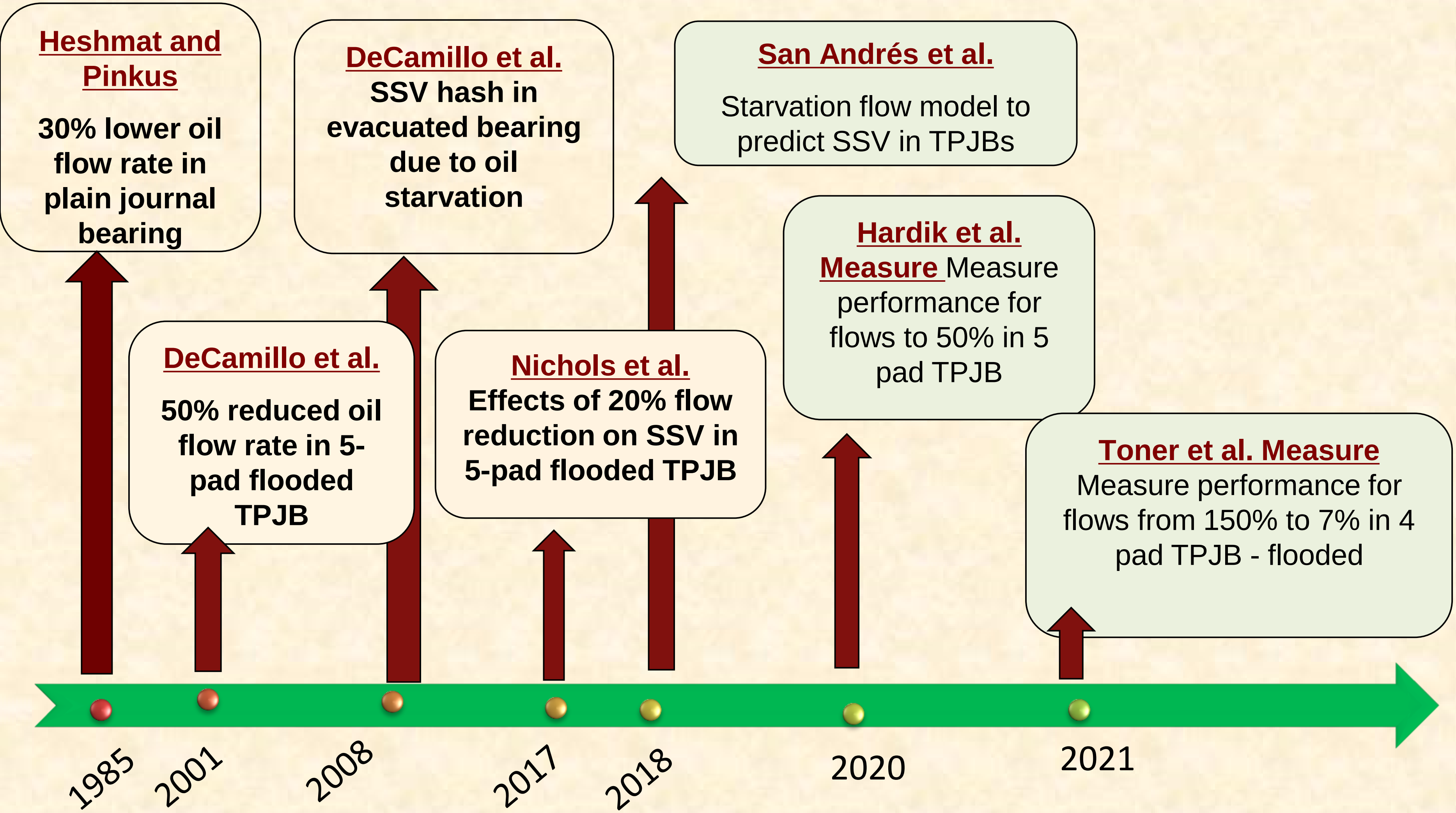
Control of flow in bearings

More cost and energy efficient bearings demand reduced flow rates and acceptance of hotter pad temperatures. Lesser flow reduces equipment footprint and cost.

how low is a low flow rate enough to maintain reliability (and energy efficient) TPJB operation ?



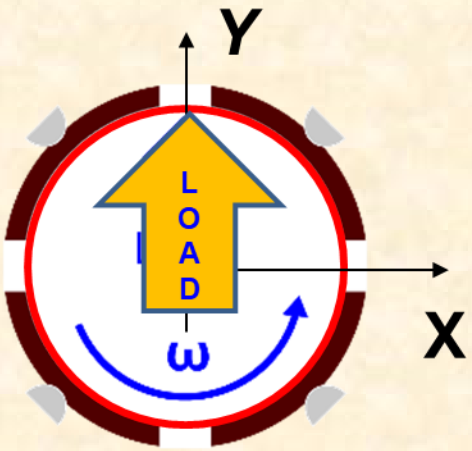
Prior art – not exhaustive



Flow reduction shows savings in drag power and increased oil and pad temperatures with magnitudes depending on specifics of application.

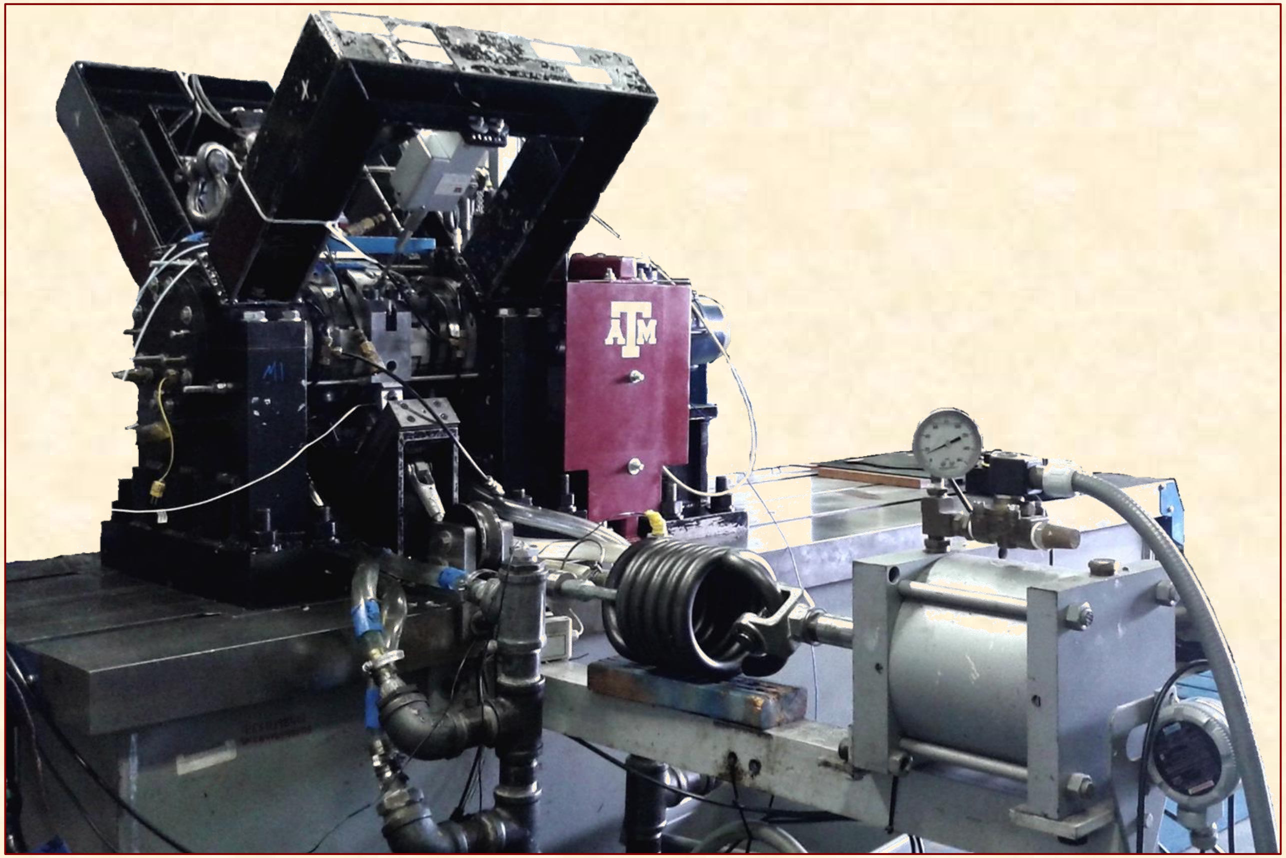
Objective

Obtain experimental data for flooded & evacuated bearings with flow rate from 150% to 25% of nominal.



- **Quantify the effect of flowrate on TPJB performance:**
  - **Load capacity and drag power**
  - **Pad metal temperatures**
  - **Force coefficients (*K, C, M*)**

Test Rig

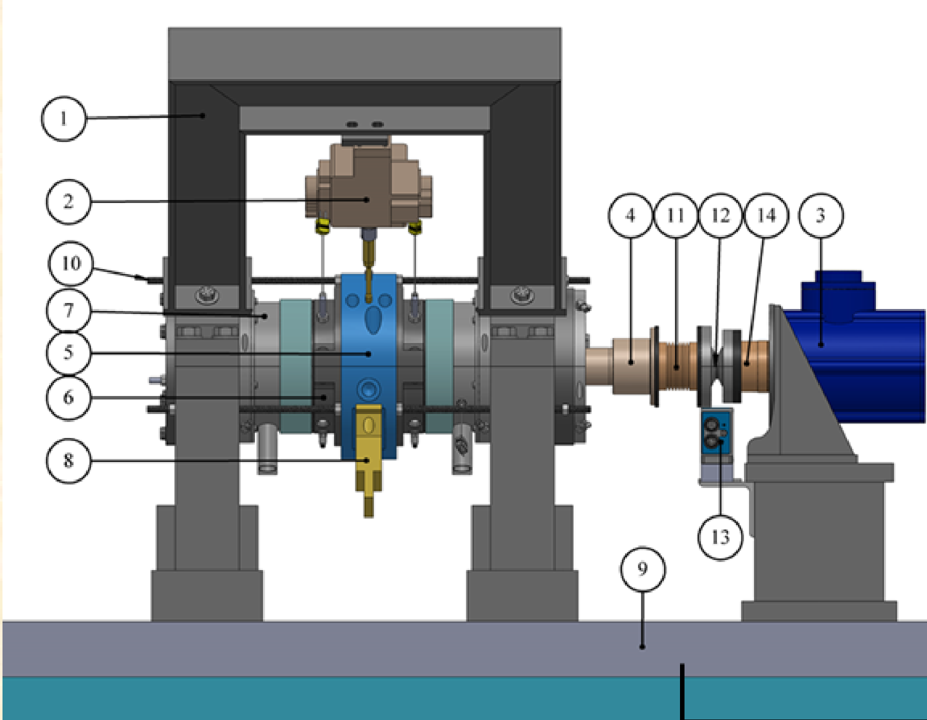


Industrial test rig for oil-lubricated bearing

Test Rig

Insert Video (480p)

Test Rig Features



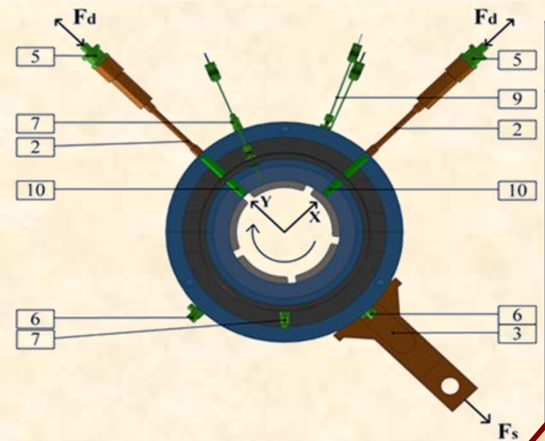
ITEM NO.	PART NO.
1	Test-rig pedestal
2	Hydraulic shaker
3	Air turbine motor
4	Test rotor
5	Bearing stator
6	Endcaps
7	Collection chambers
8	Static loader yoke
9	Bed plate
10	Pitch stabilizer bolts
11	Bellow coupling
12	Torque meter (rotor)
13	Torque meter (stator)
14	Torque limiter

Test-Rig Capability	
Max. rotor speed	16 kRPM
Max. applied static load	20 kN
Max. measurable torque	100 Nm
Max. supply oil flow rate	~20 GPM
Available shaft OD sizes	3.5”, 4”, 4.5”
Max. bearing length	3.5”

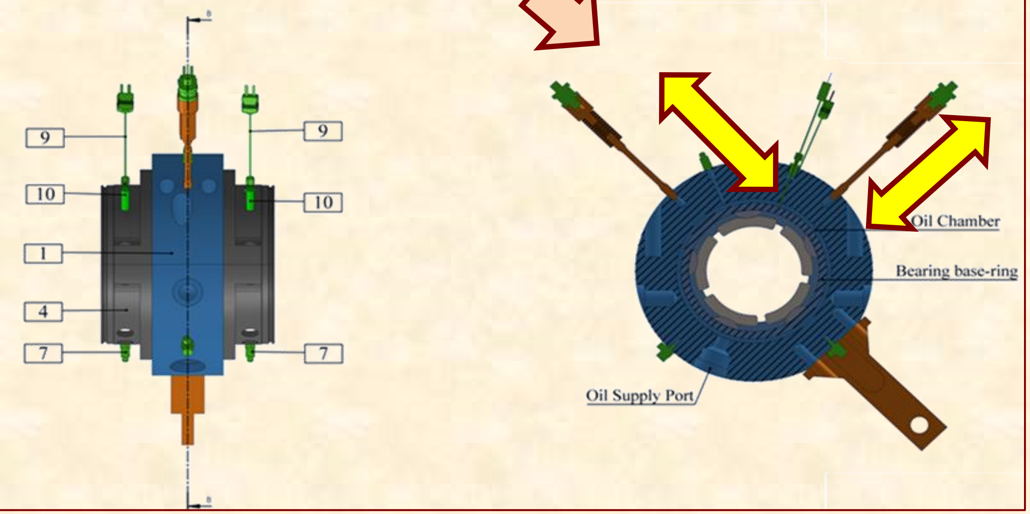
Strain gage torque meter & coupling directly measures drag torque.

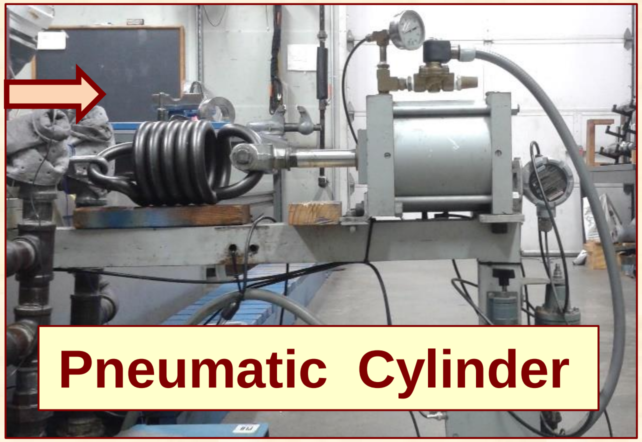
Floating bearing on rigid rotor.

Test Rig Load Devices

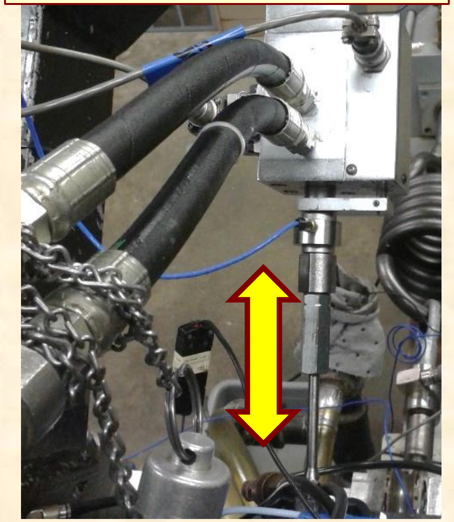


Item no.	Part Name	Qty.
1	Bearing stator	1
2	Stinger	2
3	Static loader yoke	1
4	Bearing end caps	2
5	Load cell (force transducer)	2
6	Accelerometer	2
7	Pressure sensor	3
9	Thermocouple	3
10	Eddy-current sensor	4



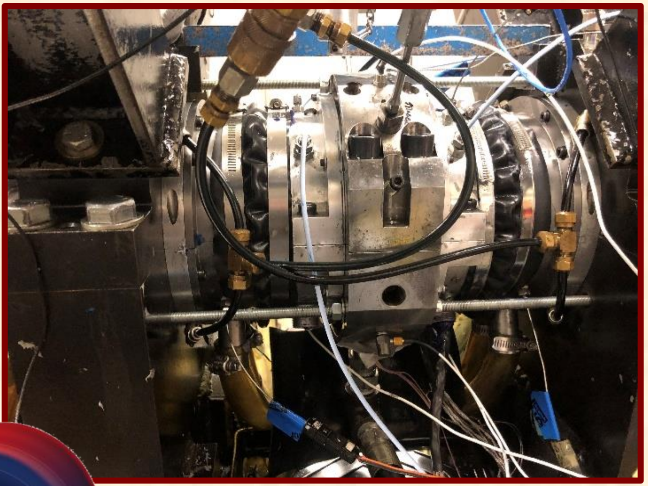


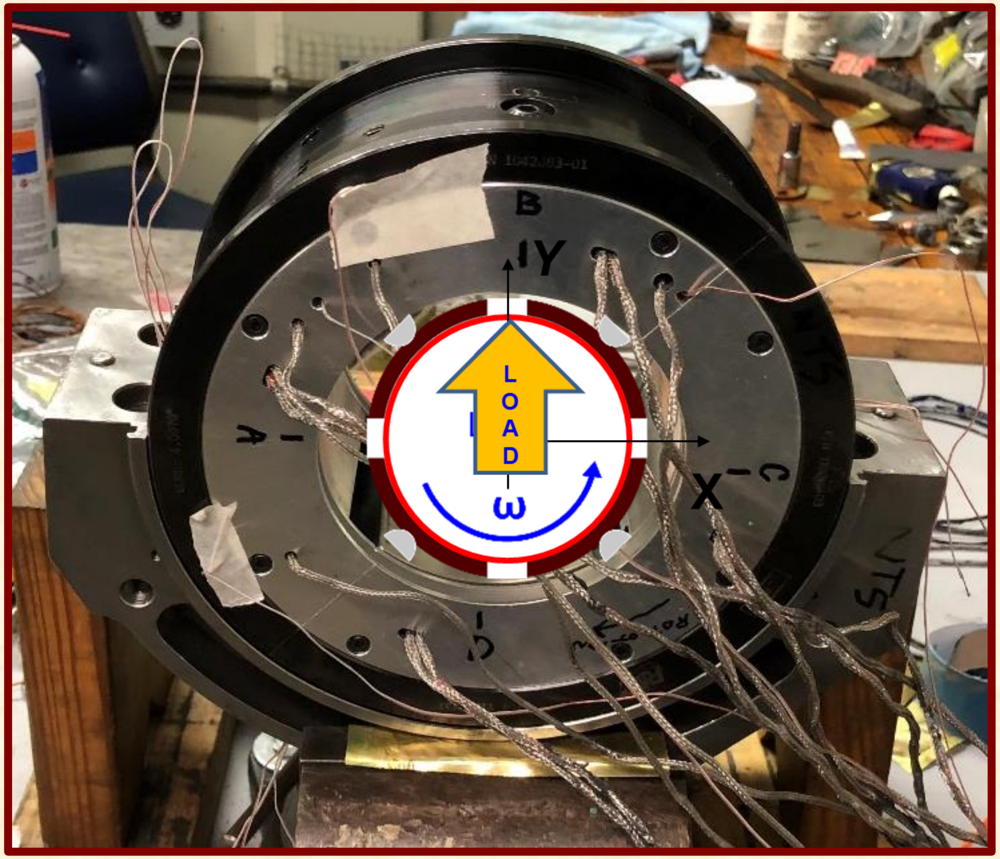
Hydraulic Actuator



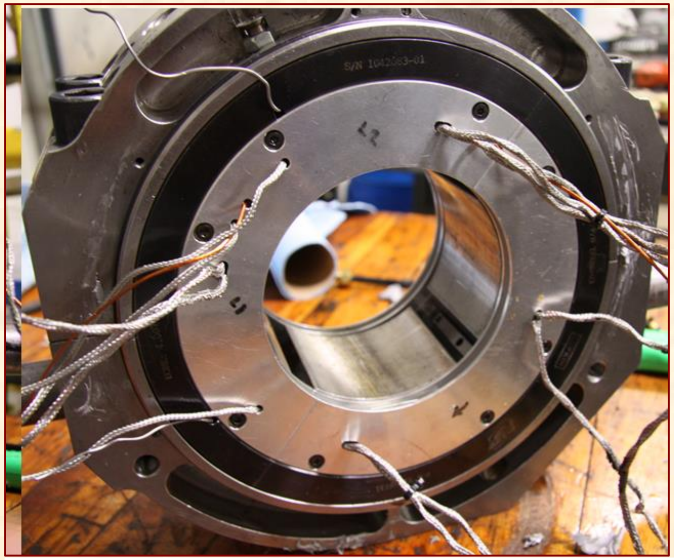
- **Pneumatic cylinder applies static load.**
- **Pair of hydraulic actuators deliver dynamic loads via stingers.**

Test Bearing





Test bearing – load between pads



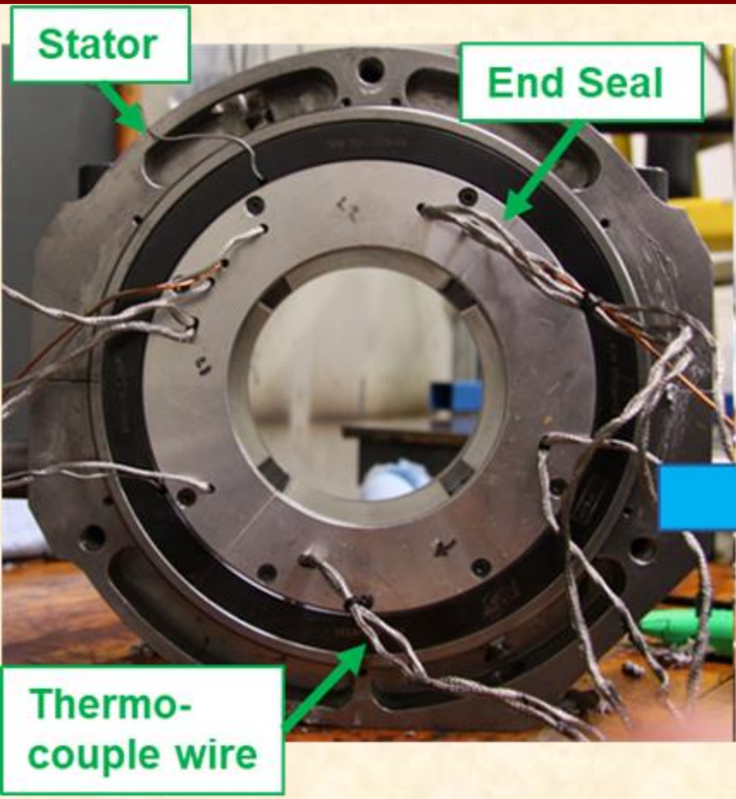
Load, *W*	2.13, 6.40, 12.8 kN
Specific Load, *W/(LD)*	345, 1,034, 2,068 kPa →

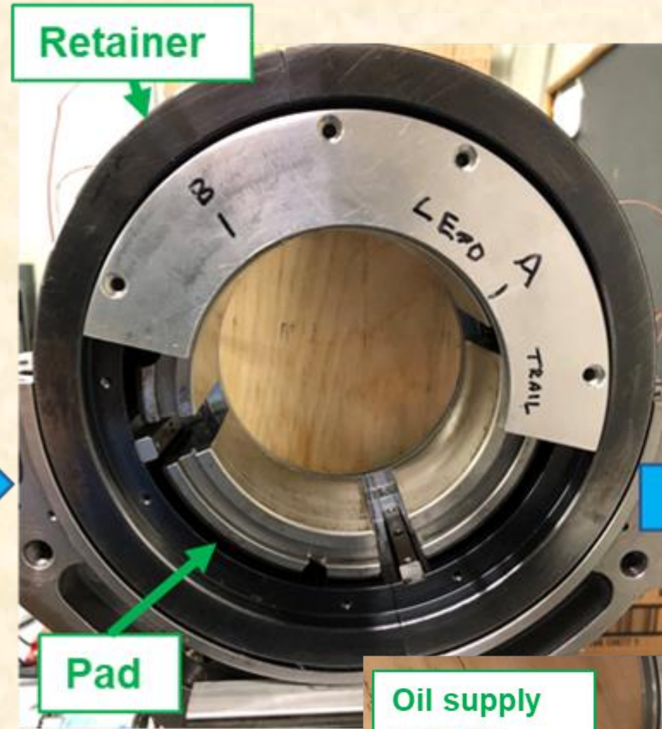
303 psi

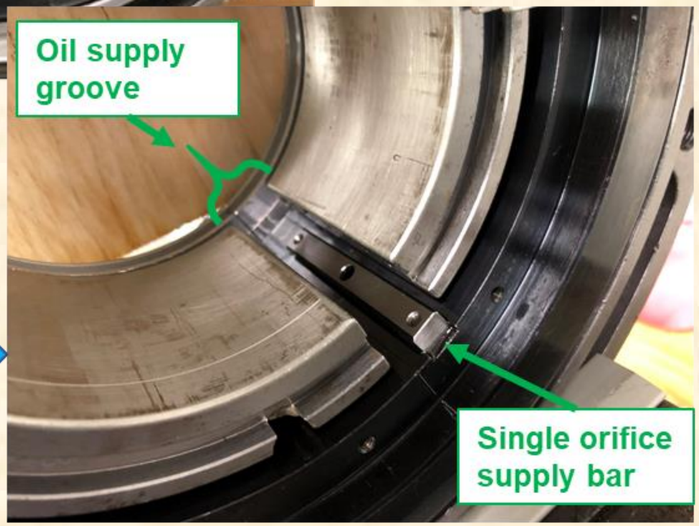
L/D	0.6
Shaft diameter	4.0 in (101 mm)
Length	2.4 in (61 mm)
B radial cold clearance	4.50 mil (0.115 mm)
Hot clearance (6 & 12 krpm)	4.20 mil (0.106 mm)
Design pad preload	0.3
Spherical Pivot Offset	0.5
Pad Arc Length (°)	72°
AISI 1018 Pad Thickness	0.75 in
Pad surface	Babbitt
Configuration	Flooded (with end seals) Evacuated (no end seals)

ISO VG 46 oil at 60C
16.4 cPoise & 837 kg/m^3

Flooded bearing configuration

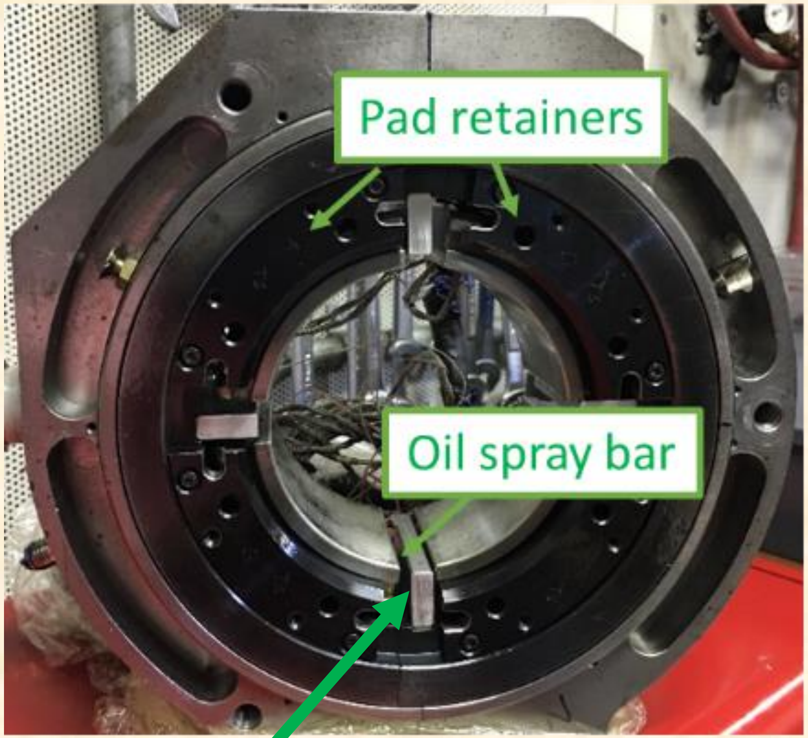


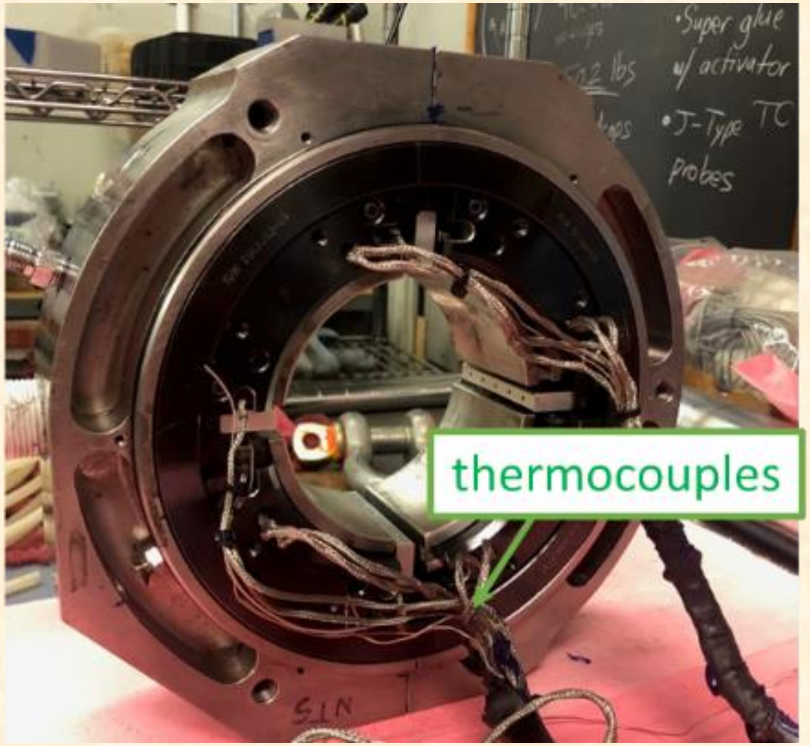


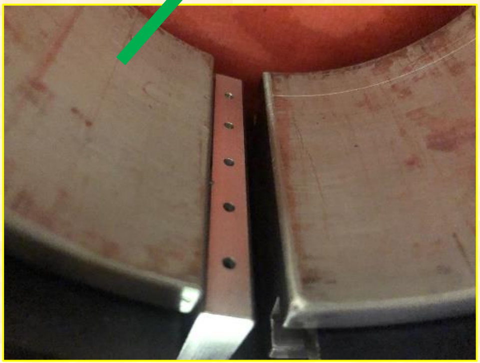


End seals with gap=0.165 mm
Single orifice with diameter = 4.4 mm

Evacuated bearing configuration

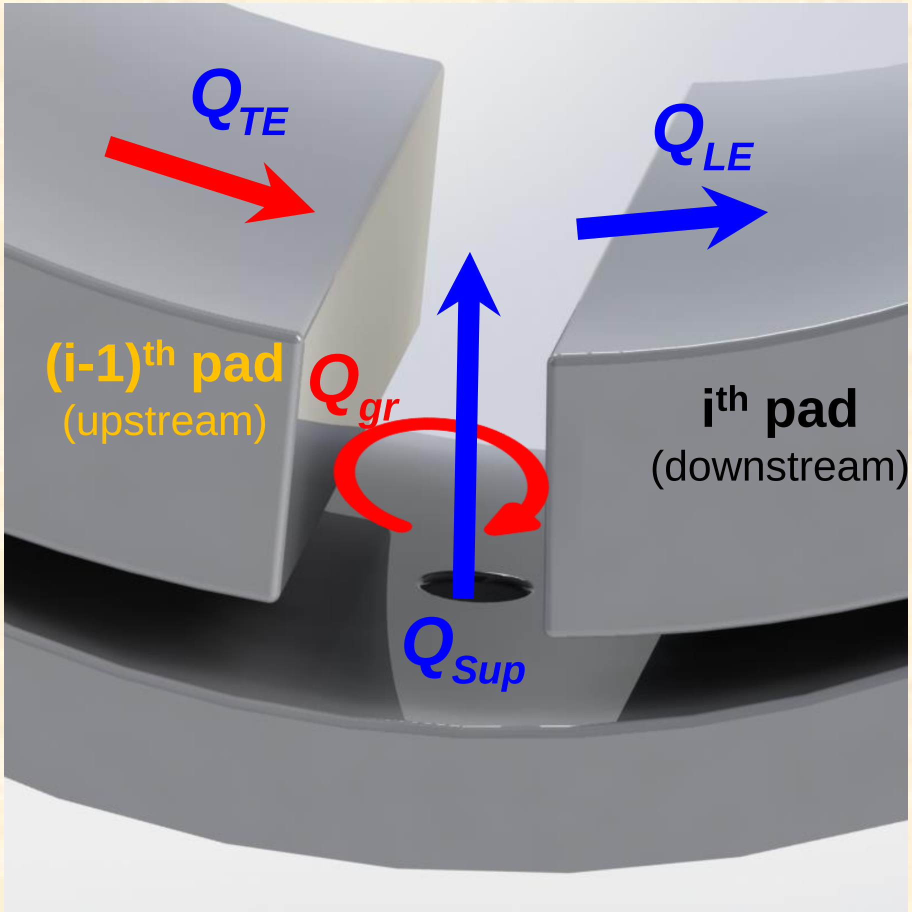






	End plates guide pads
Spray bar **5 mm from pad edge**	**Five orifices** diameter =5/64 inch (2 mm)

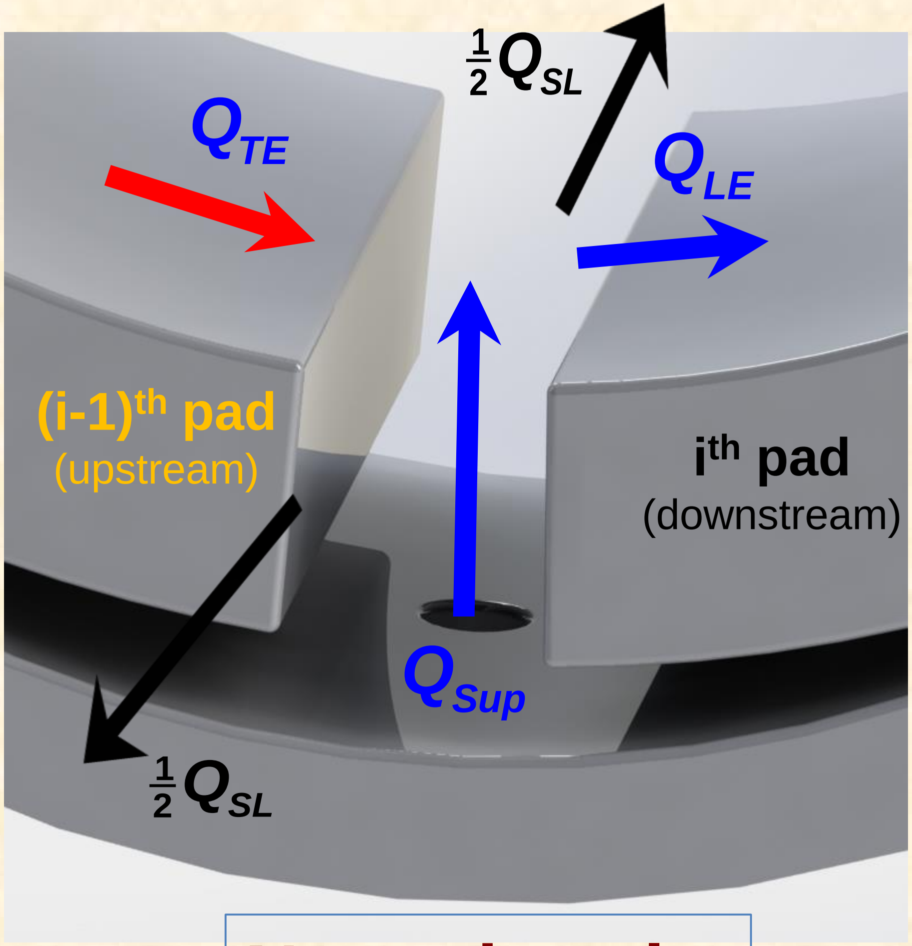
Flooded Bearing



End Seals accumulate oil in groove between pads – more churning losses and excess oil can cool pads.

Evacuated Bearing

No accumulation of oil in between pads – less churning losses and lesser heat convection to cool back of pads.



No end seals

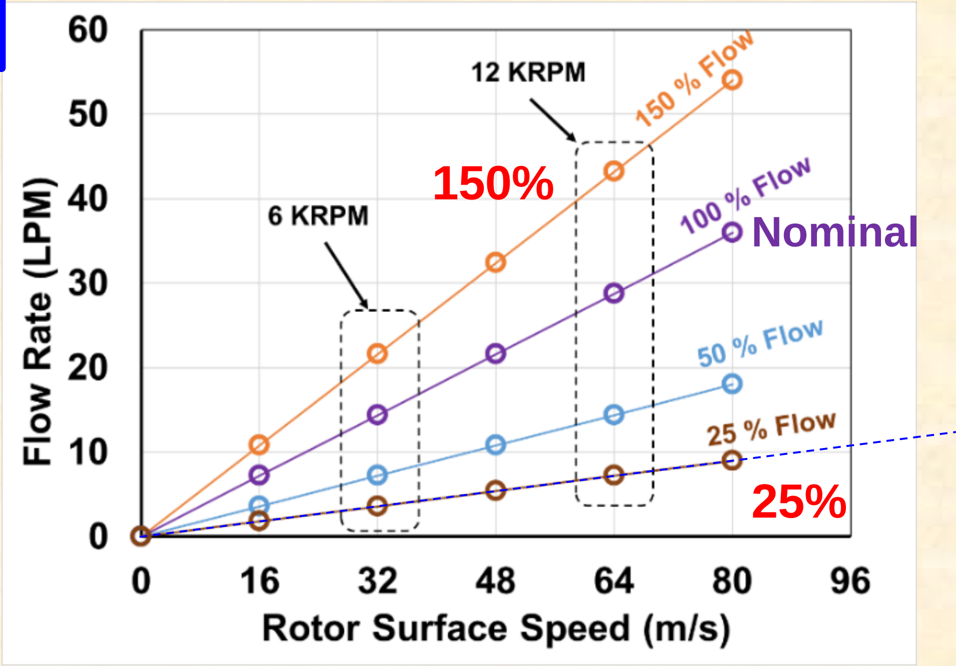
Oil supplied flow rate - theory

flow rate ~ shaft speed

VARY Flow from 150% → 100% (nominal) → 20% or less (if safe)

$$Q = N_p \tfrac{1}{2} \left(\tfrac{1}{2}\, \Omega D \right) L\, C_r\, (1-\lambda)$$

N_p = number of pads
Ω = shaft speed (rad/s)
D = shaft diameter (m)
L = bearing axial length (m)
C_r = bearing radial clearance (m)
λ = Oil mixing carry over coefficient.



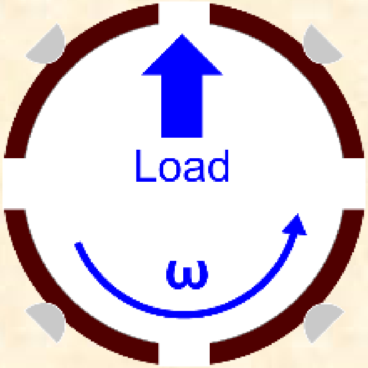
Low → **rotor speed** (krpm) → High

Tests at two shaft speeds

1. **6 krpm (32 m/s)** ➡ **~ 14. 4 LPM**
2. **12 krpm (64 m/s surface speed)** ➡ **~ 28.8 LPM**

6 & 12 krpm

Test Results



Load between pads (LBP)

Specific Load, *W/(LD)*	**345, 1,034, 2,068 kPa**

Flooded (with end seals)

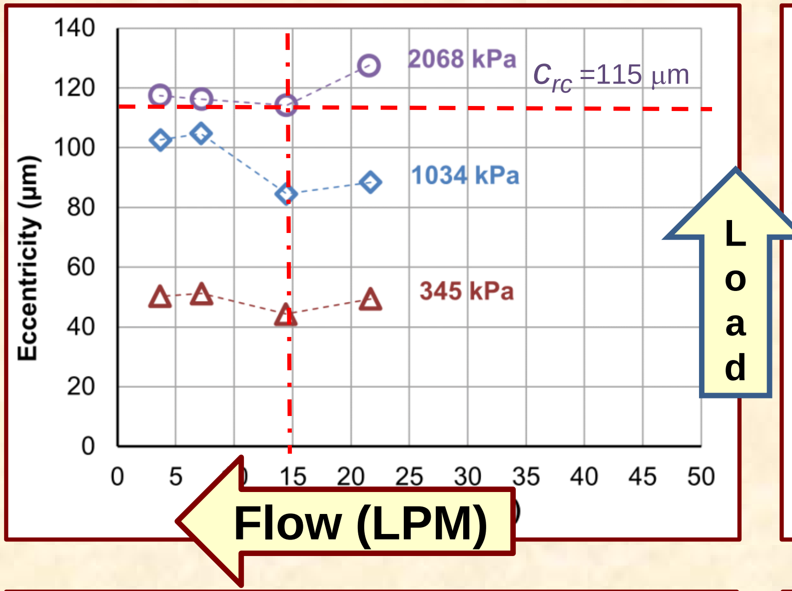
Evacuated (no end seals)

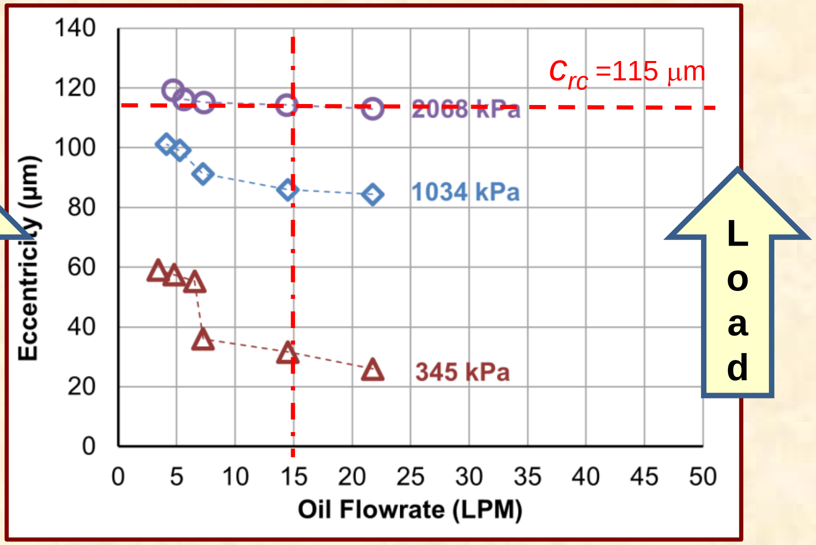
ISO VG46 inlet *T* = 60C

Eccentricity vs. speed vs. flow

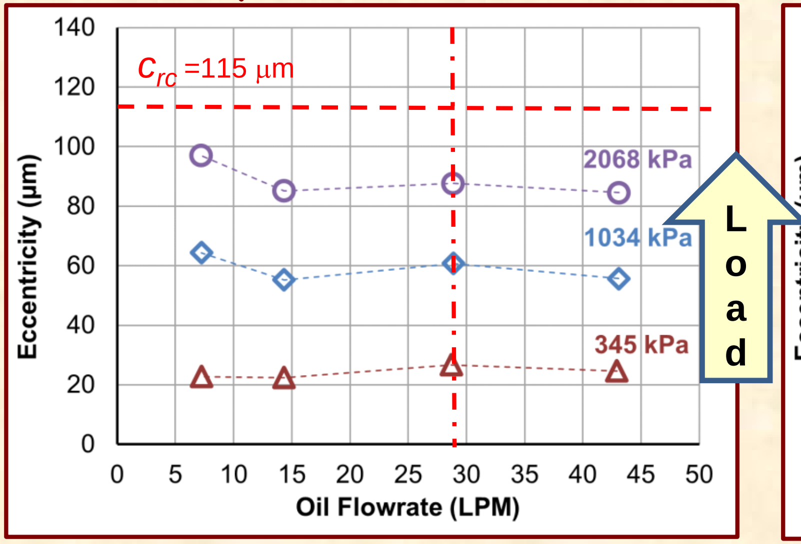
Flooded Bearing | **Evacuated Bearing**

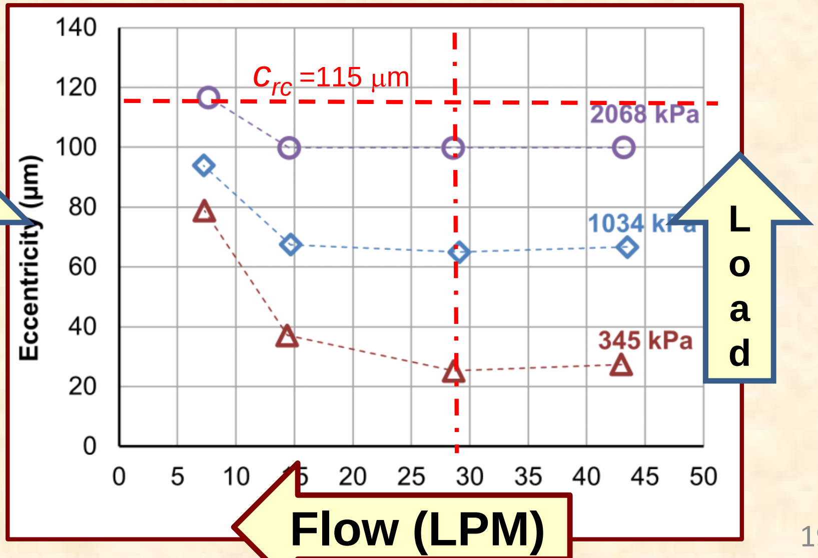
Speed = 6 krpm





Speed = 12 krpm



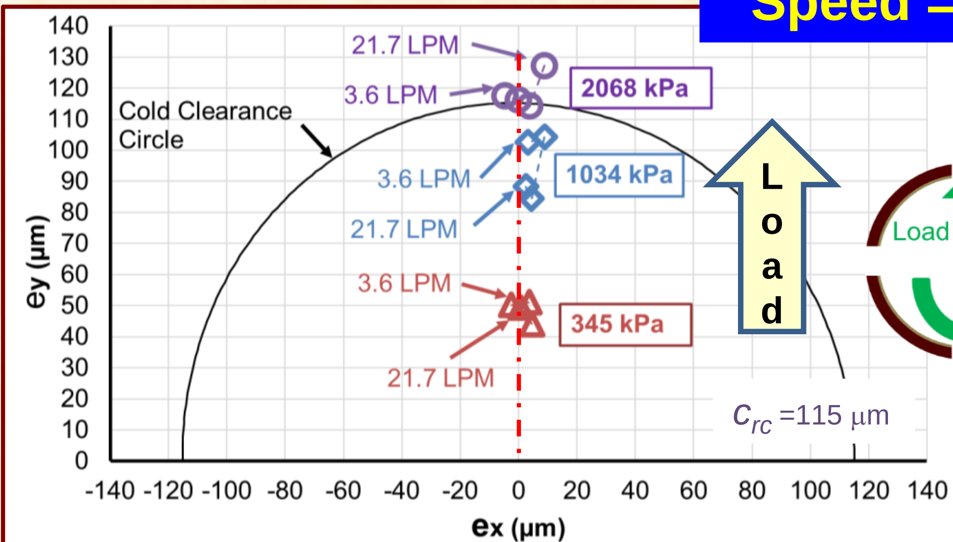


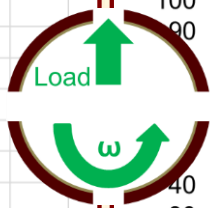
Journal locus vs. speed vs. flow

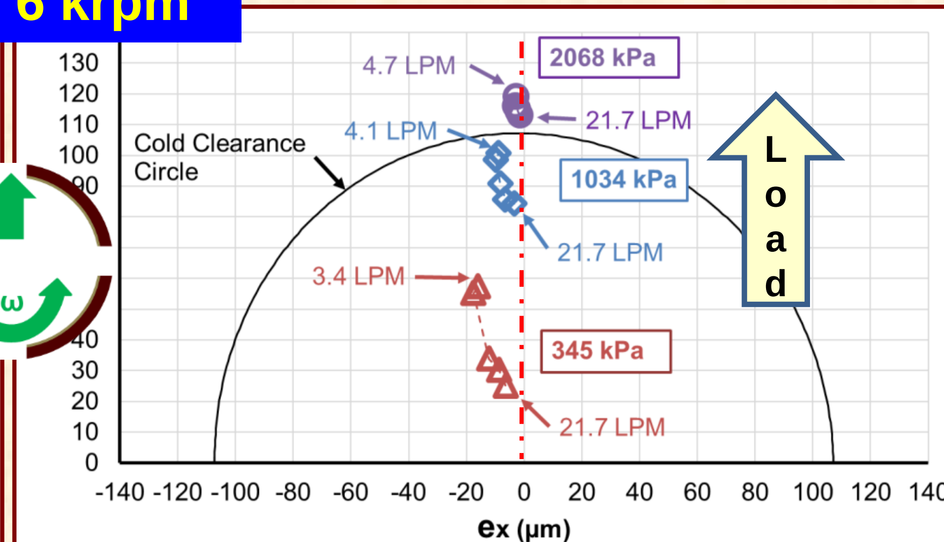
Flooded (orifice injection)

Evacuated (spray bar injection)

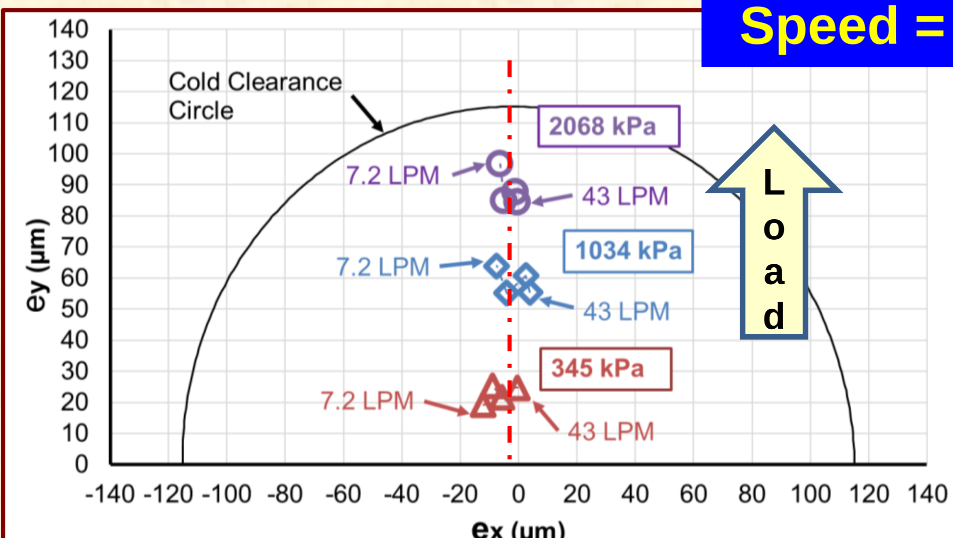
Speed = 6 krpm

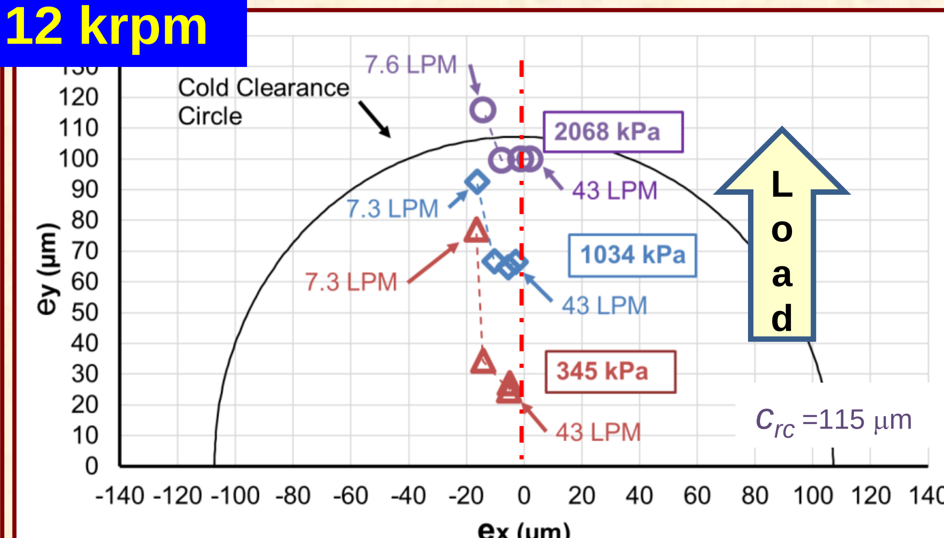






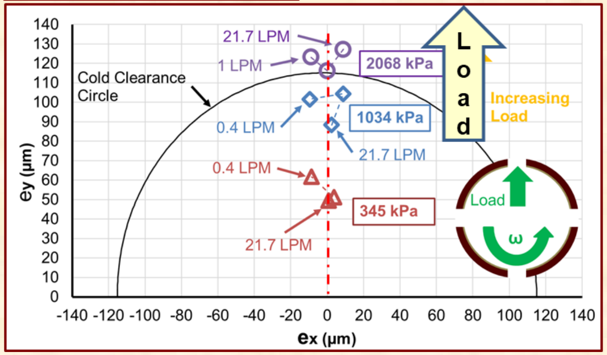
Speed = 12 krpm



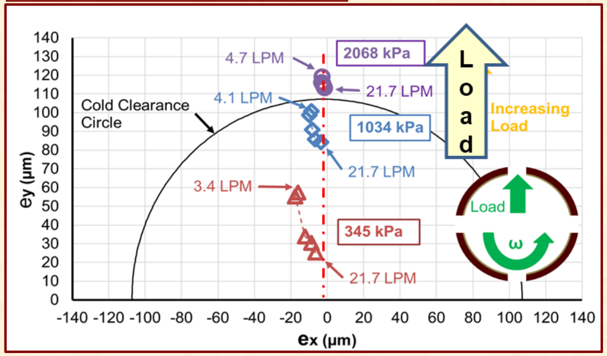


Eccentricity vs. speed vs. flow

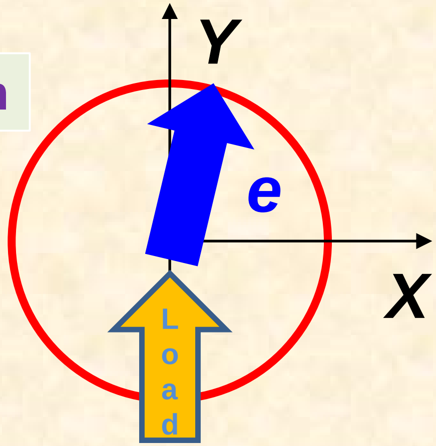
Flooded ends



Evacuated ends



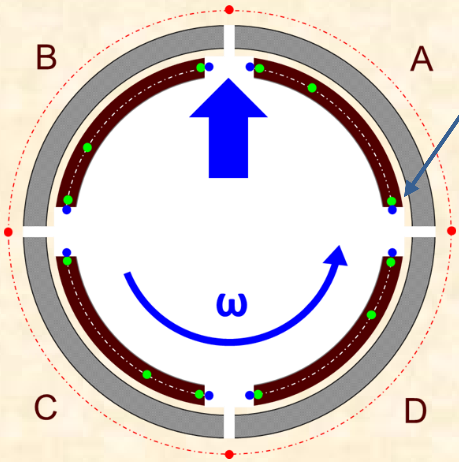
6 krpm



Eccentricity is nearly parallel to load direction and increases with load. Flooded bearing shows smaller eccentricity.

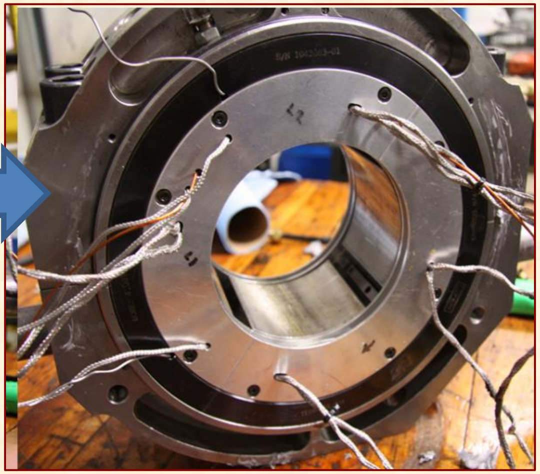
Journal eccentricity increases slightly as flow rate decreases → small impact on film thickness. Note side displacement as flow reduces.

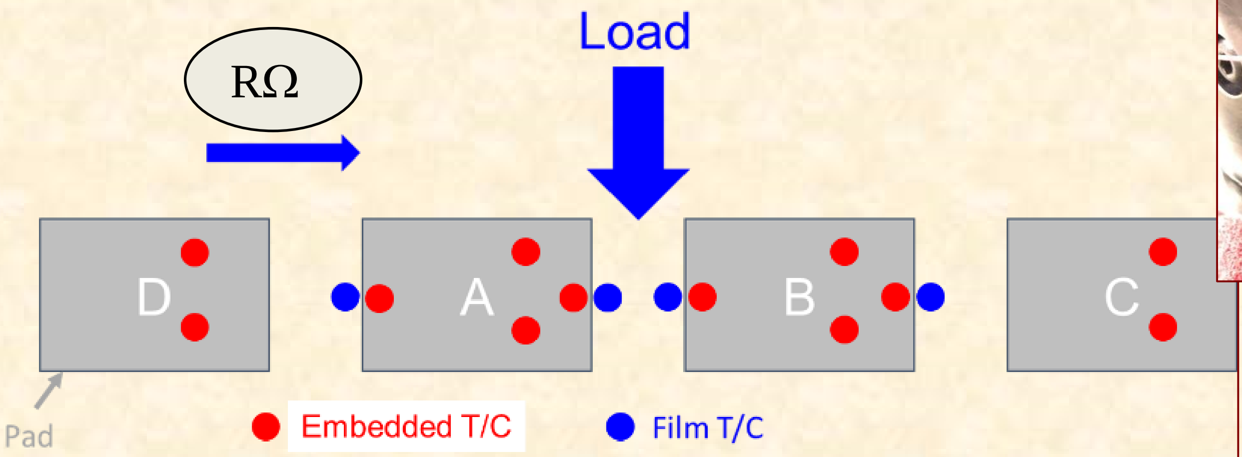
Test bearing – thermocouples



Thermocouples
in pads
and in oil supply
outer annulus

ISO VG46 inlet *T* = 60C



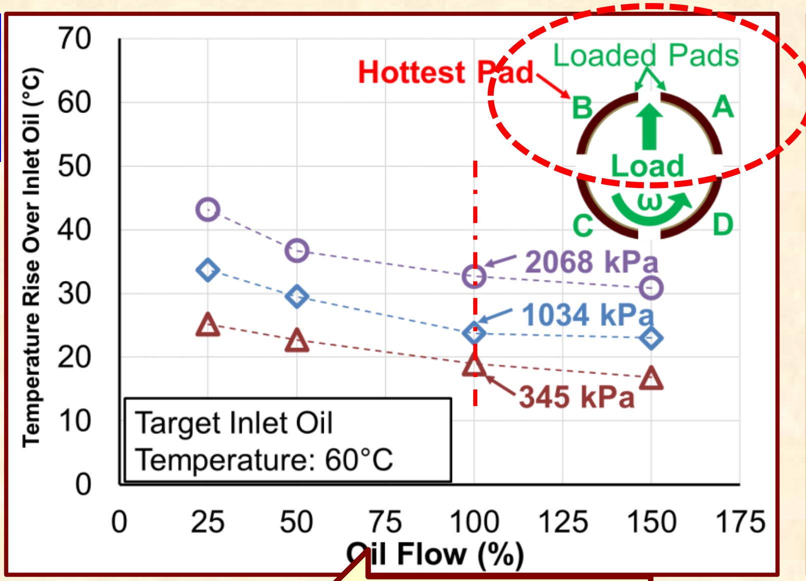


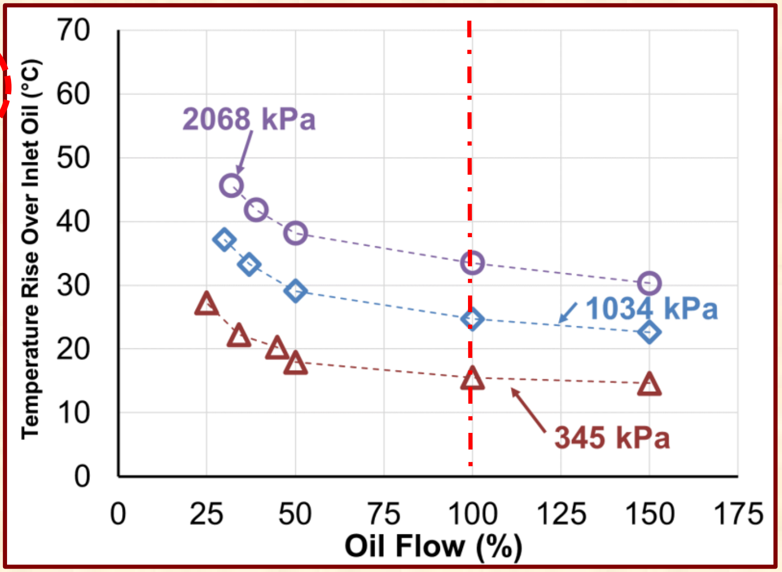
Maximum (Loaded) pad temperature rise

Flooded (orifice injection)

Evacuated (spray bar)

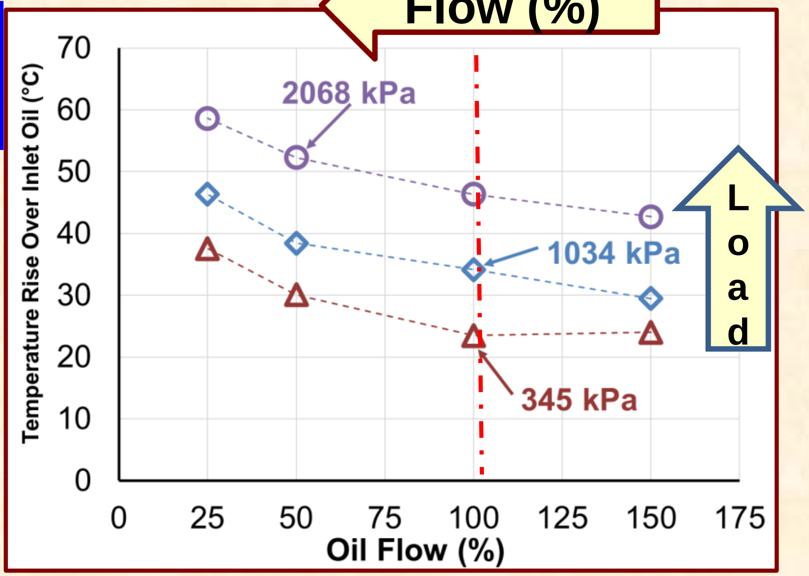
Speed = 6 krpm

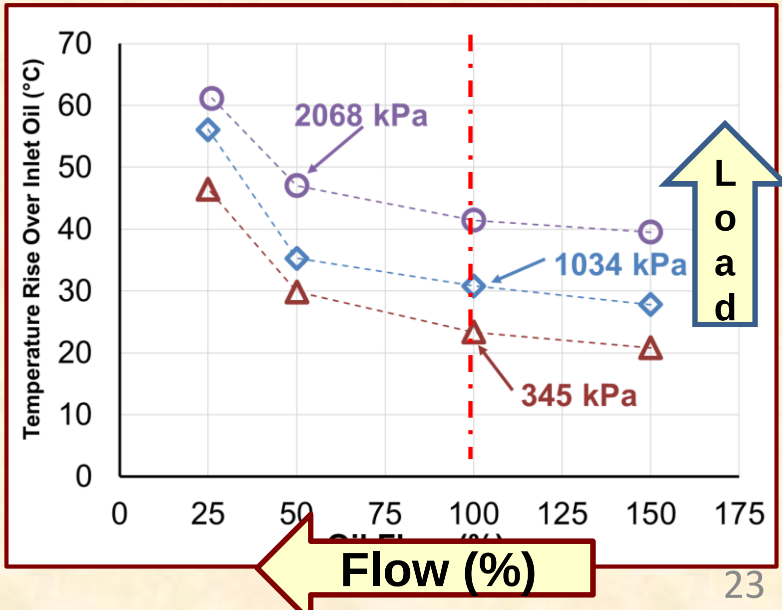




Flow (%)

Speed = 12 krpm





Babbitt Max T=130 C

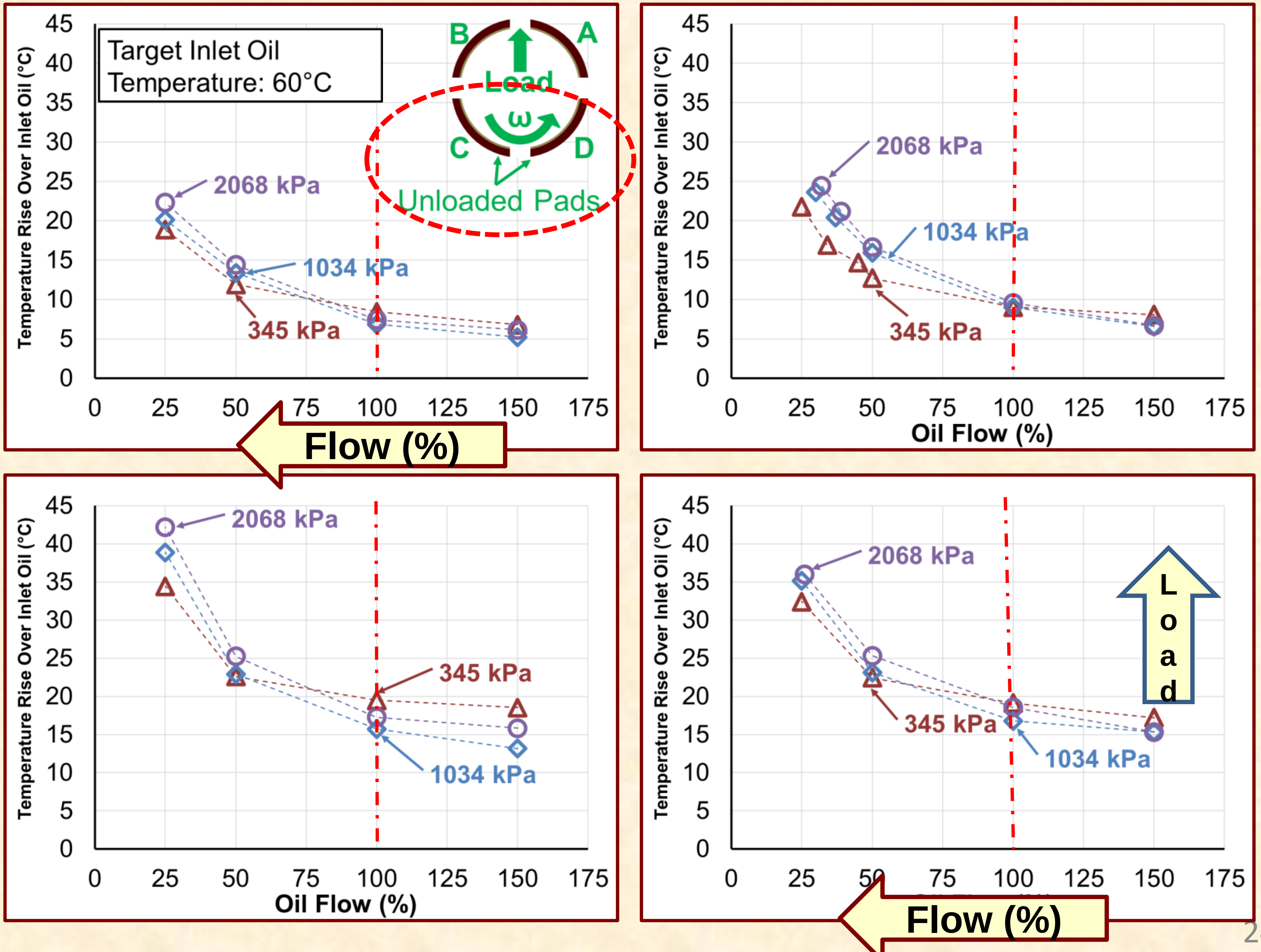
Maximum (unloaded) pad temperature rise

Flooded (orifice injection)

Evacuated (spray bar)

Speed = 6 krpm

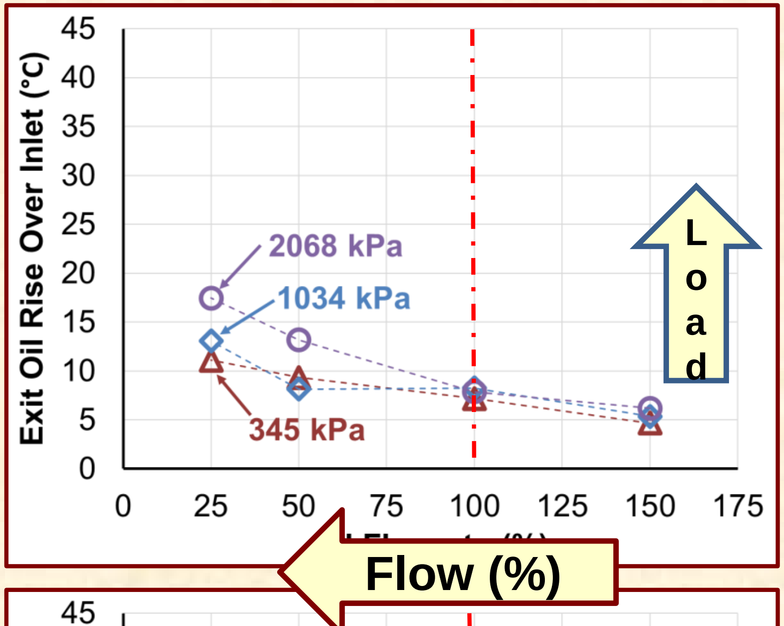
Speed = 12 krpm



Oil exit temperature rise

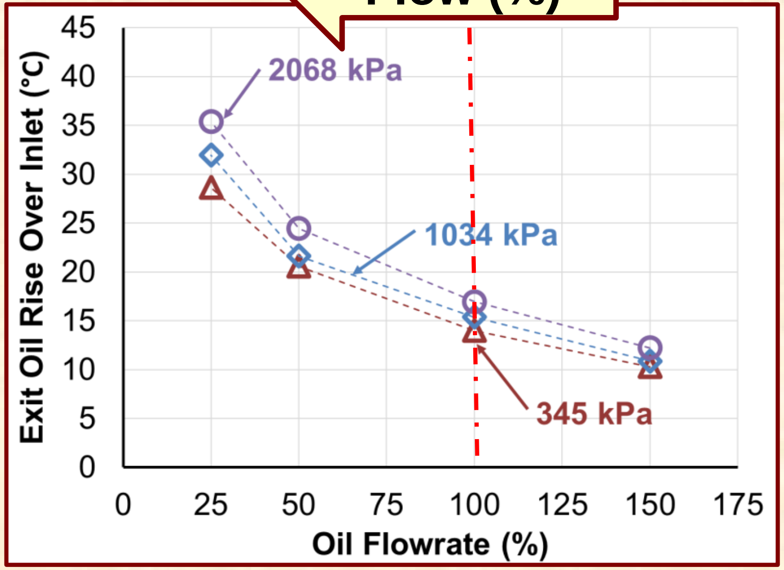
Flooded (orifice injection)

Evacuated (spray bar)

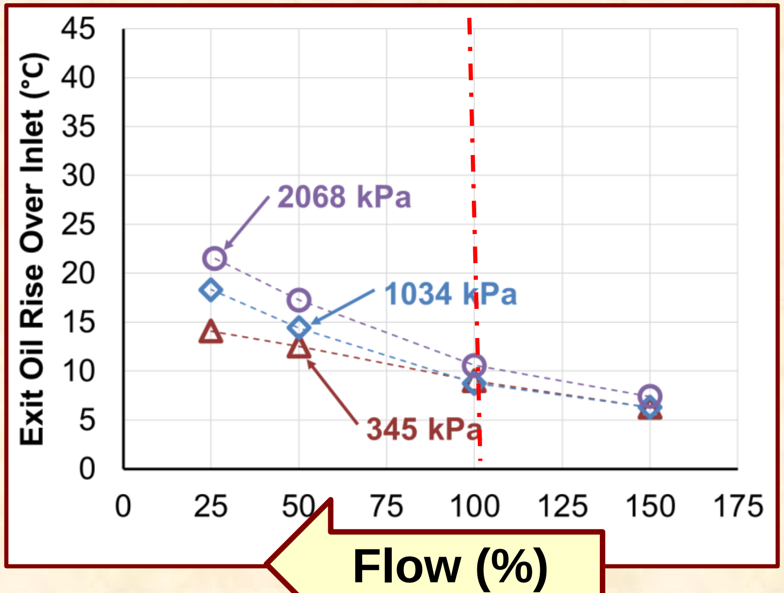


Speed = 6 krpm

Evacuated bearing (6 krpm) Temperatures recorded in oil return line (not bearing exit planes)



Speed = 12 krpm



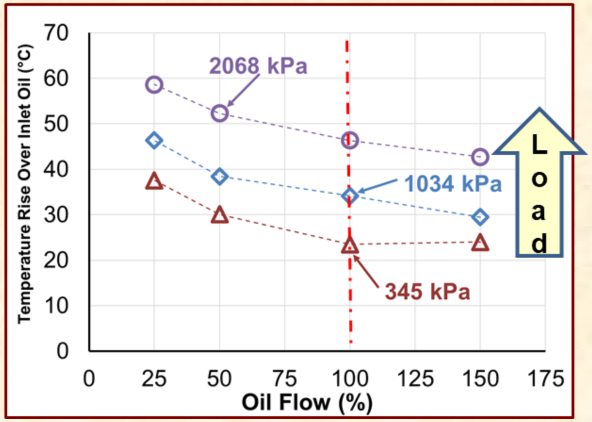
Maximum (Loaded) pad temperature rise

6 krpm

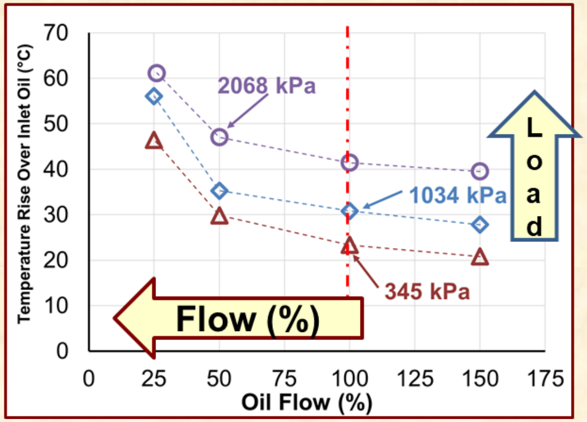
Babbitt Max DT=70 C

Speed = 12 krpm

Flooded (orifice injection)



Evacuated (spray bar)



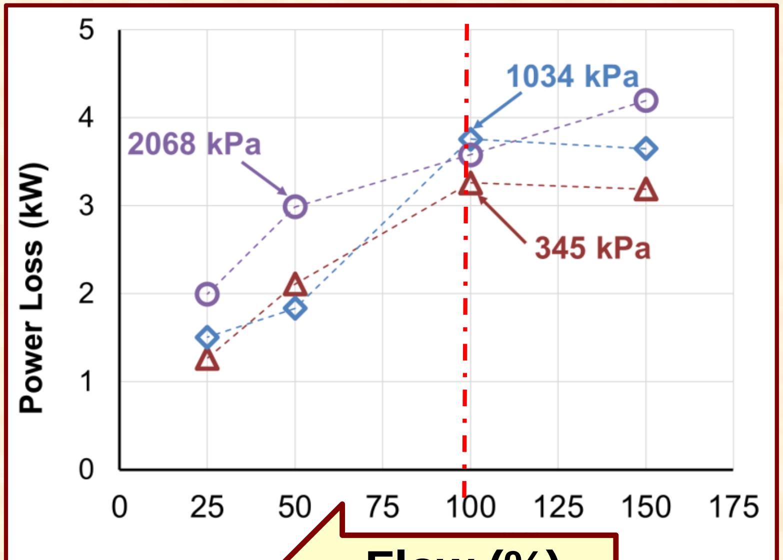
Evacuated B shows slightly larger pad temperatures (5 C) but much colder oil exit temperatures.

Oil exit temperature increases quickly as flow rate decreases. More pronounced effect in Flooded B.

Drag power loss

Flooded (orifice injection)

Evacuated (spray bar)



Speed = 6 krpm

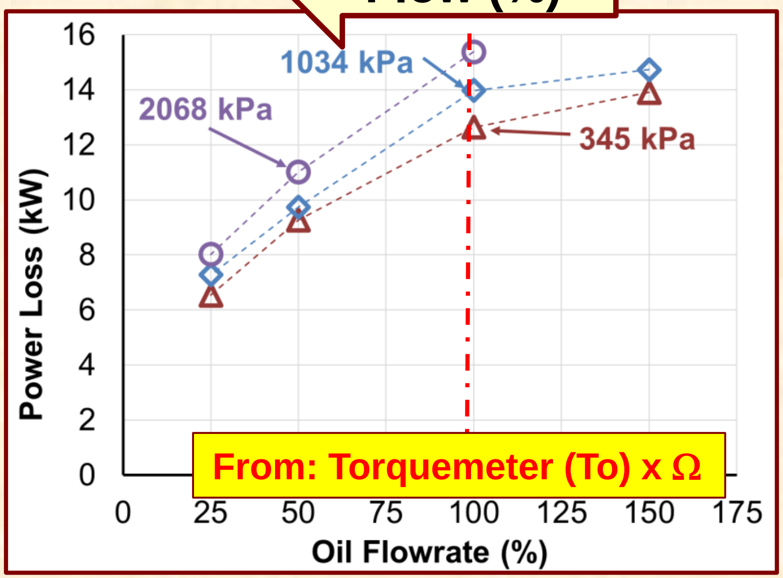
Evacuated B produces 30% less drag power loss

Flow (%)

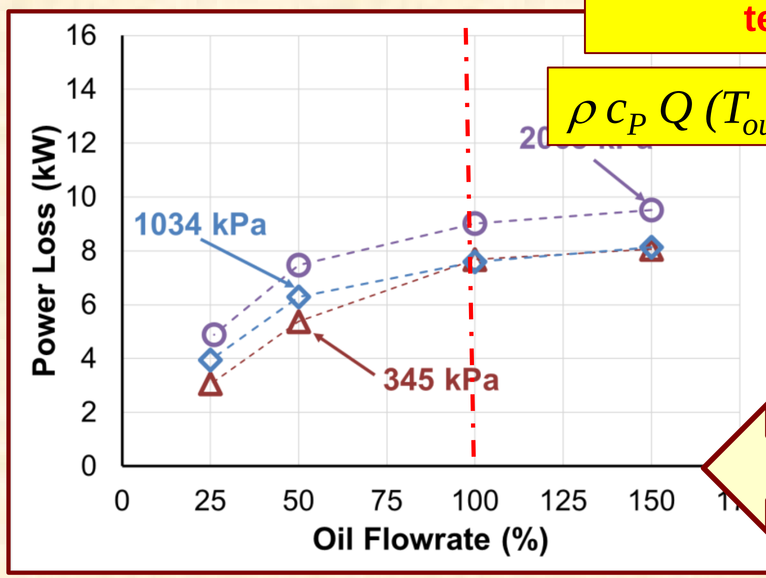
Speed = 12 krpm

Based on flow rate and temperature rise

$\rho \, c_P \, Q \, (T_{out} - T_{in}) \rightarrow P_{est}$

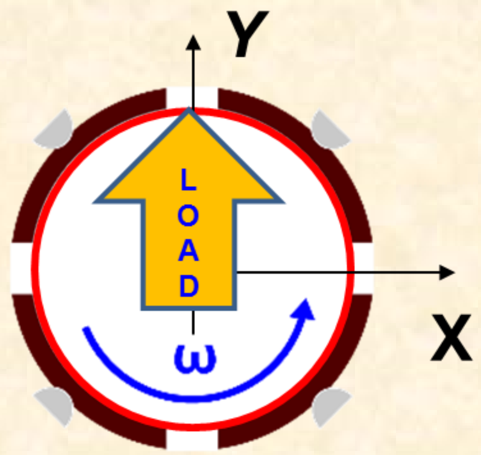


From: Torquemeter (To) x Ω



Flow (%)

Force Coefficients



Load between pads (LBP)

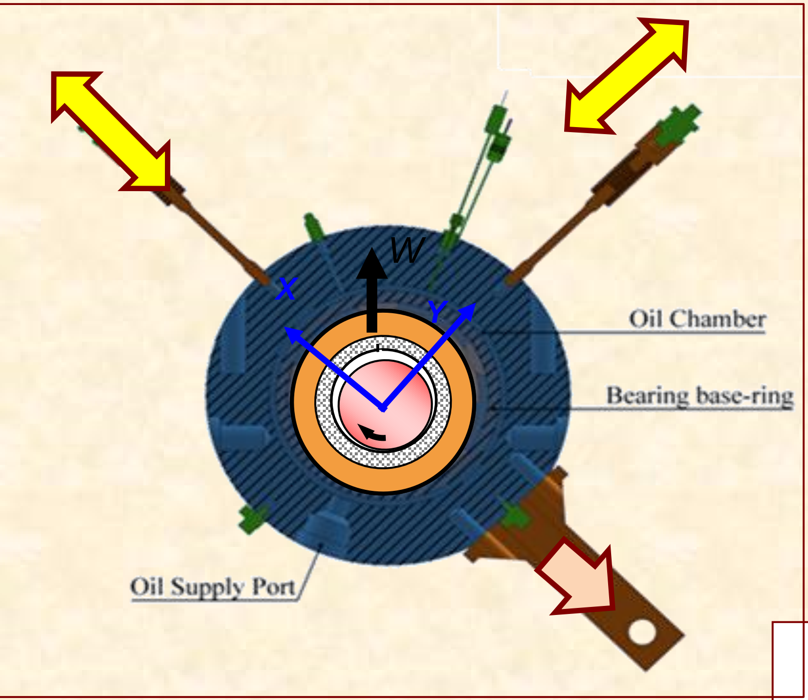
Shaft speed	6 & 12 krpm
Specific Load, *W/(LD)*	345, 1,034, 2,068 kPa

Flooded (with end seals)

Evacuated (no end seals)

ISO VG46 inlet *T* = 60C

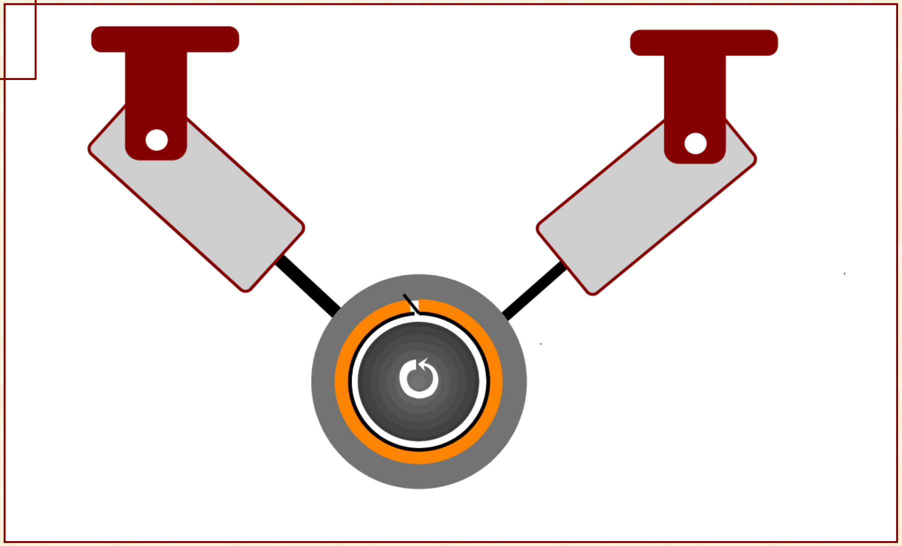
Dynamic load excitations



(X-Y 4.5 kN shakers)

Dynamic load: 1000 N

Pseudo random frequency functions 10 Hz to 250 Hz

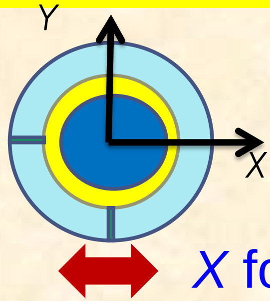


Bearing parameter identification

Step 1: Apply loads and measure bearings motion

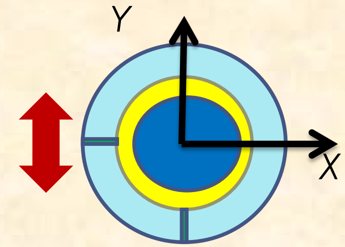
Apply forces with shakers → pseudo-random frequency

$$\mathbf{F}^1 = \mathrm{Re}\left(\begin{bmatrix} F_X{}^1 \\ 0 \end{bmatrix} e^{i\omega t}\right)$$



X force

$$\mathbf{F}^2 = \mathrm{Re}\left(\begin{bmatrix} 0 \\ F_Y{}^2 \end{bmatrix} e^{i\omega t}\right)$$



Y force

***ω* is a set of frequencies =(1, 2, 3,…, 17) x 9.77 Hz.**

Record bearing displacement z and acceleration a

$$\mathbf{z}^1 = \begin{bmatrix} x^1_{(t)} \\ y^1_{(t)} \end{bmatrix} = \begin{bmatrix} X^1 \\ Y^1 \end{bmatrix} e^{i\omega t} \qquad \mathbf{a}^1$$

$$\mathbf{z}^2 = \begin{bmatrix} x^2_{(t)} \\ y^2_{(t)} \end{bmatrix} = \begin{bmatrix} X^2 \\ Y^2 \end{bmatrix} e^{i\omega t} \qquad \mathbf{a}^2$$

EOM: Frequency domain

$$[\mathbf{K} + i\omega\mathbf{C} - \omega^2\mathbf{M}]\bar{\mathbf{z}} = \bar{\mathbf{F}} - M_S\bar{\mathbf{a}}$$

Find parameters:

$$\rightarrow \mathbf{H} = \mathbf{K} - \omega^2\mathbf{M} + i\omega\mathbf{C}$$

Estimations of complex stiffnesses

Step 2: Estimate dry structure parameters

NO lubricant

$$[\mathbf{K}_S - \omega^2 \mathbf{M}_S + i\omega \mathbf{C}_S]\bar{\mathbf{z}} = \bar{\mathbf{F}}$$

$$\rightarrow \mathbf{H}_S = \mathbf{K}_S - \omega^2 \mathbf{M}_S + i\omega \mathbf{C}_S$$

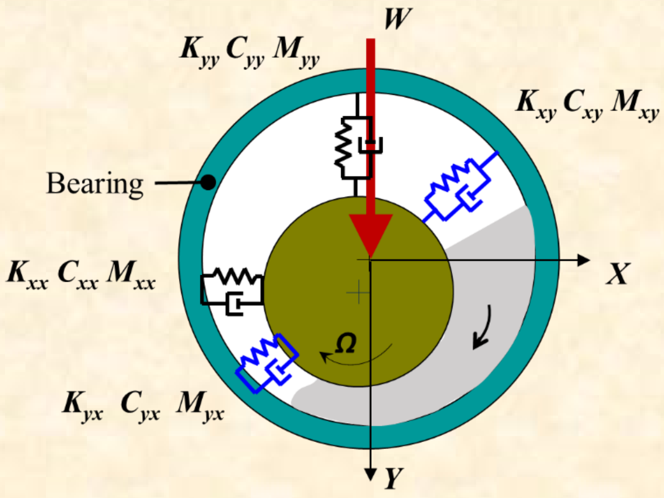
Step 3: Bearing force coefficients = Lubricated system – Dry system

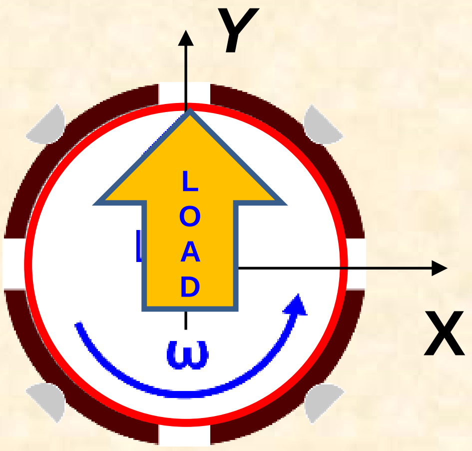
$$(K, C, M)_{bearing} = (K, C, M)_L - (K, C, M)_S$$

Bearing

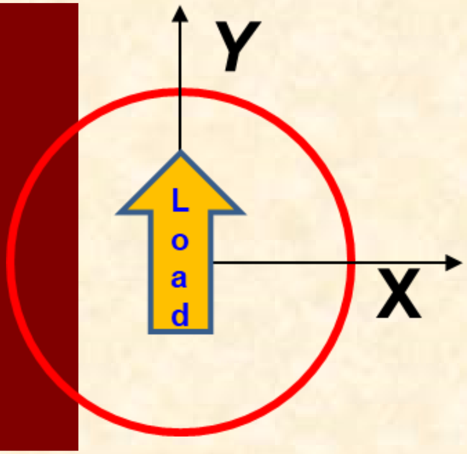
Test system (Lubricated)

Dry structure



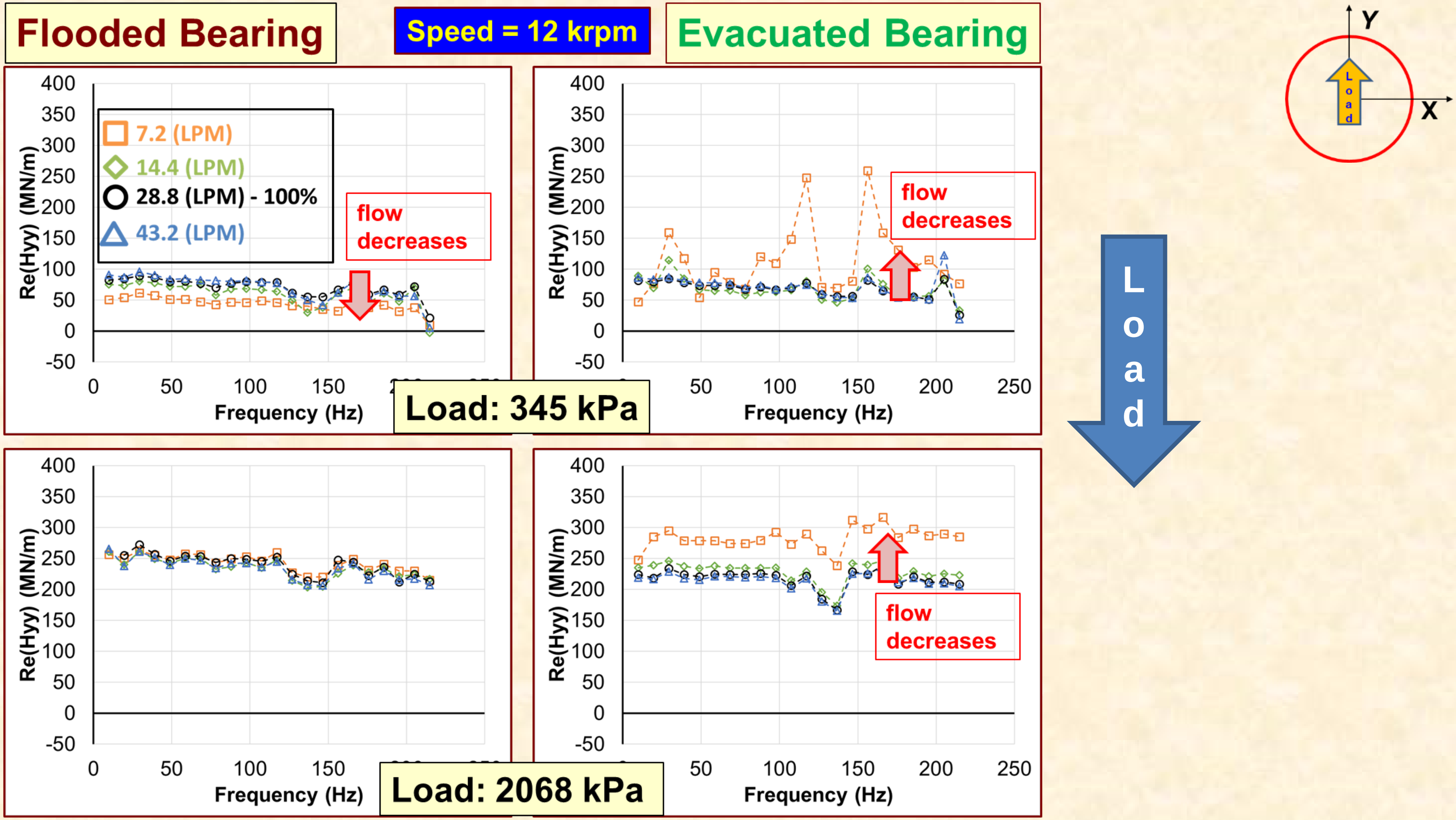


Real & imaginary parts of bearing complex stiffnesses



$$\rightarrow \mathbf{H} = \mathbf{H}_R + i\,\mathbf{H}_I = \begin{bmatrix} H_{xx} & H_{xy} \\ H_{yx} & H_{yy} \end{bmatrix}$$

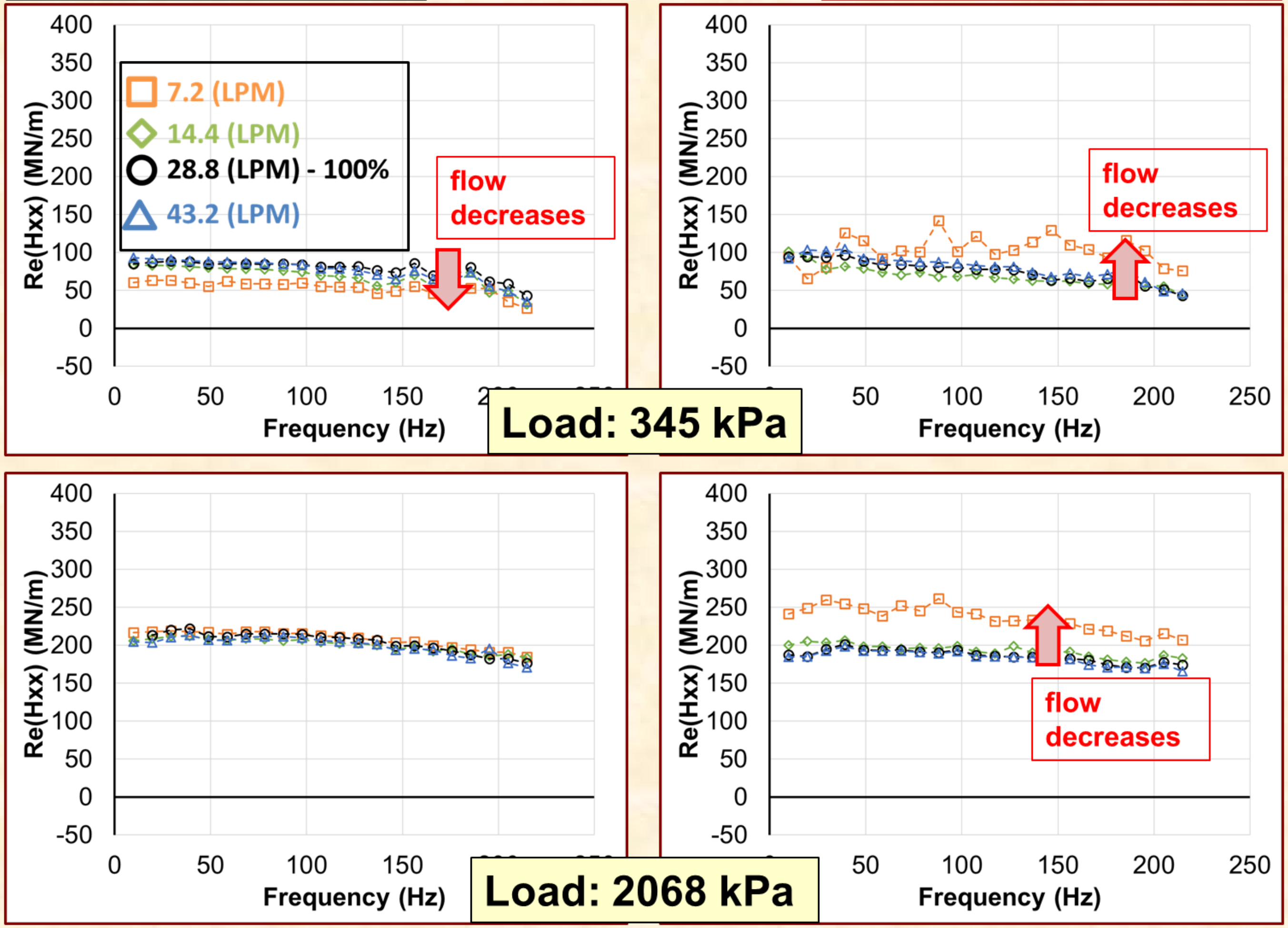
Real (H_{yy}) vs. frequency at 12 krpm & 2 loads

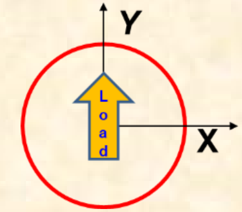


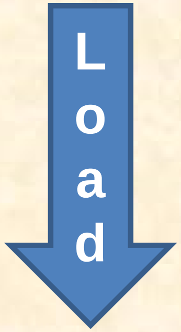
Small change with frequency, except for lowest load and lowest flow (25%) with <u>evacuated bearing</u>

Real (H_{xx}) vs. frequency at 12 krpm & 2 loads

Flooded Bearing | **Speed = 12 krpm** | **Evacuated Bearing**





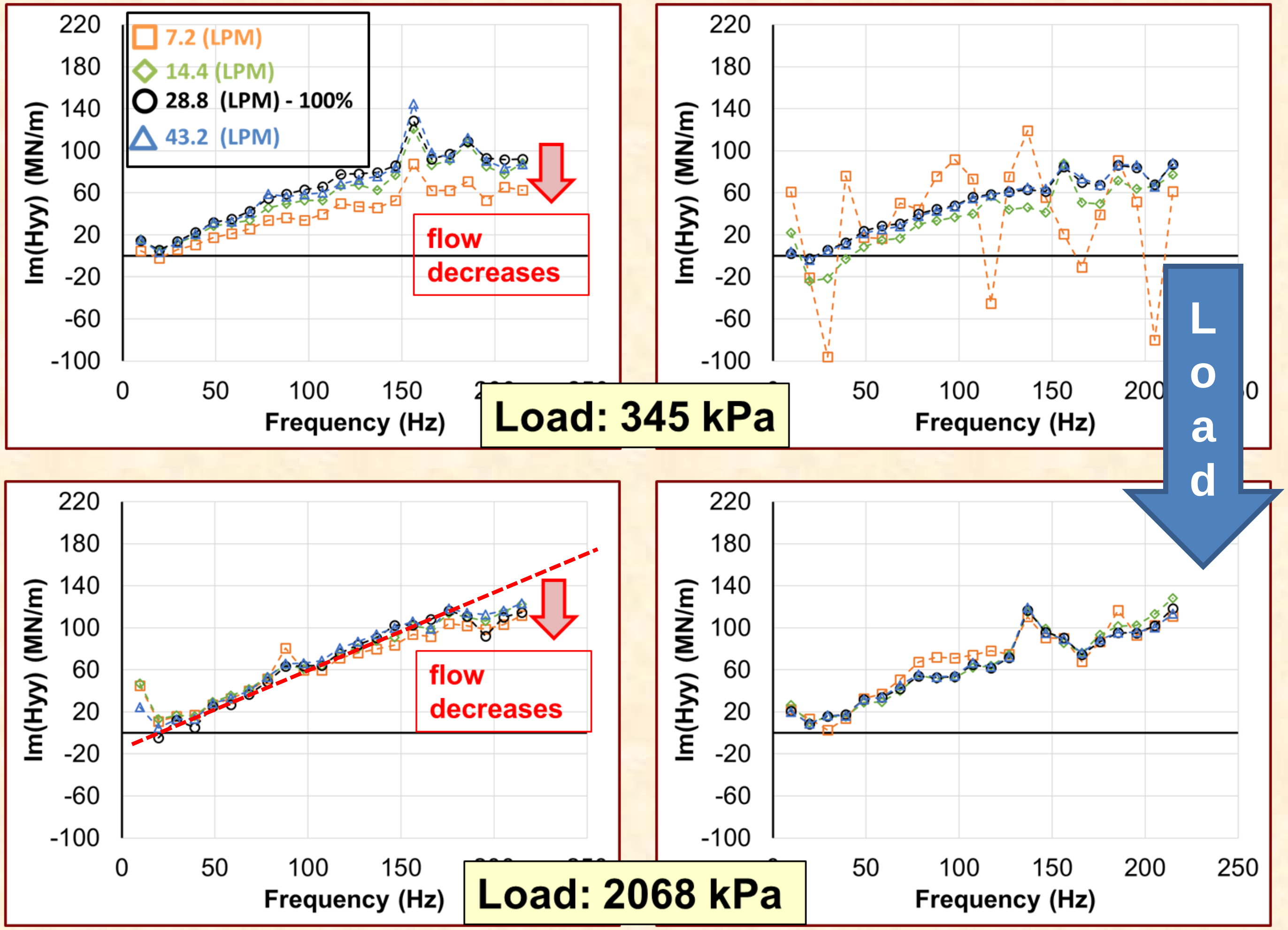


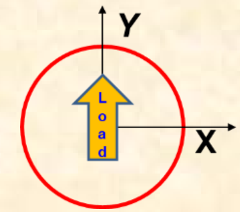
$H_{xx} < H_{yy}$,

Small change with frequency,

Ima (H_{yy}) vs. frequency at 12 krpm & 2 loads

Flooded Bearing | **Speed = 12 krpm** | **Evacuated Bearing**





H_{yy} **proportional to frequency → viscous damping to 150 Hz**

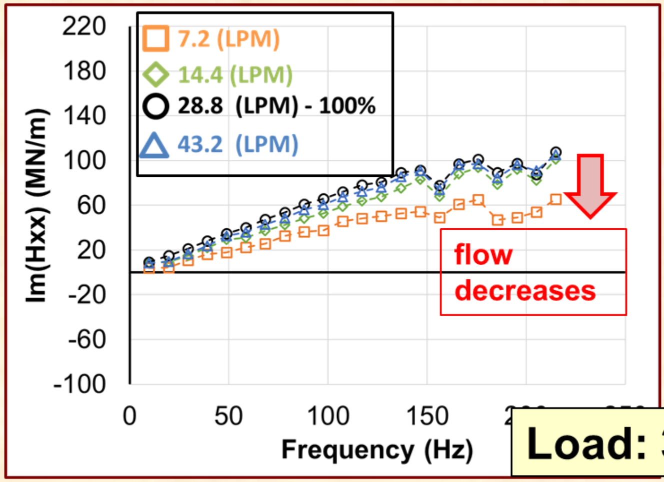
<u>Evacuated bearing:</u> odd data for lowest load and lowest flow (25%) – starved!

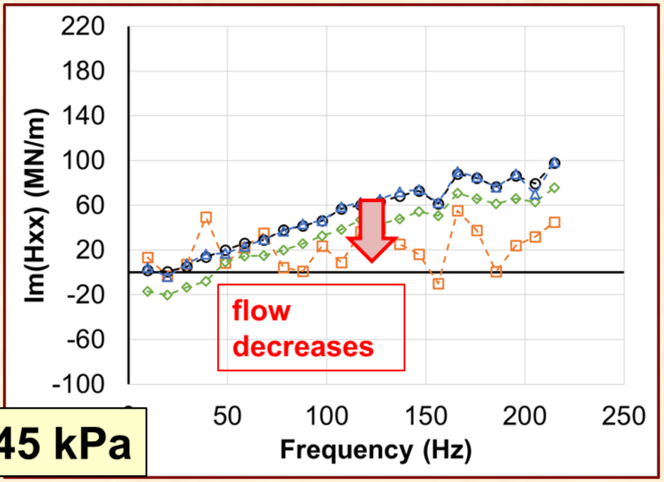
Ima (H_{xx}) vs. frequency at 12 krpm & 2 loads

Flooded Bearing

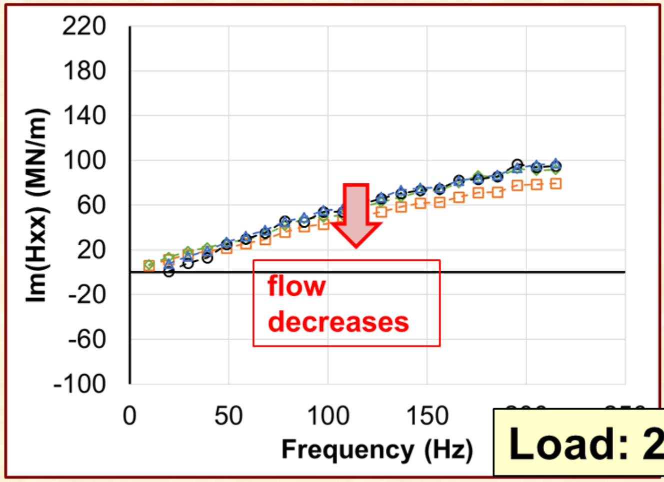
Speed = 12 krpm

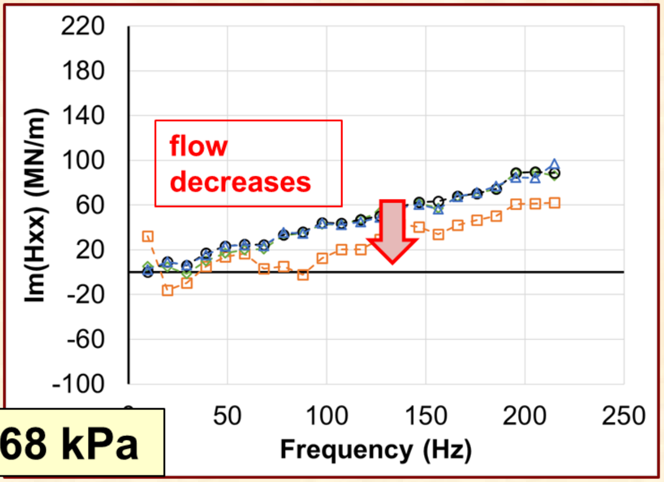
Evacuated Bearing



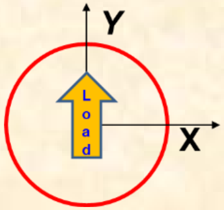


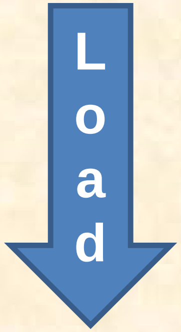
Load: 345 kPa





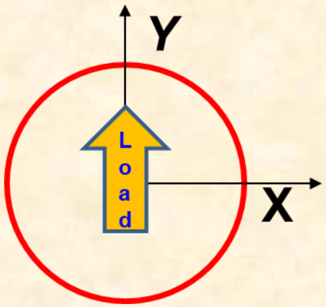
Load: 2068 kPa





$H_{xx} < H_{yy}$. Mostly proportional to frequency → viscous damping

Curve fits of complex stiffnesses



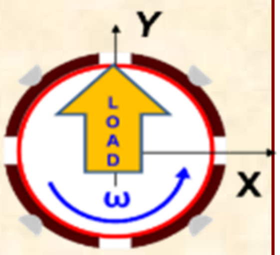
$$\mathbf{H} \rightarrow \left[\mathbf{K} - \omega^2 \mathbf{M} \right] + i \left(\omega \mathbf{C} \right)$$

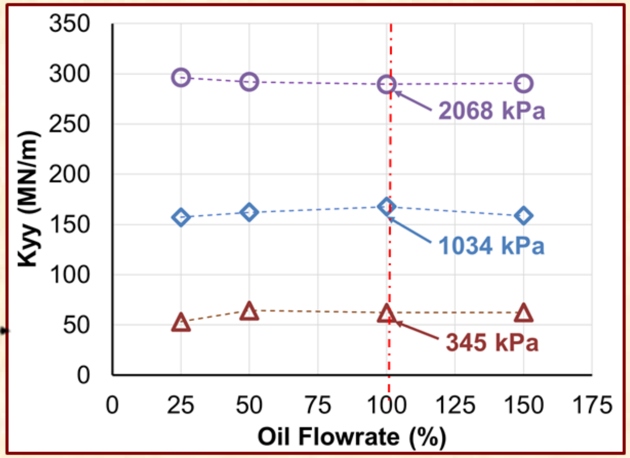
Stiffness coefficients *K*

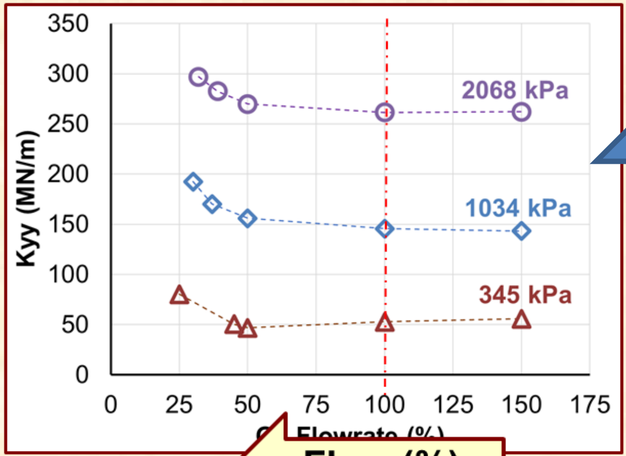
K_{yy} Direct stiffness vs. flow & load

Flooded Bearing | **Evacuated Bearing**

Speed = 6 krpm



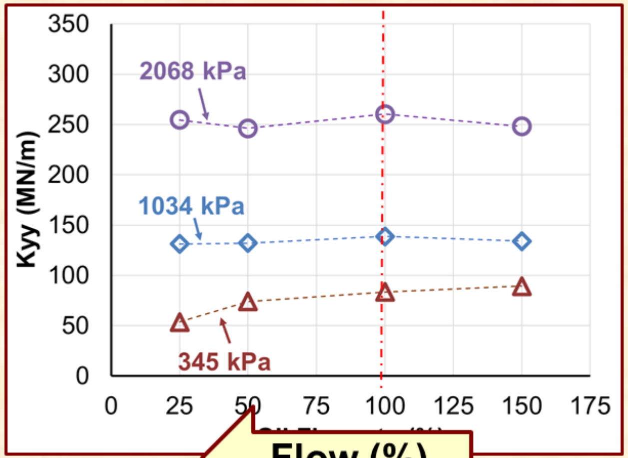


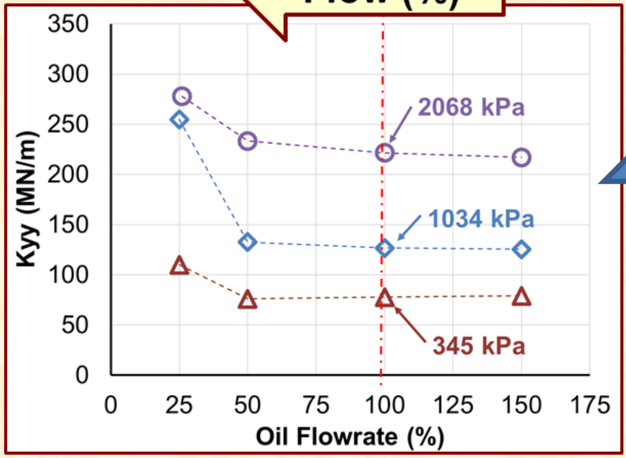


Load

Flow (%)

Speed = 12 krpm





Load

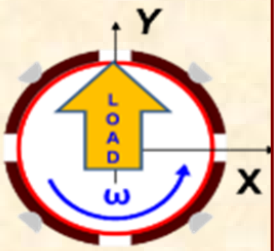
Flow (%)

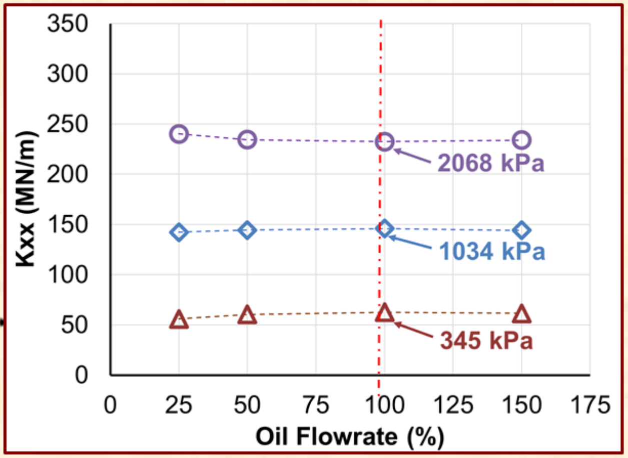
K_{YY} is mainly a function of load.
For largest load: K_{YY} (flooded) > K_{YY} (evacuated)

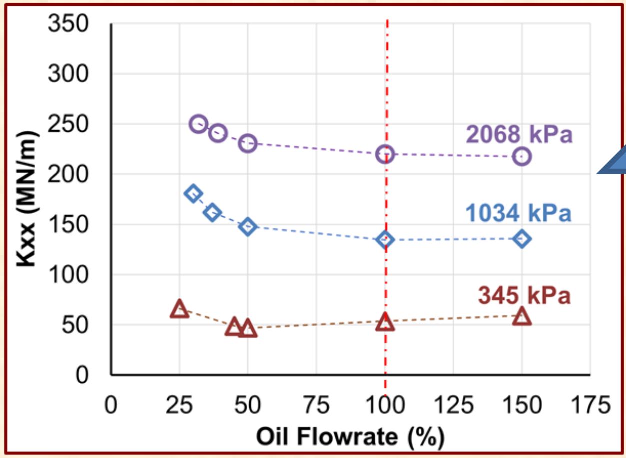
K_{XX} Direct stiffness vs. flow & load

Flooded Bearing | **Evacuated Bearing**

Speed = 6 krpm

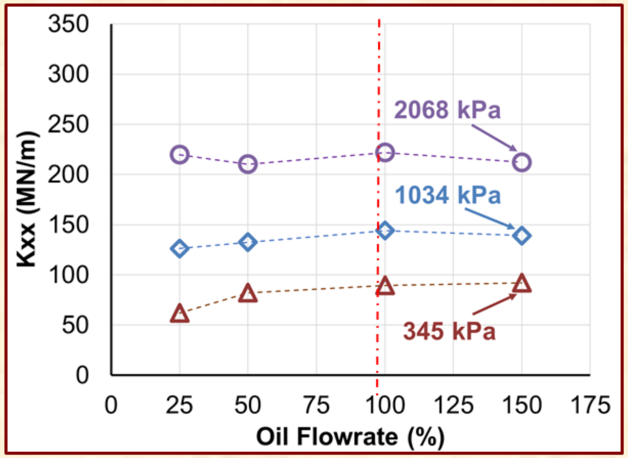


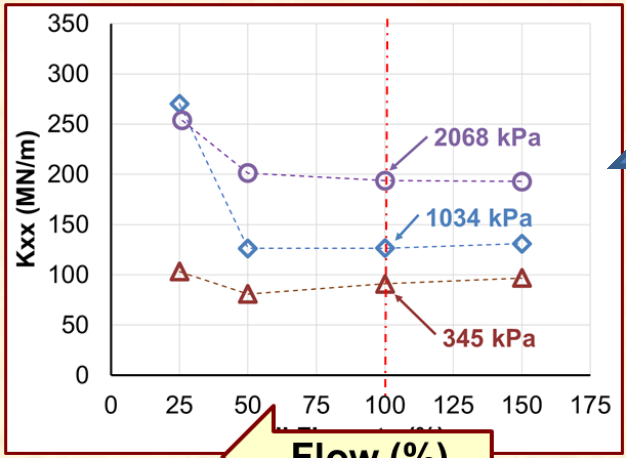




Load

Speed = 12 krpm



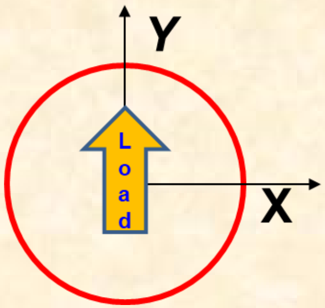


Load

Flow (%)

At largest load, K_{XX} decreases compared to K_{YY}.

Evacuated B has slightly lesser stiffness.



$$\mathbf{H} \rightarrow \left[\mathbf{K} - \omega^2 \mathbf{M} \right] + i \left(\omega \mathbf{C} \right)$$

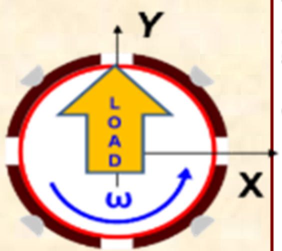
Damping coefficients *C*

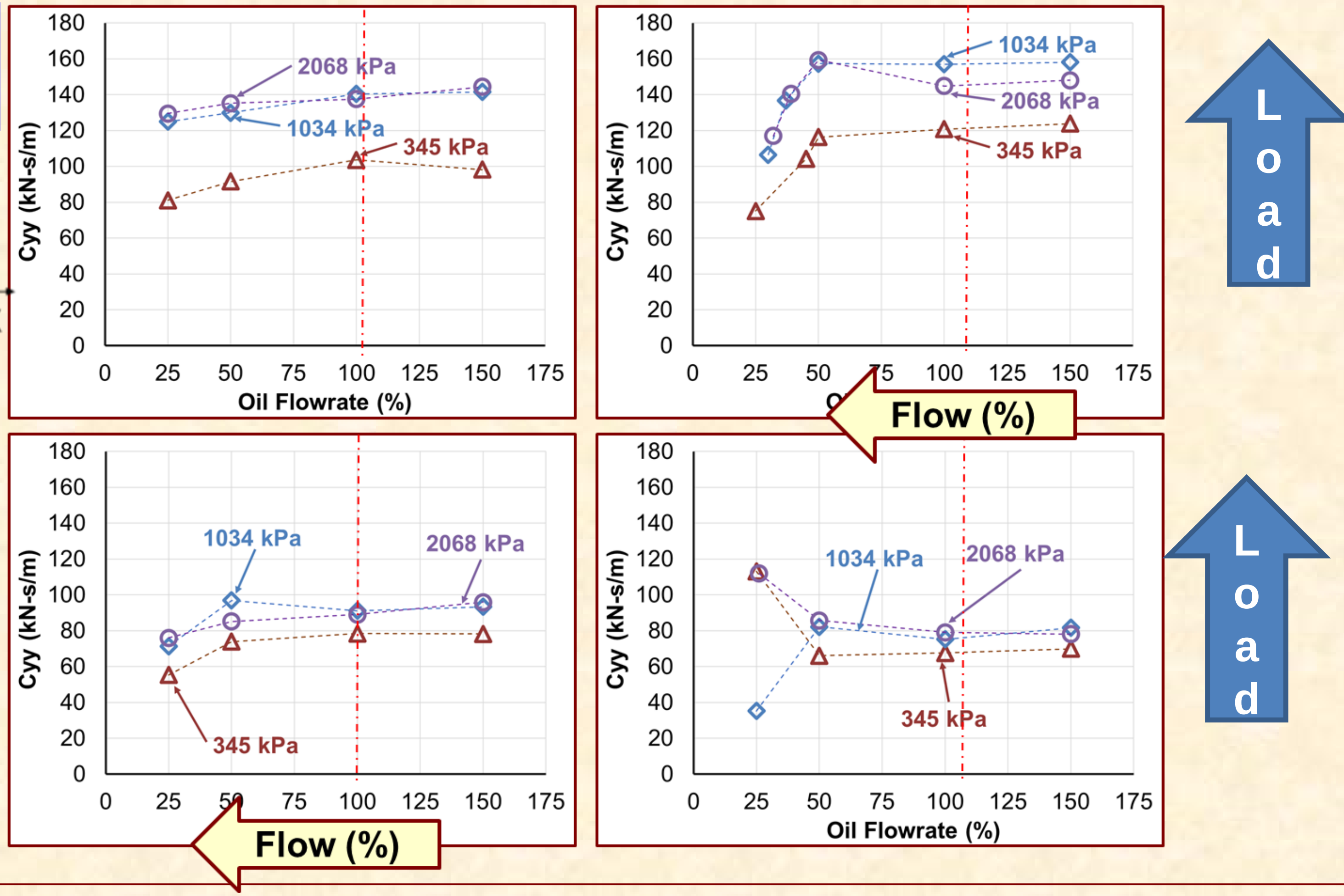
C_{yy} Direct damping vs. flow & load

Flooded Bearing | **Evacuated Bearing**

Speed = 6 krpm

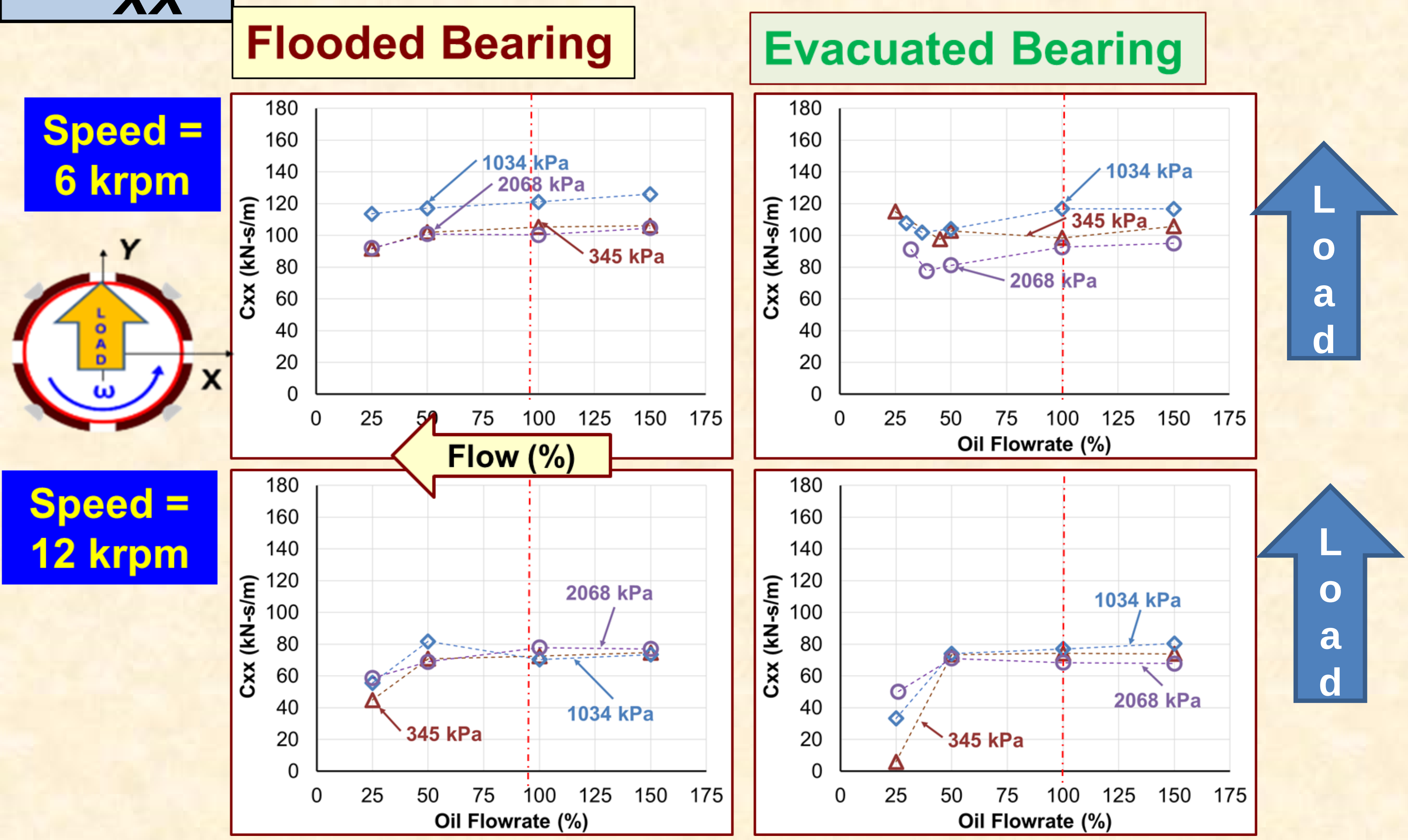
Speed = 12 krpm



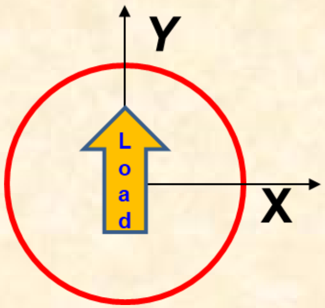


C_{YY} **decreases as flow rate drops.** **<u>At 6 krpm</u>: Evacuated B shows large drop in damping for lowest flow.**

C_{XX} Direct damping vs. flow & load



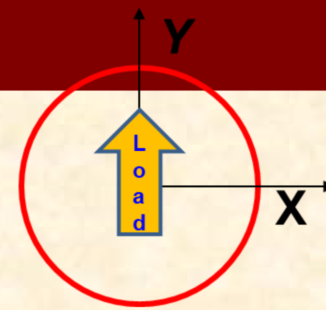
Evacuated B produces lesser direct damping & w/o + reduction as flow decreases (except at 12 krpm).



$$\mathbf{H} \rightarrow \left[\mathbf{K} - \omega^2 \mathbf{M}\right] + i\left(\omega \mathbf{C}\right)$$

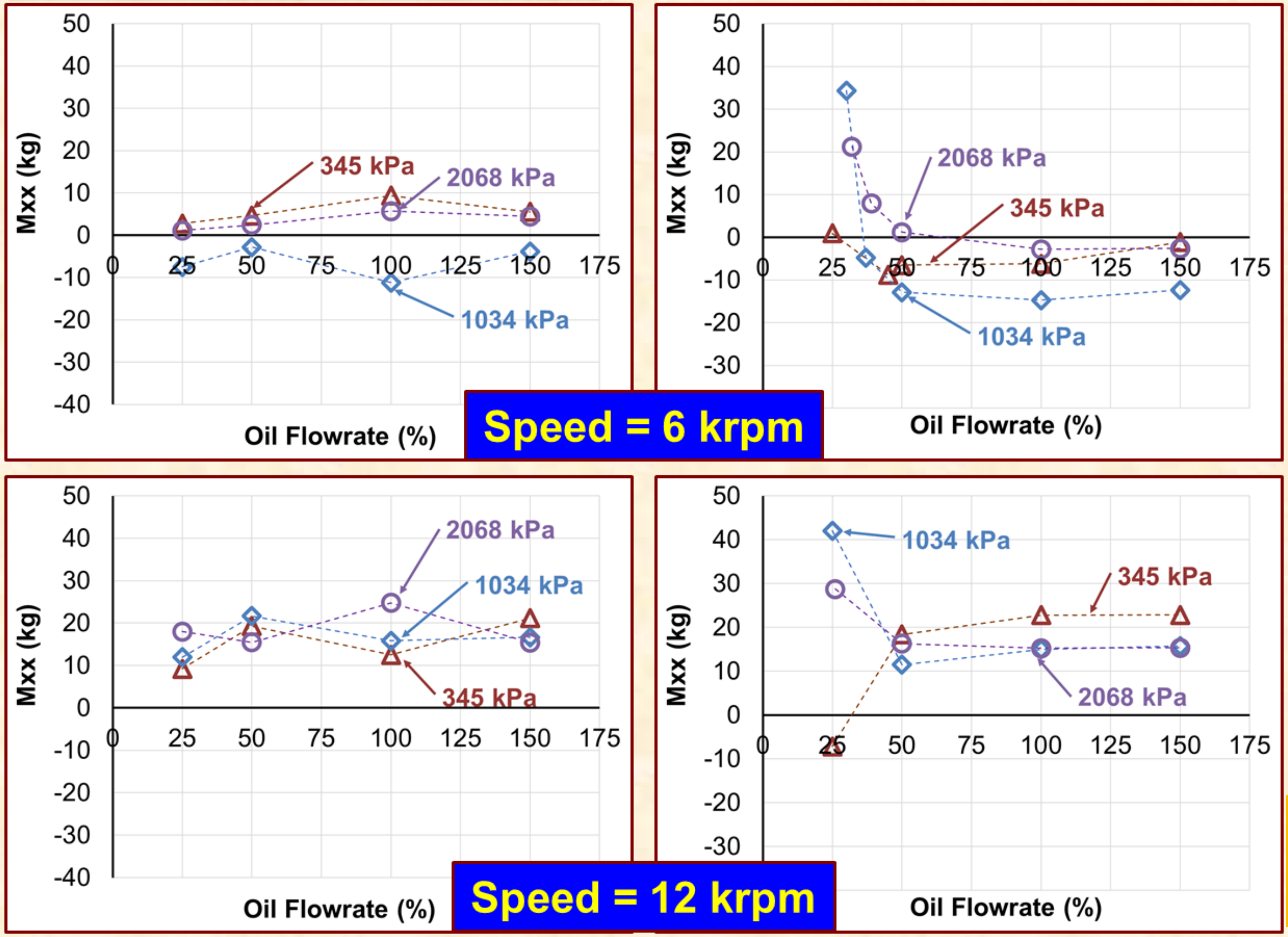
Virtual mass coefficients *M*

M_{XX} Virtual mass coefficients



Flooded (orifice injection)

Evacuated (spray bar)



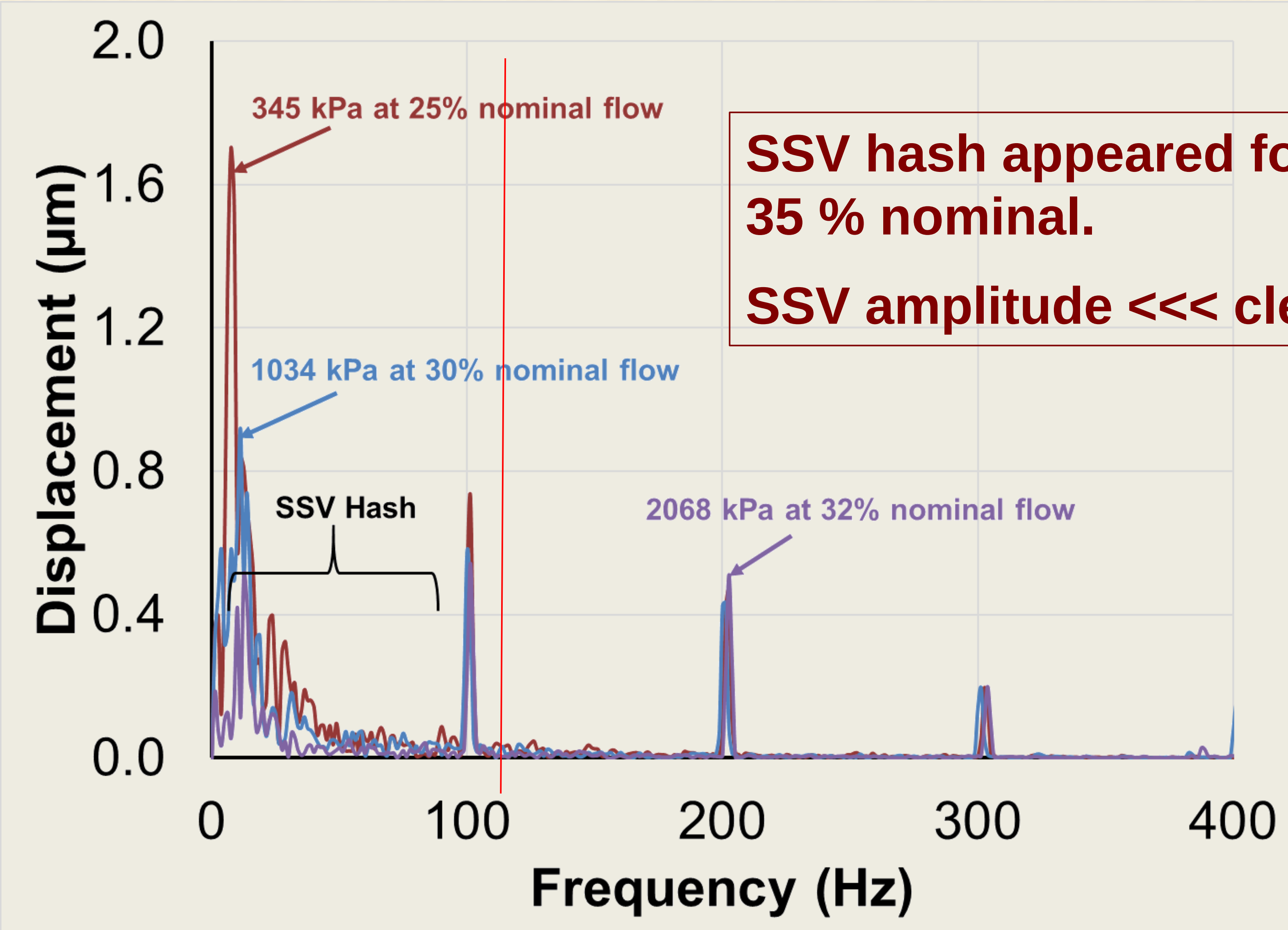
$$\left[\mathbf{K}-\omega^2\mathbf{M}\right]\sim\mathbf{K}$$

Virtual masses are same size as bearing cartridge mass.
However, effect of (-Mω^2) on dynamic stiffness is small.

Subsynchronous shaft vibrations

Typical in evacuated bearings operating with low flow (starved) and under a low load.

Evacuated bearing at 6 krpm



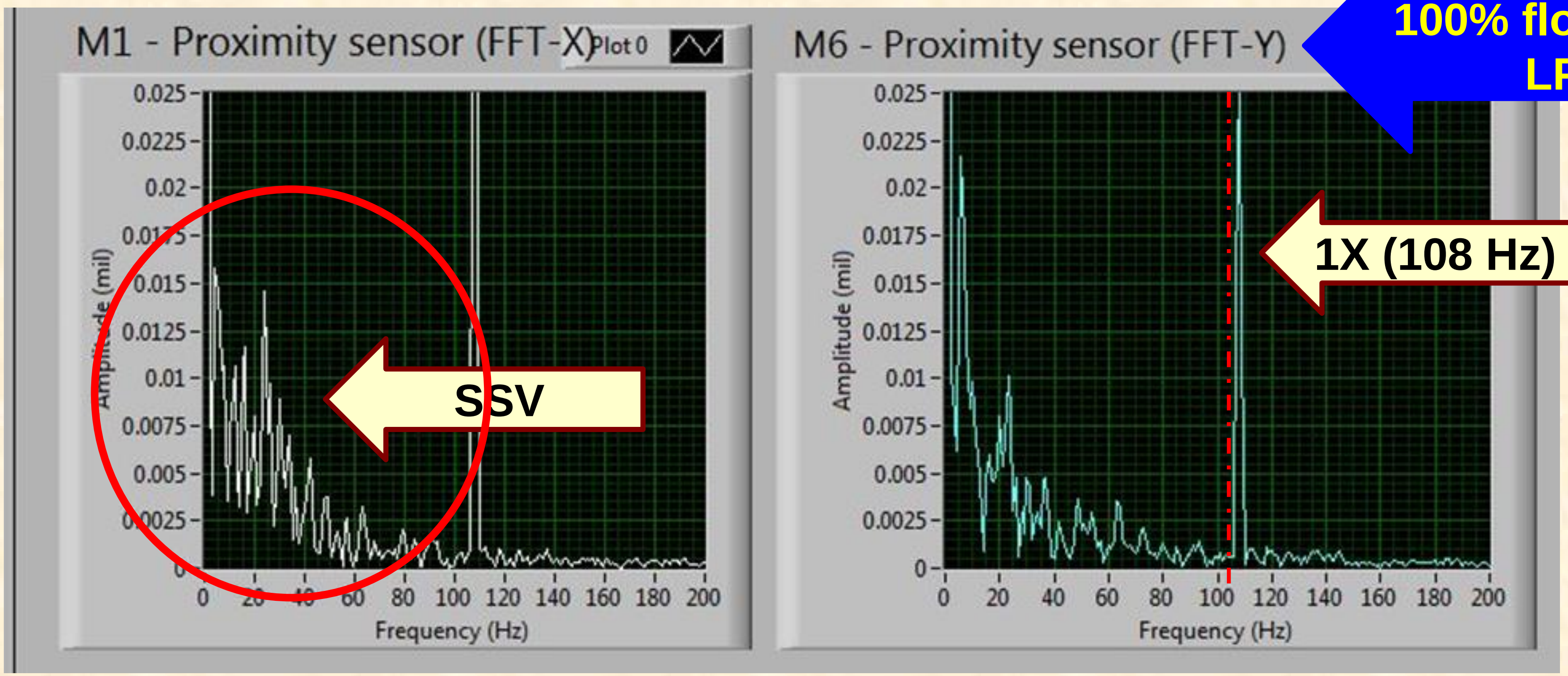
SSV hash appeared for flow < 35 % nominal.

SSV amplitude <<< clearance

Flooded bearing 6.5 krpm

6.5 kRPM, 345 kPa (50 psi) load, 0.36 LPM (4%)

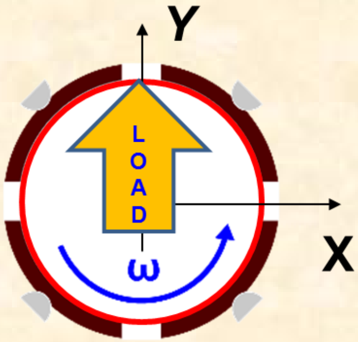
100% flow ~ 14.4 LPM



SSV hash appeared for operation with very-very low flow rates (& a small load).

SSV "breathed in" and needed to be excited.

CONCLUSION



EFFECT OF REDUCED FLOW RATE ON TILTING PAD BEARING PERFORMANCE:

FLOODED ENDS vs EVACUATED ENDS

Conclusion

- **Flow reduction results in:**
  - **Reduced drag power loss (more for evacuated B)**
  - **Increased pad metal temperatures.** **The efficiency gains depend on the bearing configuration and the acceptance criteria for increased pad temperatures.**
- **Flow has minor effect on bearing stiffness; damping reduces moderately as flow reduces.**
- **SSV did emerge under <u>very low</u> flow/light load operating conditions, but w/o excessive amplitudes or becoming unstable.** **Evacuated bearing more sensitive to SSV has when flow rate decreased below 32 % nominal.**

Acknowledgments

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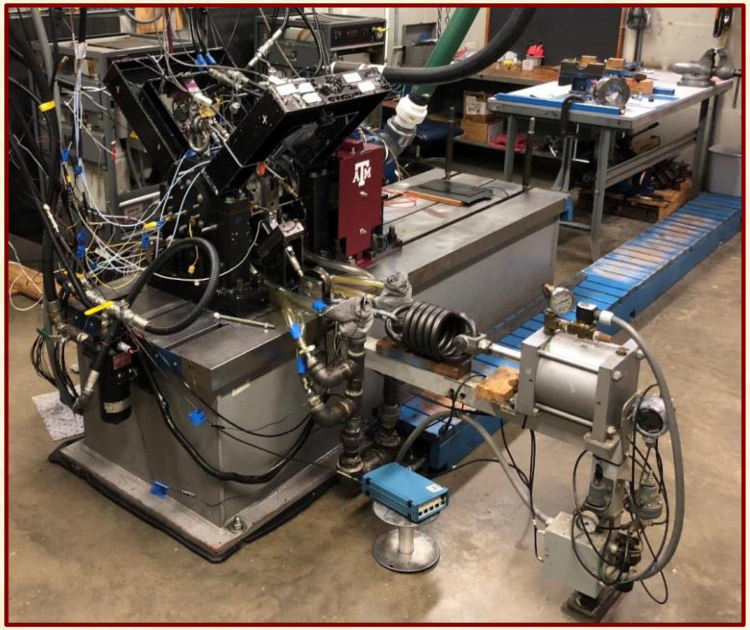
Turbomachinery Research Consortium.

TURBOMACHINERY LABORATORY
TEXAS A&M ENGINEERING EXPERIMENT STATION

Thanks to Mr. Jon Toner and Mr. John Whalen



Questions (?)

TURBOMACHINERY & PUMP SYMPOSIA

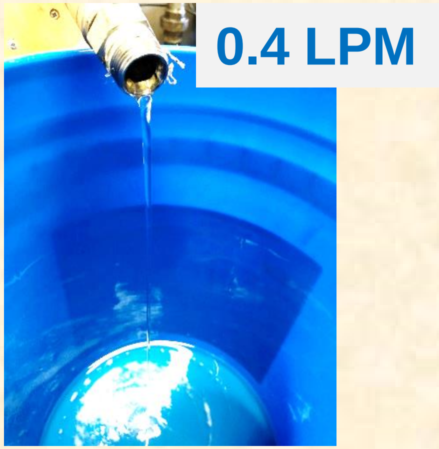
Experimental Results - Low Flow Limit Tests

Reducing flow rate reduces power consumption.

Yet How low is too low?

The minimum flow is application specific but must prevent too large pad/film temperatures to avoid:

- **Babbitt failure**
- **Varnishing of pads or (long term) degradation of oil**
- **Collapse of load capacity with excessive reduction in stiffness and damping coefficients**



Recall nominal flow rate at 6 krpm: ~ 14.4 LPM

Results of Low Flow Limit Tests

Low flow limit found by reducing oil flowrate at a constant rotor speed and applied load until:

1) Pad Temperature exceeds 121C (250F) or

2) SSV vibration appears

3) Inlet temperature below target 60°C and/or annulus temperatures not uniform → Cannot maintain control flowrate and/or oil inlet temperature)

Limit of Low Oil Supply			
	Load	Flow	Limit
6 kRPM (32 m/s) Flow=14.4 LPM	345 kPa	2% (0.36 LPM)	3
	1034 kPa	10% (1.4 LPM)	3
	2068 kPa	5% (1 LPM)	1
12 kRPM (64 m/s) Flow=28.8 LPM	345 kPa	15% (4.3 LPM)	SSV
	1034 kPa	15% (4.3 LPM)	1
	2068 kPa	23% (6.8 LPM)	1