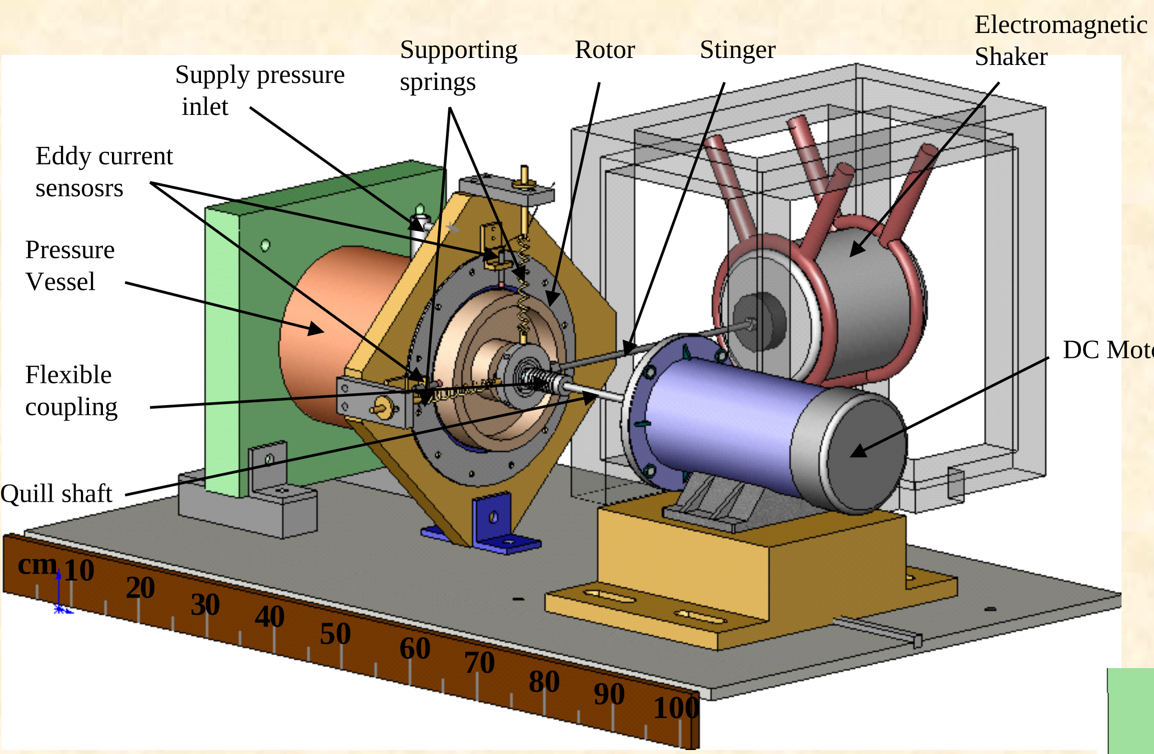
A 2nd example of system parameter identification (Hybrid Brush Seal)

Luis San Andrés (lecturer)

Thanks to Adolfo Delgado, José Baker & the support from Siemens Power Generation

Experimental Facility



Test Rig: Rotordynamic Configuration

Structural parameters

K_{shaft}= 243 lbf/in (42.5 kN/m)

M_{s+d}= 9.8 lb (4.45 kg)

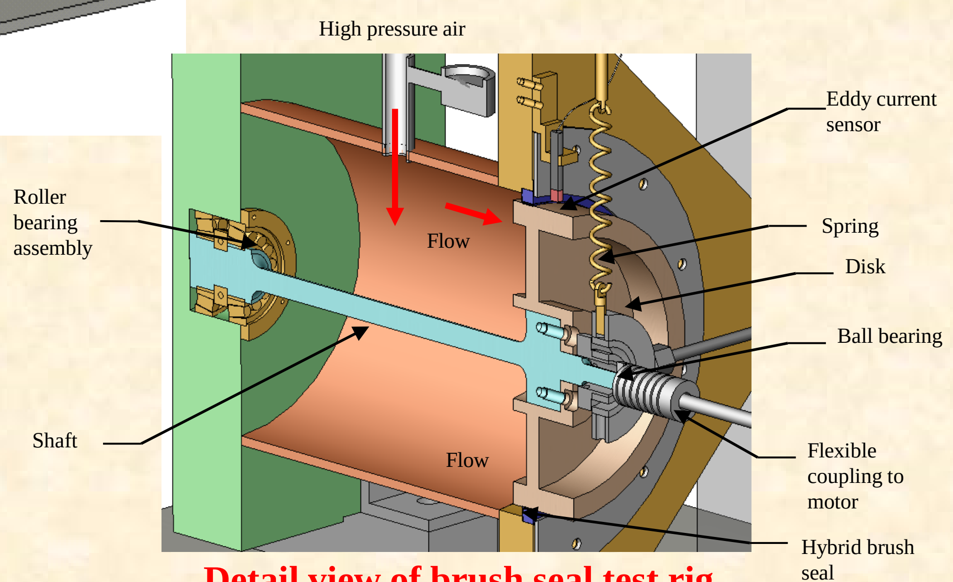
ζ: 0.01 % (damping ratio)

Installation:

6.550” diameter brush seal

Max. air Pressure: 60 psig

Shaker (20 lb max)

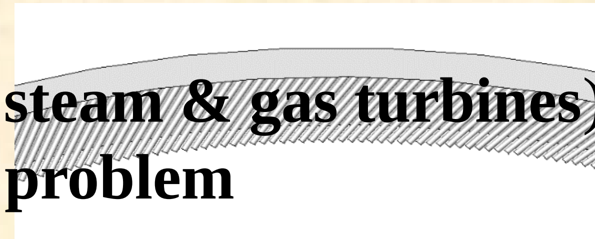


Detail view of brush seal test rig

GT2008-50532

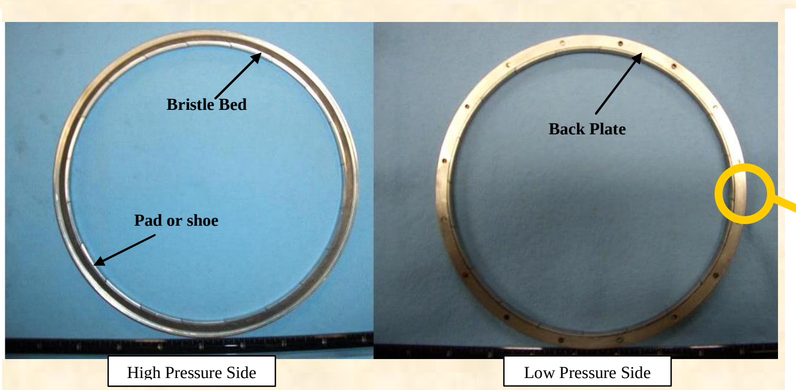
Brush Seals

Reduce secondary leakage in turbomachinery
Replace labyrinth seals in HP TM (hot side of steam & gas turbines)
Wear and thermal distortions are a reliability problem



Hybrid Brush Seals

Novel improvement over BS. Reduce more leakage and do not introduce wear or thermal distortion. Allow bi-directional rotation

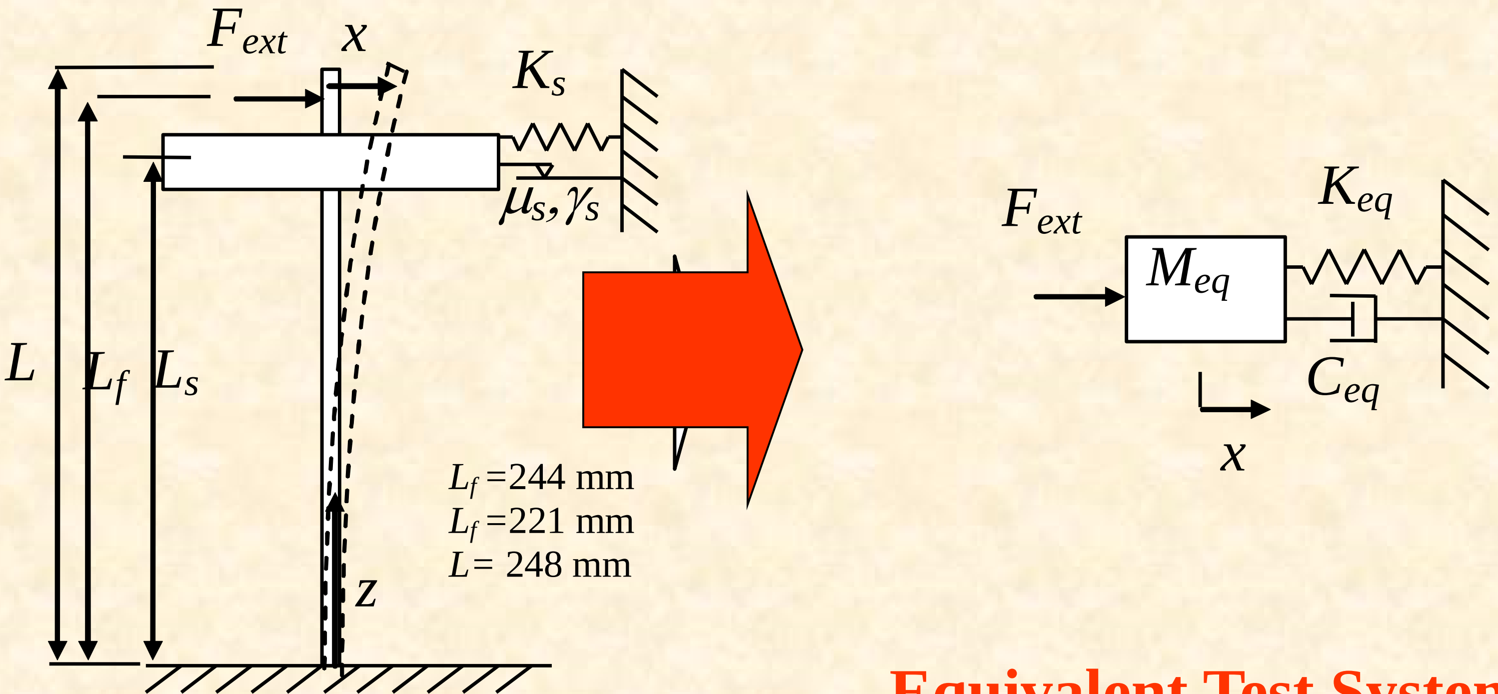


Courtesy of Advanced Turbomachinery Group®

*** Close-up courtesy of Advanced Technologies Group, Inc.**

Spring Lever Mechanism

Dynamic Load Tests (no shaft rotation)



Equivalent Test System

$$M_{eq}\ddot{x} + K_{eq}x + C_{eq}\dot{x} = F_{ext}$$

Parameter Identification (no shaft rotation)

ASME DETC2005-84159

$$\bar{x} = x e^{i\omega t} \qquad \bar{F} = F_{ext} e^{i\omega t}$$

Harmonic force & displacements

$$Z = \frac{\bar{F}}{\bar{x}} = (K_{eq} - \omega^2 M_{eq}) + i\,\omega C_{eq}$$

Impedance Function

$$W = \oint F_{ext}\, \dot{x}\, dt$$

Work External

$$E_{dis} = \pi \omega C_{eq} |\bar{x}|^2$$

Viscous Dissipation

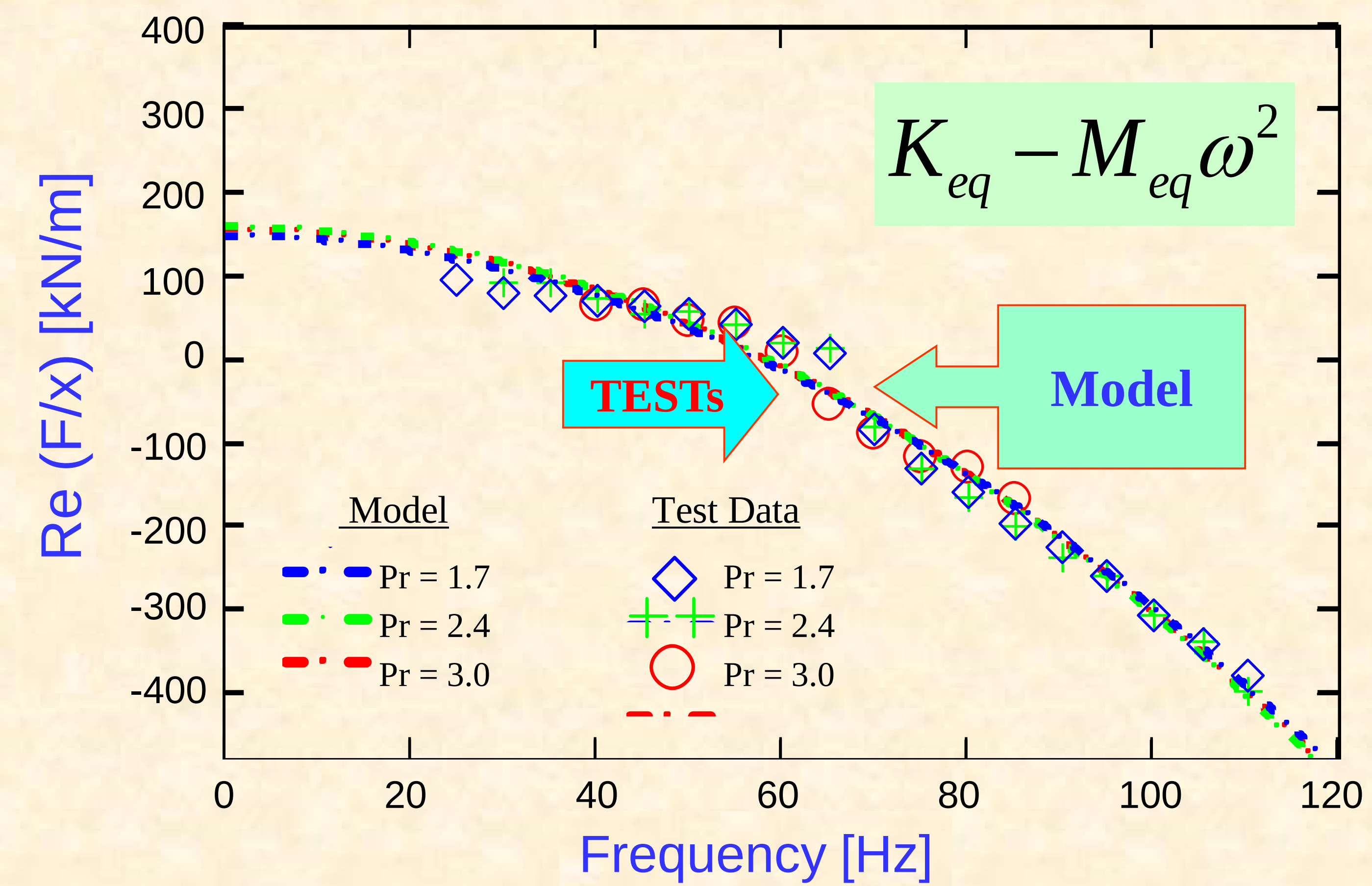
$$E_{dis} = \gamma_{eq} \pi K_{eq} |\bar{x}|^2 + 4\mu |\bar{F}| |\bar{x}|$$

DRY FRICTION & STRUCTURAL DAMPING

HBS Dynamic Stiffness vs. Frequency (no shaft rotation)

Load = 63 N, frequency: 20-100Hz

Pressure ratio ($P_r = P_s/P_d$) = Discharge/Supply



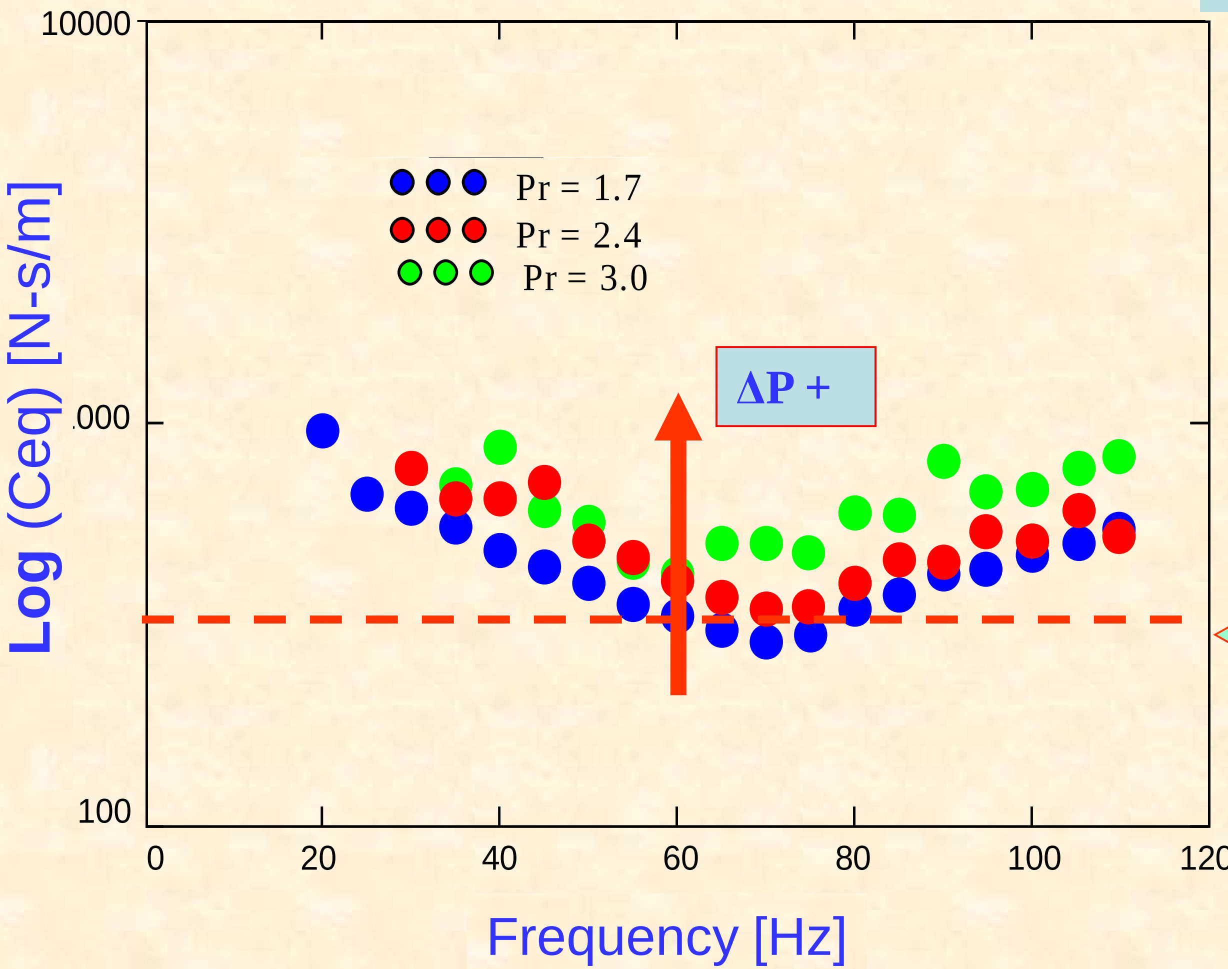
Model reproduces real part of the impedance under the given supply pressure conditions.

HBS Equivalent Viscous Damping vs. Frequency (no shaft rotation)

$$C_{eq} = \frac{\gamma_{eq} K_{eq}}{\omega} + \frac{4\mu\left|\bar{F}\right|}{\pi\omega\left|\bar{x}\right|}$$

Load = 63 N, frequency 20-110Hz

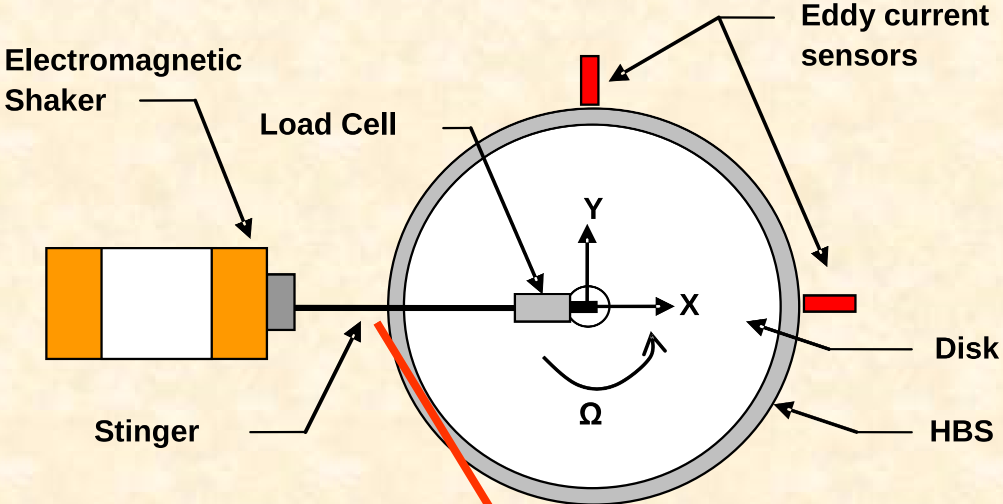
Pressure ratio ($P_r = P_s/P_d$) = Discharge/Supply

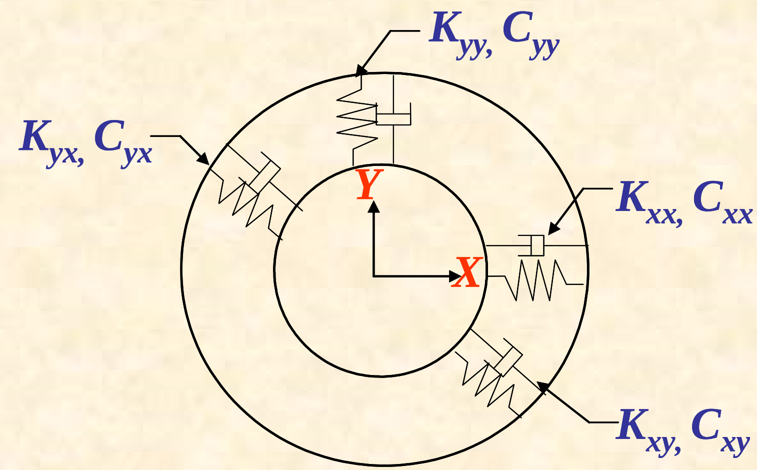


Viscous Model

Equivalent damping increases slightly with pressure differential. Results typical of a system with dry-friction & material damping energy dissipation

Identification of Rotordynamic Force Coefficients



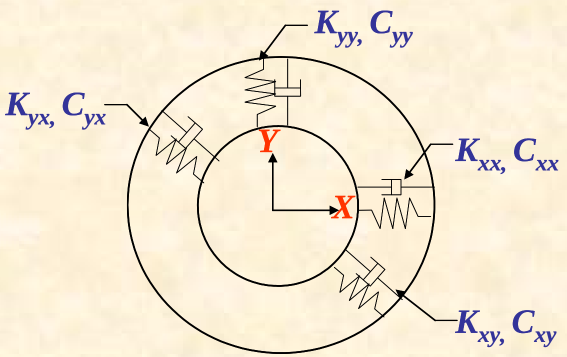


Excitation force (frequency ω)

$$\begin{bmatrix} M_{xx} & M_{xy} \\ M_{yx} & M_{yy} \end{bmatrix} \begin{Bmatrix} \ddot{x} \\ \ddot{y} \end{Bmatrix} + \begin{bmatrix} K_{xx} & K_{xy} \\ K_{yx} & K_{yy} \end{bmatrix} \begin{Bmatrix} x \\ y \end{Bmatrix} + \begin{bmatrix} C_{xx} & C_{xy} \\ C_{yx} & C_{yy} \end{bmatrix} \begin{Bmatrix} \dot{x} \\ \dot{y} \end{Bmatrix} = \begin{pmatrix} F_x \\ 0 \end{pmatrix} + \begin{pmatrix} F_{ix} \\ F_{iy} \end{pmatrix}$$

Imbalance forces (1X=Ω)

Identification of Rotordynamic Force Coefficients



For periodic force excitation:

$$\overline{F}_x = F_x e^{i\omega t} \Rightarrow \begin{array}{l} \overline{y} = y e^{i\omega t} \\ \overline{x} = x e^{i\omega t} \end{array}$$

EOMS reduce to:

$$Z_{xx} \cdot \overline{x} + Z_{xy} \cdot \overline{y} = \overline{F}_x$$

$$Z_{yx} \cdot \overline{x} + Z_{yy} \cdot \overline{y} = 0$$

With **impedances**:

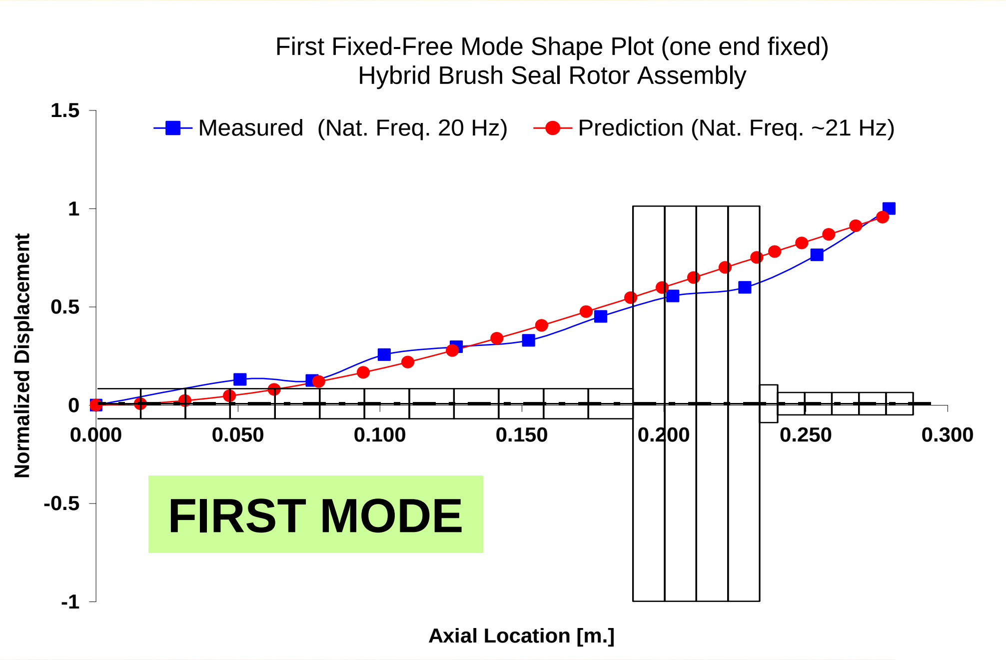
$$Z_{\alpha\beta} = \left\{ K_{\alpha\beta} - M_{\alpha\beta}\omega^2 + iC_{\alpha\beta}\omega \right\}_{\alpha\beta = x,y}$$

For centered operation (axi-symmetry)

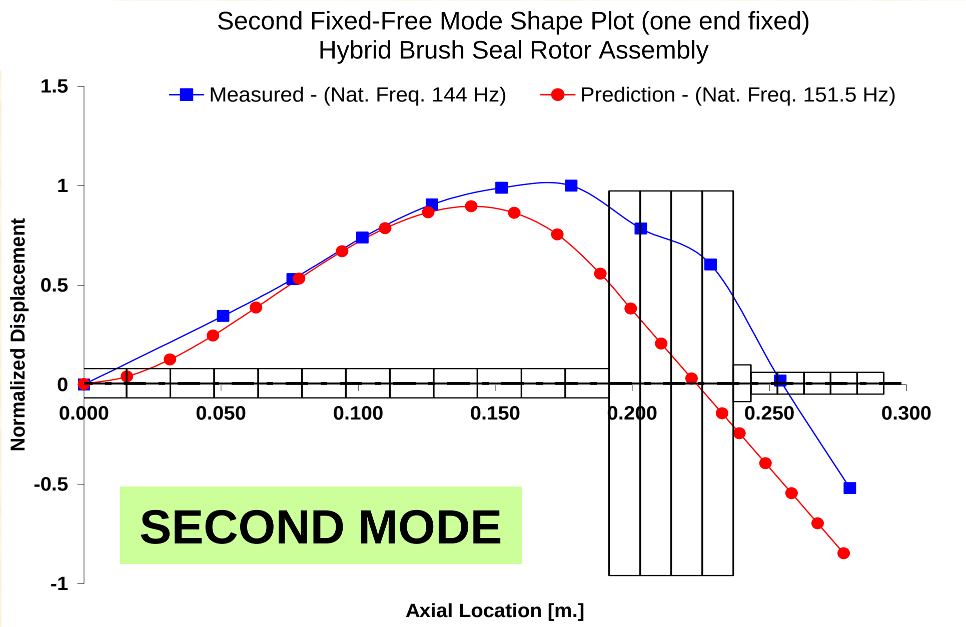
$$Z_{xx} = Z_{yy} \qquad Z_{xy} = -Z_{yx}$$

$$\begin{bmatrix} M_{xx} & 0 \\ 0 & M_{xx} \end{bmatrix} \begin{Bmatrix} \ddot{x} \\ \ddot{y} \end{Bmatrix} + \begin{bmatrix} K_{xx} & K_{xy} \\ -K_{xy} & K_{xx} \end{bmatrix} \begin{Bmatrix} x \\ y \end{Bmatrix} + \begin{bmatrix} C_{xx} & 0 \\ 0 & C_{xx} \end{bmatrix} \begin{Bmatrix} \dot{x} \\ \dot{y} \end{Bmatrix} = \begin{pmatrix} F_x \\ 0 \end{pmatrix}$$

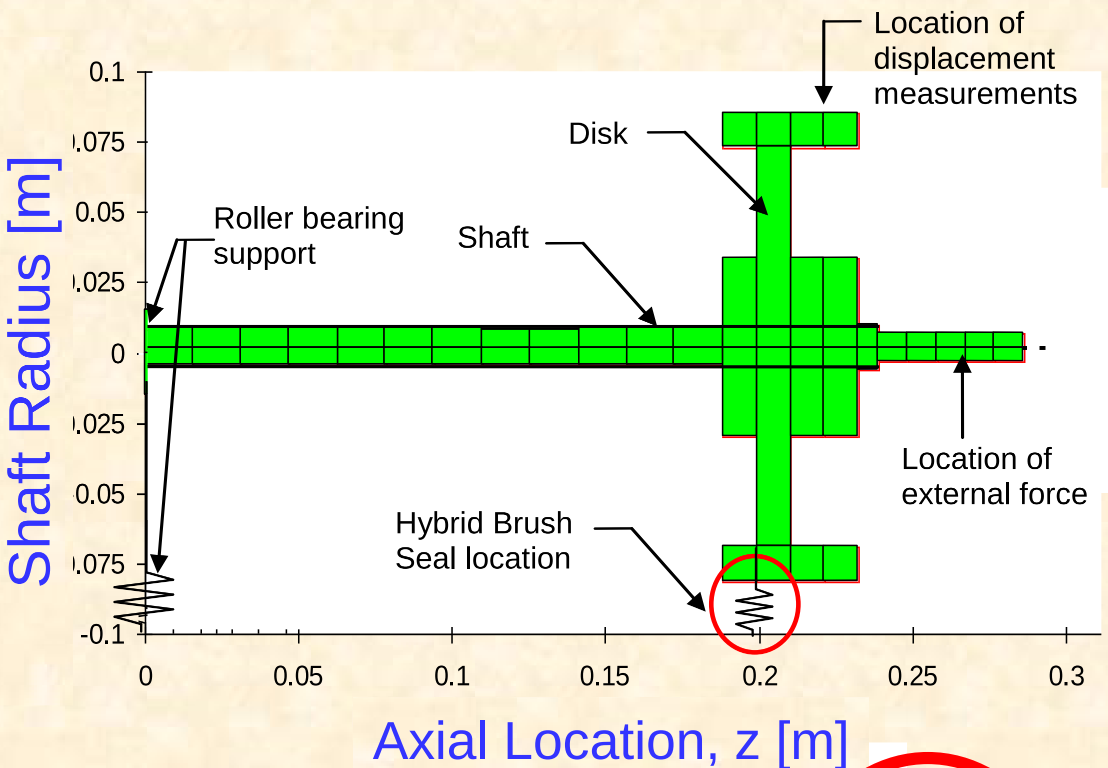
Rotor mode shapes



Only first mode excited in rotor speed range (0-1200 rpm)



Effect of rotor speed on rotor-HBS natural frequency



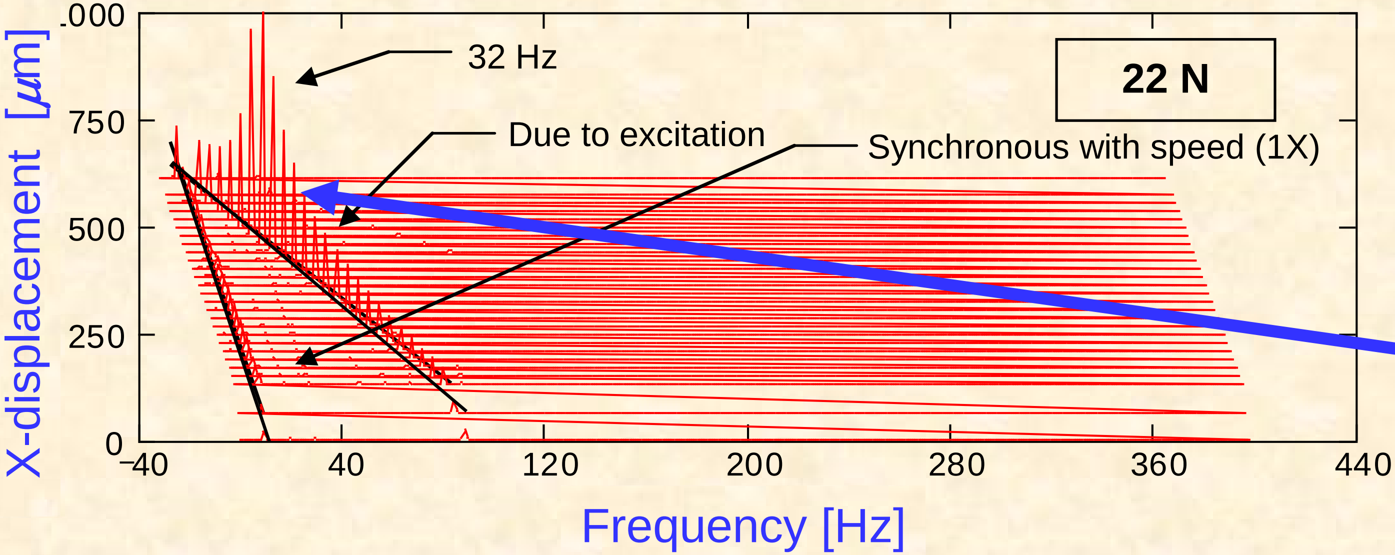
Gyroscopic effects negligible for test rotor speeds (600 and 1,200 rpm [20 Hz])

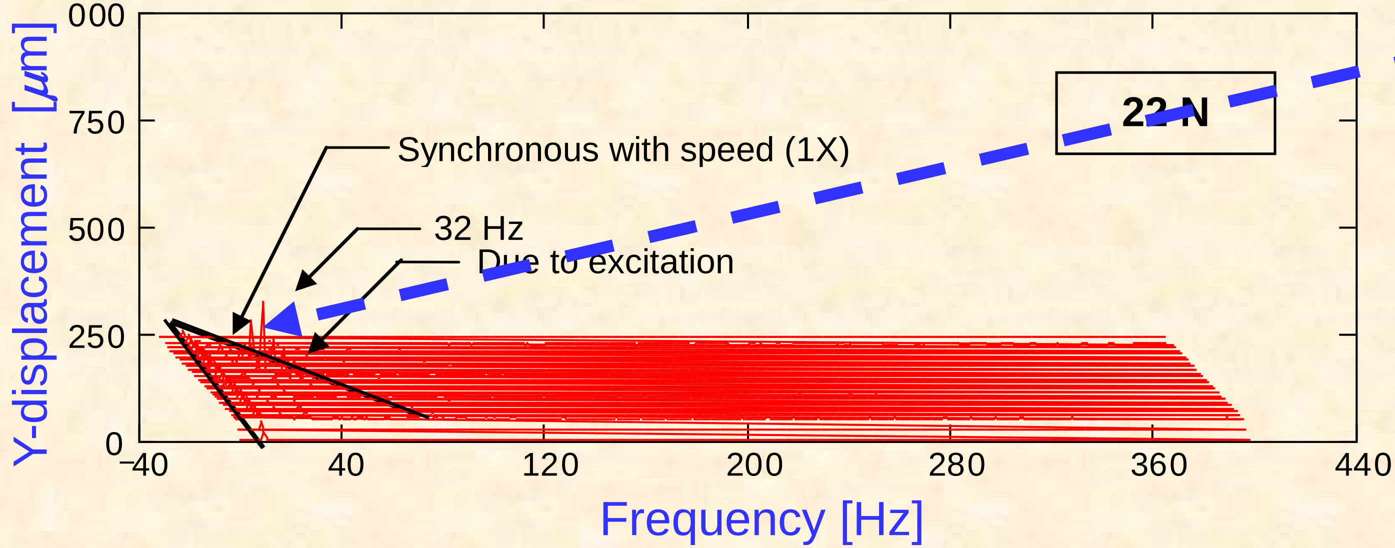
Rotor Speed [RPM]	1st Backward Nat. Frequency, [Hz]	1st Forward Nat. Frequency, [Hz]	2nd Forward Nat. Frequency, [Hz]	3rd Forward Nat. Frequency, [Hz]
0	30.5	30.5	146	1351
600	29.7	31.4	154	1351
1200	28.8	32.2	163	1351

T=32 Hz

CROSS-Coupling Effects under rotation

Load=22 N, 600 rpm





For load along *X* direction, rotor principal (*X*) motions

\>>>

cross (*Y*) motions

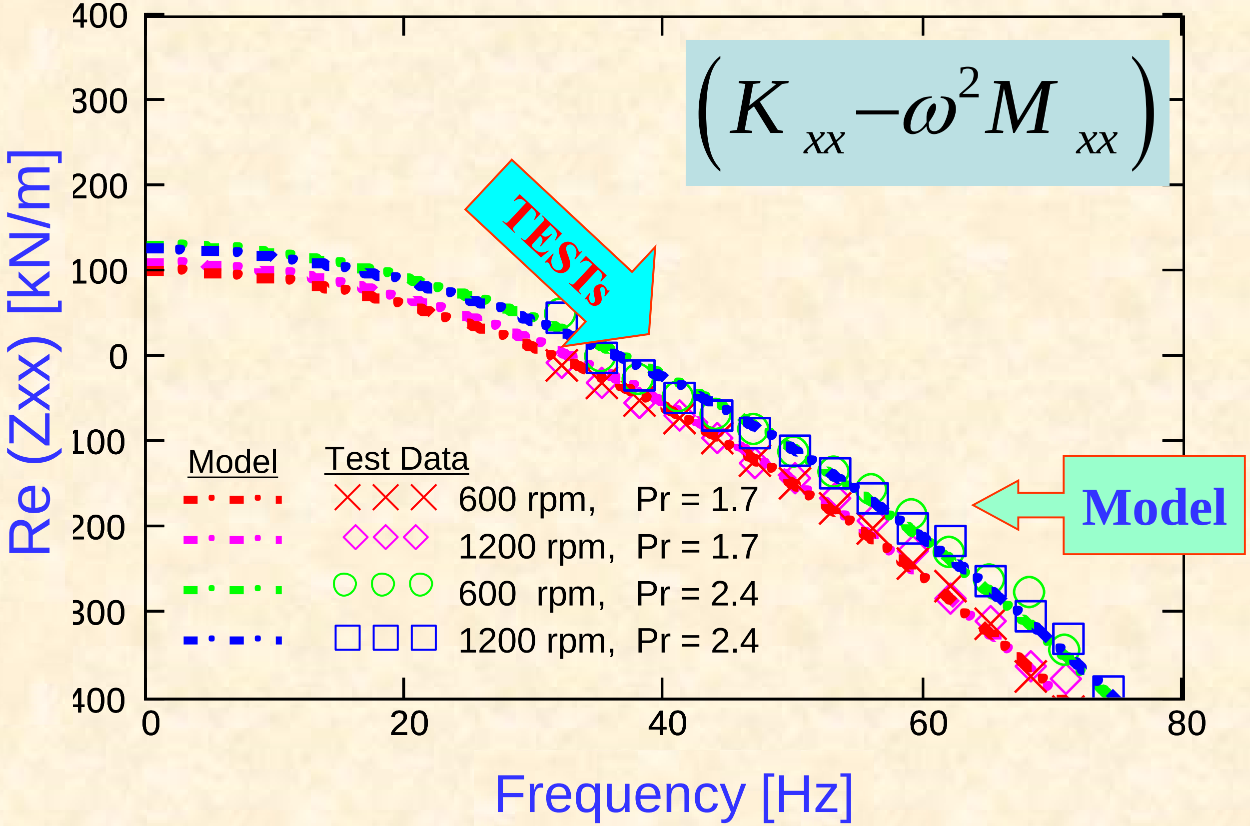
***3X* motions always small**

Test dynamic stiffness vs. frequency

Load = 22 N, frequency 25-80Hz

Pressure ratio ($P_r = P_s/P_d$) = Discharge/Supply

$$Z_{xx} = \frac{\overline{F}_x \cdot \overline{x}}{(\overline{x}^2 + \overline{y}^2)}$$



Model reproduces the measured real part of impedance.

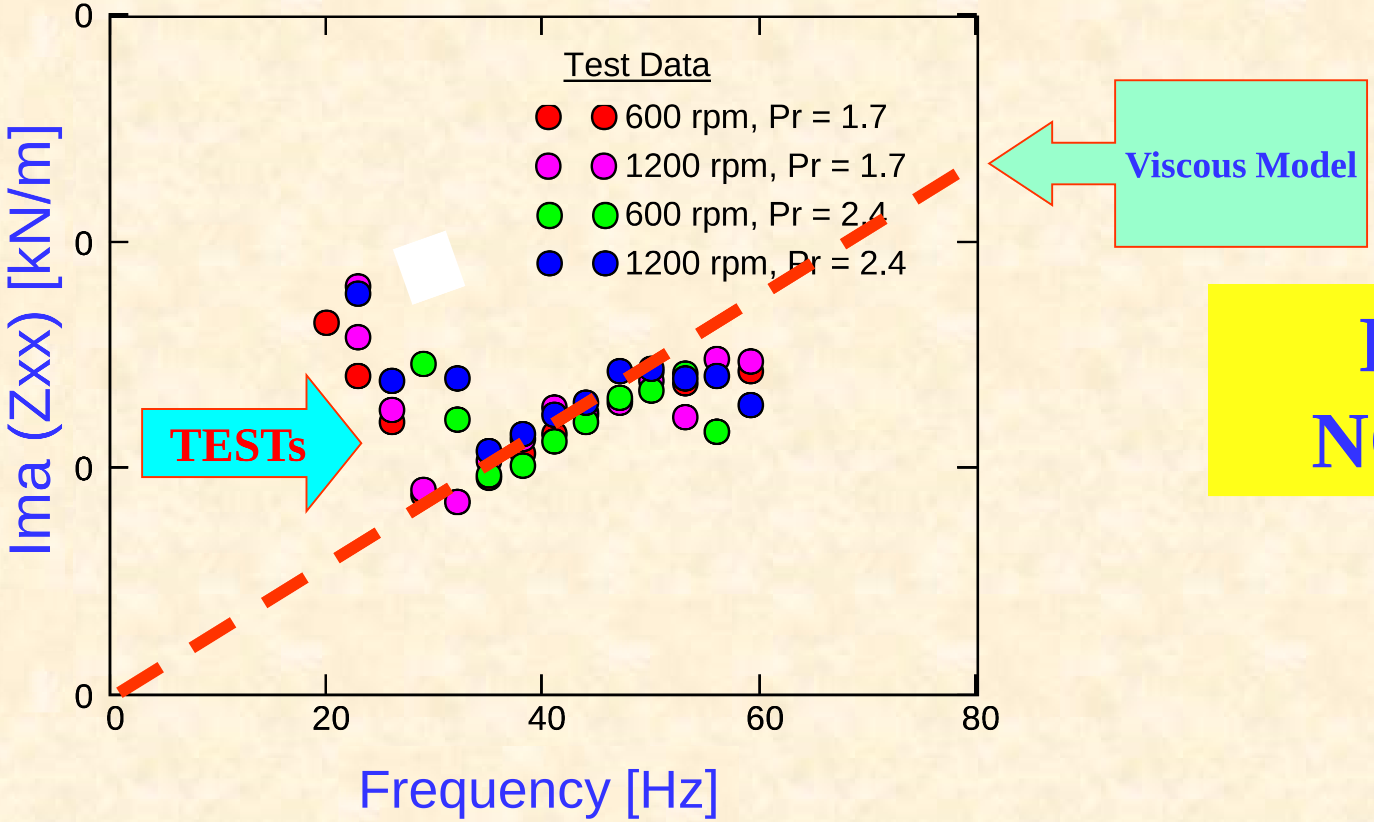
Little effect of pressurization

Test quadrature stiffness vs. frequency

Load = 22 N, frequency 25-80Hz

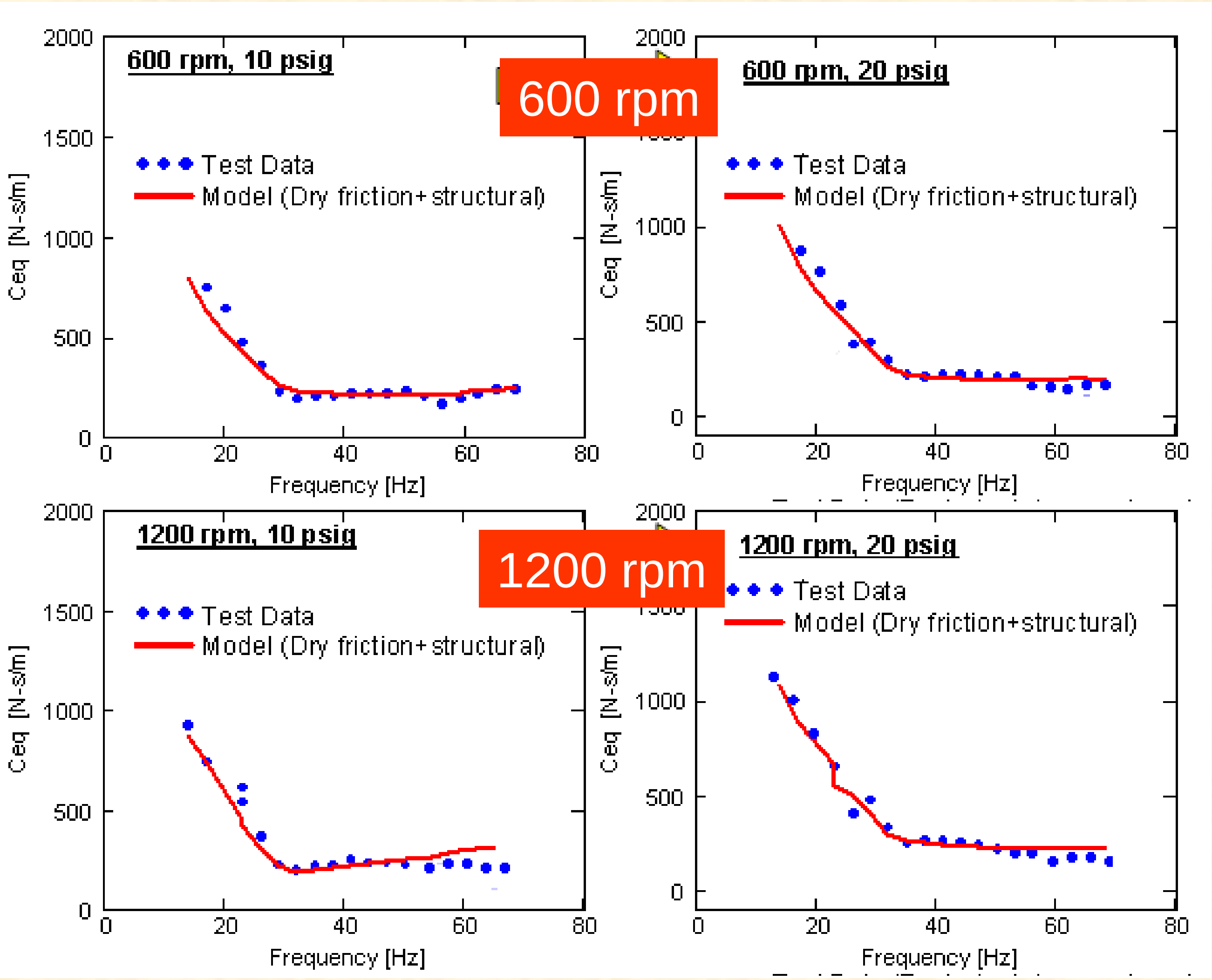
Pressure ratio ($P_r = P_s/P_d$) = Discharge/Supply

$$Z_{yx} = -Z_{yy}\,\frac{\bar{y}}{\bar{x}}$$



Damping is NOT viscous

Equivalent Viscous Damping (C_{xx}~C_{eq}) vs. Frequency



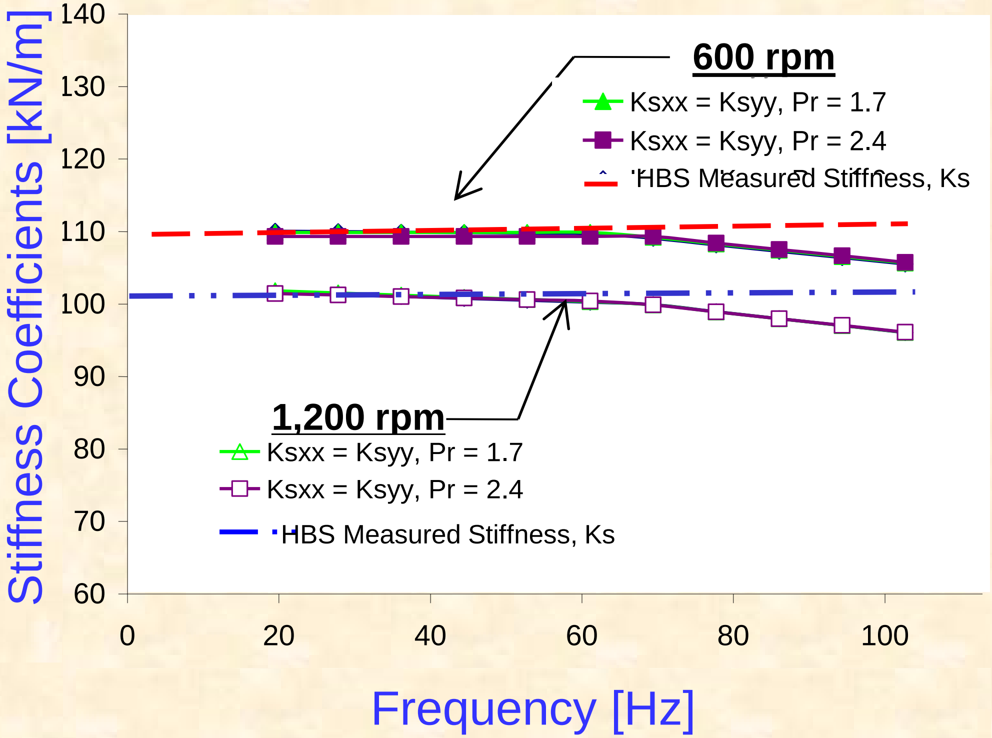
Pr=1.7

Pr=2.4

Damping decreases with frequency, with little effect of supply pressure. Minimum value at test system natural frequency (~32 Hz)

HBS predicted & test direct stiffness vs. frequency

Frequency 25-100 Hz



Predicted HBS stiffness (K_{sxx}) drops slightly in range from 20- 100 Hz. Tests show nearly constant K_{sxx}

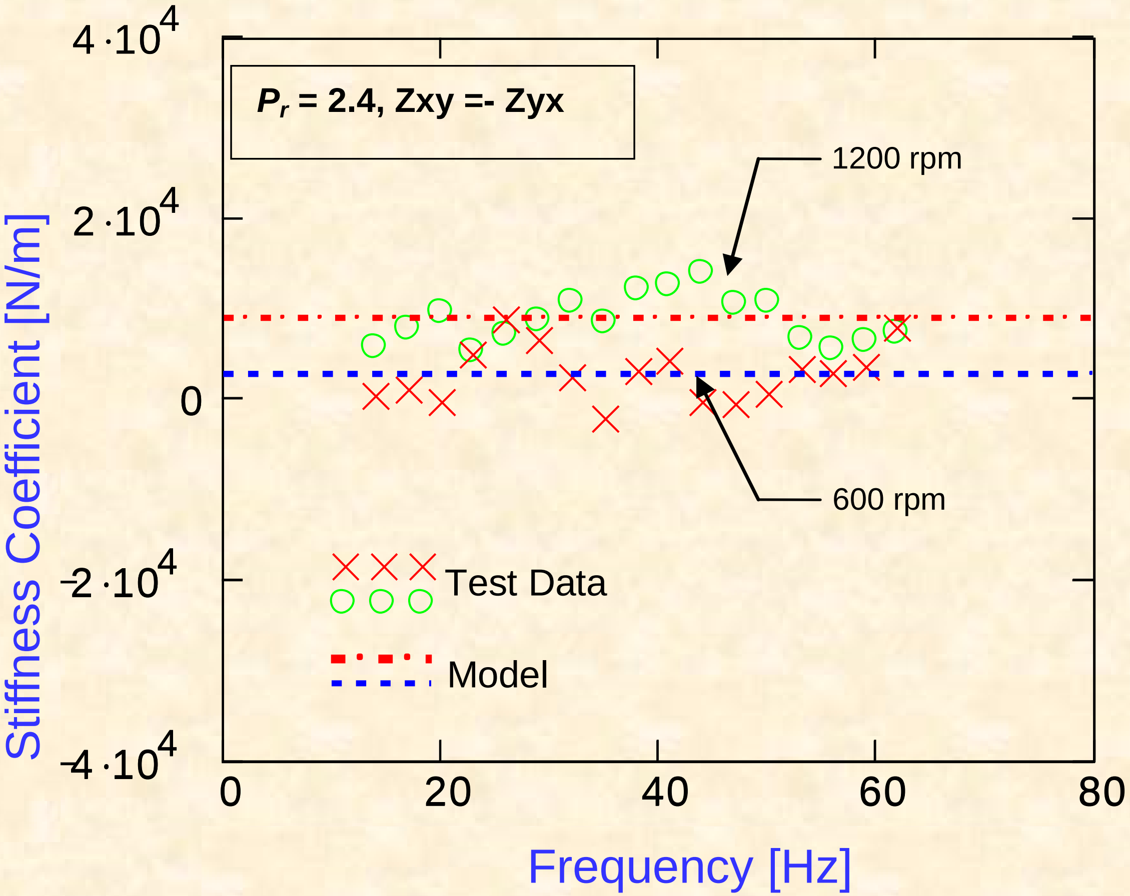
Pressure ($P_r = P_s/P_d$) has negligible effect on seal direct stiffness, K_{sxx}

ASME GT2004-53614

Code: XLTPGASBEAR

HBS predicted & test cross stiffness vs. frequency

Frequency 25-100 Hz



$$Z_{yx} = -Z_{xx}\frac{\bar{y}}{\bar{x}}$$

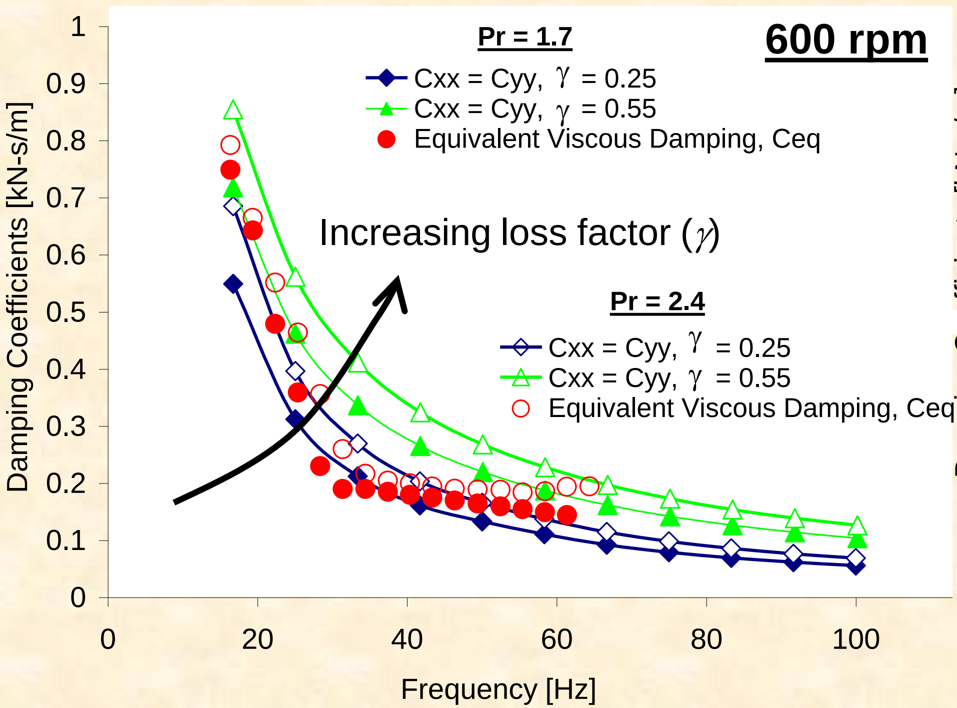
HBS cross stiffness (K_{sxy}) << direct stiffness (K_{sxx})

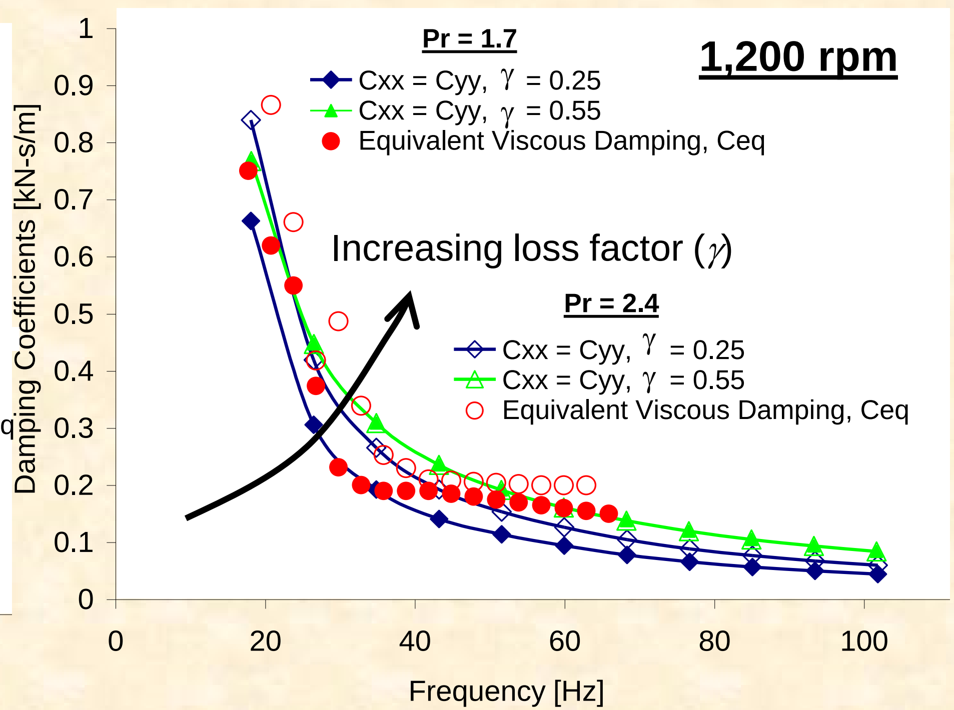
Pressure ($P_r = P_s/P_d$) has negligible effect on seal cross stiffness, K_{sxy}

HBS predicted & test damping vs. frequency

Frequency 25-100 Hz

Pressure ratio ($P_r = P_s/P_d$) = Discharge/Supply





HBS direct damping (C_{sxx}) decreases with excitation frequency. Loss factor coefficient (γ) models well seal structural (hysteresis) damping

Conclusions

• **A structural loss factor (γ) and a dry friction coefficient(μ) effectively characterize the energy dissipation mechanism of a Hybrid Brush Seal (HBS).**

• **HBS Direct stiffness ($K_{sxx} = K_{syy}$) decreases minimally with rotor increasing rotor speed for P_r = 1.7 and 2.4 HBS Cross-coupled stiffness ($K_{sxy} = -K_{syx}$) is much smaller than the direct stiffnesss.**

• **HBS Direct viscous damping coefficients decrease as a function of increasing excitation frequency.**