June 2011 - Update

Thermohydrodynamic Analysis of Bump Type Gas Foil Bearings: A Model Anchored to Test Data

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This material is based upon work supported by NASA NNH06ZEA001N-SSRW2, Fundamental Aeronautics: Subsonic Rotary Wing Project and the Texas A&M Turbomachinery Research Consortium

Outline

- Statement of Work & Sources for Presentation
- Objectives and accomplished work in 2007-08
  - Computational model. Validation with published data.
  - Rotordynamic measurements at TAMU
- Objectives and accomplished work in 2008-09
  - Description of test rig and foil bearings at TAMU
  - Effect of temperature on bearing temperatures, coastdown speed and rotor motions
  - Effect of cooling flow on shaft and bearing temperatures. Validation of computational model

* The computational code
  - Graphical User Interface. Further predictions

- GFB thermal management tests and preds.
- Added Value and Closure

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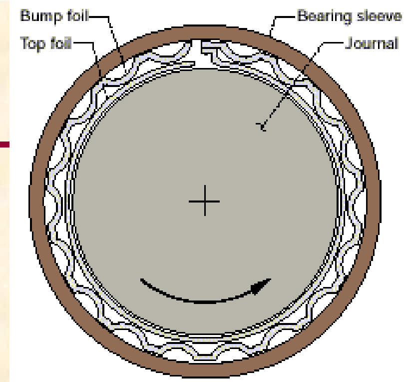
Gas Foil Bearings (+/-)



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- **Increased reliability: large load capacity (< 100 psi)**
- **No lubricant supply system, i.e. reduce weight**
- **High and low temperature capability** (up to 2,500 K)
- **No scheduled maintenance**
- **Ability to sustain high vibration and shock load. Quiet operation**

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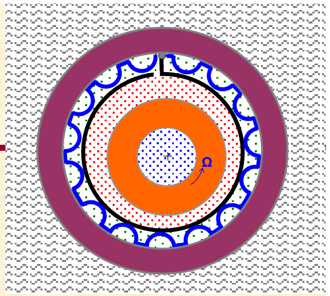
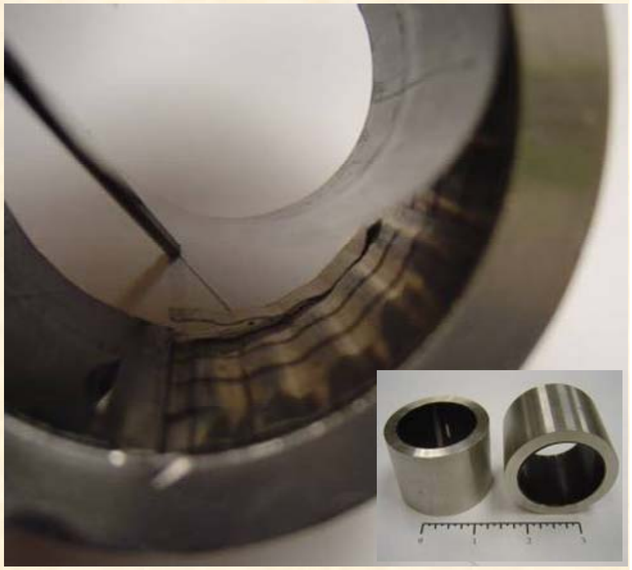
- **Less load capacity than rolling or oil bearings**
- **Wear during start up & shut down**
- **No test data for rotordynamic force coefficients**
- **Thermal management issues**
- **Predictive models lack validation.** Difficulties in modeling + dry-friction damping + effects of temperature on material properties and components' expansion.

Applications: ACMs, micro gas turbines, turbo expanders

SOW – Main Objective



To develop a detailed, physics-based computational model of gas-lubricated foil journal bearings including thermal effects to predict bearing performance.



The result of this work shall include a fully tested and experimentally verified design tool for predicting gas foil journal bearing torque, load, gas film thickness, pressure, flow field, temperature distribution, thermal deformation, foil deflections, stiffness, damping, and any other important parameters.

Agreement NASA NNH06ZEA001N-SSRW2

References Foil Bearings

ASME GT2011-46767	De Santiago, O., and San Andrés, L., 2011, “Parametric Study of Bump Foil Gas Bearings for Industrial Applications”
ASME GT2011-45763	San Andrés, L.., and Ryu, K., 2011, “On the Nonlinear Dynamics of Rotor-Foil Bearing Systems: Effects of Shaft Acceleration, Mass Imbalance and Bearing Mechanical Energy Dissipation.”
ASME GT2010-22508 NASA/TM 2010-216354	Howard, S., and San Andrés, L., 2011, “A New Analysis Tool Assessment for Rotordynamic Modeling of Gas Foil Bearings,” ASME J. Eng. Gas Turbines and Power, v 133
ASME GT2010-22981	San Andrés, L., Ryu, K., and Kim, T-H, 2011, “Thermal Management and Rotordynamic Performance of a Hot Rotor-Gas Foil Bearings System. Part 2: Predictions versus Test Data,” ASME J. Eng. Gas Turbines and Power, v 133
ASME GT2010-22981	San Andrés, L., Ryu, K., and Kim, T-H, 2011, “Thermal Management and Rotordynamic Performance of a Hot Rotor-Gas Foil Bearings System. Part 1: Measurements,” ASME J. Eng. Gas Turbines and Power, v 133
J Eng Gas Turbines & Power	San Andrés, L., Ryu, K., and Kim, T.H., 2011, “Identification of Structural Stiffness and Energy Dissipation parameters in a 2nd Generation Foil Bearing; Effect of Shaft Temperature”, ASME J. Eng. Gas Turbines Power, vol. 133 (March) , pp. 032501
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ASME GT2009-59920	San Andrés, L., Kim, T.H., Ryu, K., Chirathadam, T. A., Hagen, K., Martinez, A., Rice, B., Niedbalski, N., Hung, W., and Johnson, M., 2009, “Gas Bearing Technology for Oil-Free Microturbomachinery – Research Experience for Undergraduate (REU) Program at Texas A&M University
AHS 2009 paper	Kim, T. H., and San Andrés, L., 2010, “Thermohydrodynamic Model Predictions and Performance Measurements of Bump-Type Foil Bearing for Oil-Free Turboshaft Engines in Rotorcraft Propulsion Systems,” ASME J. of Tribology, v132
ASME GT2009-59919	San Andrés, L., and Kim, T.H., 2010, “Thermohydrodynamic Analysis of Bump Type gas Foil Bearings: A Model Anchored to Test Data,” ASME J. Eng. Gas Turbines and Power, v 132

References **Foil Bearings**

IJTC2008-71195	Kim, T.H., and San Andrés, L., 2009, "Effects of a Mechanical Preload on the Dynamic Force Response of Gas Foil Bearings - Measurements and Model Predictions," Tribology Transactions, v52
ASME GT2008-50571 **IJTC2007-44047**	Kim, T. H., and San Andrés, L., 2009, "Effect of Side End Pressurization on the Dynamic Performance of Gas Foil Bearings – A Model Anchored to Test Data," ASME J. Eng. Gas Turbines and Power, v131. 2008 Best PAPER Rotordynamics IGTI
ASME GT2007-27249	San Andrés, L., and Kim, T.H., 2009, "Analysis of Gas Foil Bearings Integrating FE Top Foil Models," Tribology International, v42
J of Tribology	Kim, T.H., Breedlove, A., and San Andrés, L., 2009, "Characterization of Foil Bearing Structure at Increasing Temperatures: Static Load and Dynamic Force Performance," ASME Journal of Tribology, v 131(3)
Tribology International	San Andrés, L., and **Kim, T.H.,** 2008, "Forced Nonlinear Response of Gas Foil Bearing Supported Rotors," Tribology International, **41**(8), pp. 704-715. **Kim, T-H,** and L., San Andrés, 2007, "Analysis of Gas Foil Bearings with Piecewise Linear Elastic Supports**."** Tribology International, **40**, pp. 1239-1245.
AIAA-2007-5094	San Andrés, L., and T.H. Kim, 2007, "Issues on Instability and Force Nonlinearity in Gas Foil Bearing Supported Rotors," 43rd AIAA/ASME/SAE/ASEE Joint Propulsion Conference, Cincinnati, OH, July 9-11
ASME GT2005-68486	Kim, T.H., and L. San Andrés, 2008, "Heavily Loaded Gas Foil Bearings: a Model Anchored to Test Data," ASME J. Eng. Gas Turbines and Power, v130
ASME GT2006-91238	San Andrés, L., D. Rubio, and T.H. Kim, 2007, "Rotordynamic Performance of a Rotor Supported on Bump Type Foil Gas Bearings: Experiments and Predictions," ASME J. Eng. Gas Turbines and Power, v129
ASME GT2005-68384	**Rubio, D.,** and L. San Andrés, 2007, "Structural Stiffness, Dry-Friction Coefficient and Equivalent Viscous Damping in a Bump-Type Foil Gas Bearing," ASME J. Eng. Gas Turbines and Power, v 129 2005 Best PAPER Rotordynamics IGTI Structures and Dynamics Committee
ASME GT2004-53611	San Andrés, L., and D. Rubio, 2006, "Bump-Type Foil Bearing Structural Stiffness: Experiments and Predictions," ASME J. Eng. Gas Turbines and Power, v128

References Metal mesh foil bearings

ASME GT2011-45274	San Andrés, L., and Chirathadam, T., 2011, **“**Metal Mesh Foil Bearings: Effect of Excitation Frequency on Rotordynamic Force Coefficients
ASME GT2010-22440	San Andrés, L., and Chirathadam T.A., 2010, “Identification of Rotordynamic Force Coefficients of a Metal Mesh Foil Bearing Using Impact Load Excitations.”
ASME GT2009-59315	San Andrés, L., Chirathadam, T. A., and Kim, T.H., 2009, “Measurements of Structural Stiffness and Damping Coefficients in a Metal Mesh Foil Bearing.”
AHS Paper	San Andrés, L., Kim, T.H., Chirathadam, T.A., and Ryu, K., 2009, “Measurements of Drag Torque, Lift-Off Journal Speed and Temperature in a Metal Mesh Foil Bearing,” American Helicopter Society 65th Annual Forum, Grapevine, Texas, May 27-29

Other

Journal of Tribology	**Kim, T.H.,** and L. San Andrés, 2006, “Limits for High Speed Operation of Gas Foil Bearings,” ASME Journal of Tribology, **128,** pp. 670-673
ASME DETC2007-34136	Gjika, K., C. Groves, L. San Andrés, and G. LaRue, 2007, “Nonlinear Dynamic Behavior of Turbocharger Rotor-Bearing Systems with Hydrodynamic Oil Film and Squeeze Film Damper in Series: Prediction and Experiment.”

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  Computational model. Validation with published data. Rotordynamic measurements at TAMU

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  Description of test rig and foil bearings at TAMU

  Effect of temperature on bearing temperatures, coastdown speed and rotor motions

  Effect of cooling flow on bearing and shaft temperatures. Validation of computational model

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Research Objectives (2007-08)

THD model for prediction of GFB performance

- Perform physical analysis, derive governing equations, and implement numerical solution.
- Develop GUI for User ready use
- Compare GFB predictions to limited published test data (NASA mainly)
- Revamp existing test rig with cartridge heater, acquire new bearings, machine new rotor
- Perform structural tests on bearings and measure rotordynamic response for increasing shaft temperatures

Scheduled Timeline & completion

Luis San Andres	MS student
Tae Ho Kim	UG worker

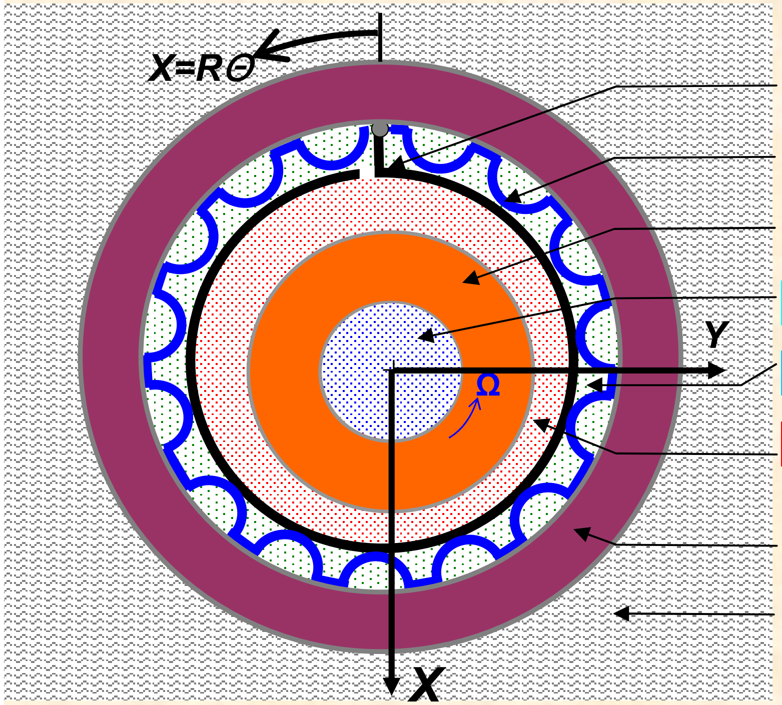
Task	Q1	Q2	Q3	Q4	Q5	Q6	Q7	Q8
Computational analysis GFBS								
Development physical model for thermal transport in foil bearings								
Implementation thermal model (Finite Element Based) and coupling to existing STRUCTURAL MODEL								
Integration of thermal model with GFB FD computational code (gas film)								
Predictions of GFB performance for parametric studies								
Comparison of GFB predictions to measured performance from TAMU test rig								
Nonlinear analysis GFBS								
Development simple NONLINEAR physical model for foil bearings								
Prediction of performance and comparisons to available rotordynamic test data								
Test rig for identification of FB structure (High Temperature)								
Planning of modification, selection of instrumentation and cartridge heater, design of insulation cover								
Reception of parts and assembly of components, troubleshooting, connection to static loader and shaker								
Measurements of load & bearing deflection for increasing shaft temperatures (max 500 C), identification of FB structural parameters								
Rotordynamic-GFBs Test rig (High Temperature)								
Planning of modifications to existing, selection of instrumentation and cartridge heater, design of insulation cover and rotor								
Reception of parts and assembly of components, troubleshooting, connection to static loader and shaker								
Rotordynamic Measurements for increasing shaft temperatures (max 500 C), identification of GFB synchronous force coefficients								

Accomplishments: Proposed & Actual

Task	Planned & Actual	comment
Thermohydrodynamic Analysis of GFBS		
Physical model for thermal transport in foil bearings. Integration of thermal model with GFB FD computational code (gas film). Prediction of GFB performance: parametric study	✓	Analysis completed. Code delivered on June 10, 2009
Validation of GFB predictions with measured temperatures from NASA & TAMU published research	✓	See Q4 & Q7 reports
Nonlinear structural analysis of GFBS		
Development simple NONLINEAR physical model for foil bearings. Prediction of performance and comparisons to rotordynamic test data	✓	Implementation in XLTRC2 for ready rotordynamic analyses
Test rig for identification of FB structure (High Temperature)		
Design & construction; selection & procurement of instrumentation and bearings; assembly, troubleshooting and operation at high temperature. Measurements of static load performance & comparison to predictions	✓	See Q4, Q7 reports
Rotordynamic-GFBs Test rig (High Temperature)	✓	
Design & construction; selection & procurement of instrumentation and bearings; assembly, troubleshooting and operation at high temperature rotor-bearing test rig. Measurements of temperatures and rotordynamic performance with Foster-Miller GFBs completed (see Q7). Tests with MiTi® bearings at higher temperatures in progress. Validation of computational model also in progress.	✓ **100%**	**Completed Dec 2010**

1st & 2nd Years

Thermohydrodynamic model in a GFB



Side view of GFB with hollow shaft

- Ideal gas with density, $\rho = P / \Re_g T$

- Gas viscosity, $\mu = \alpha_v T$

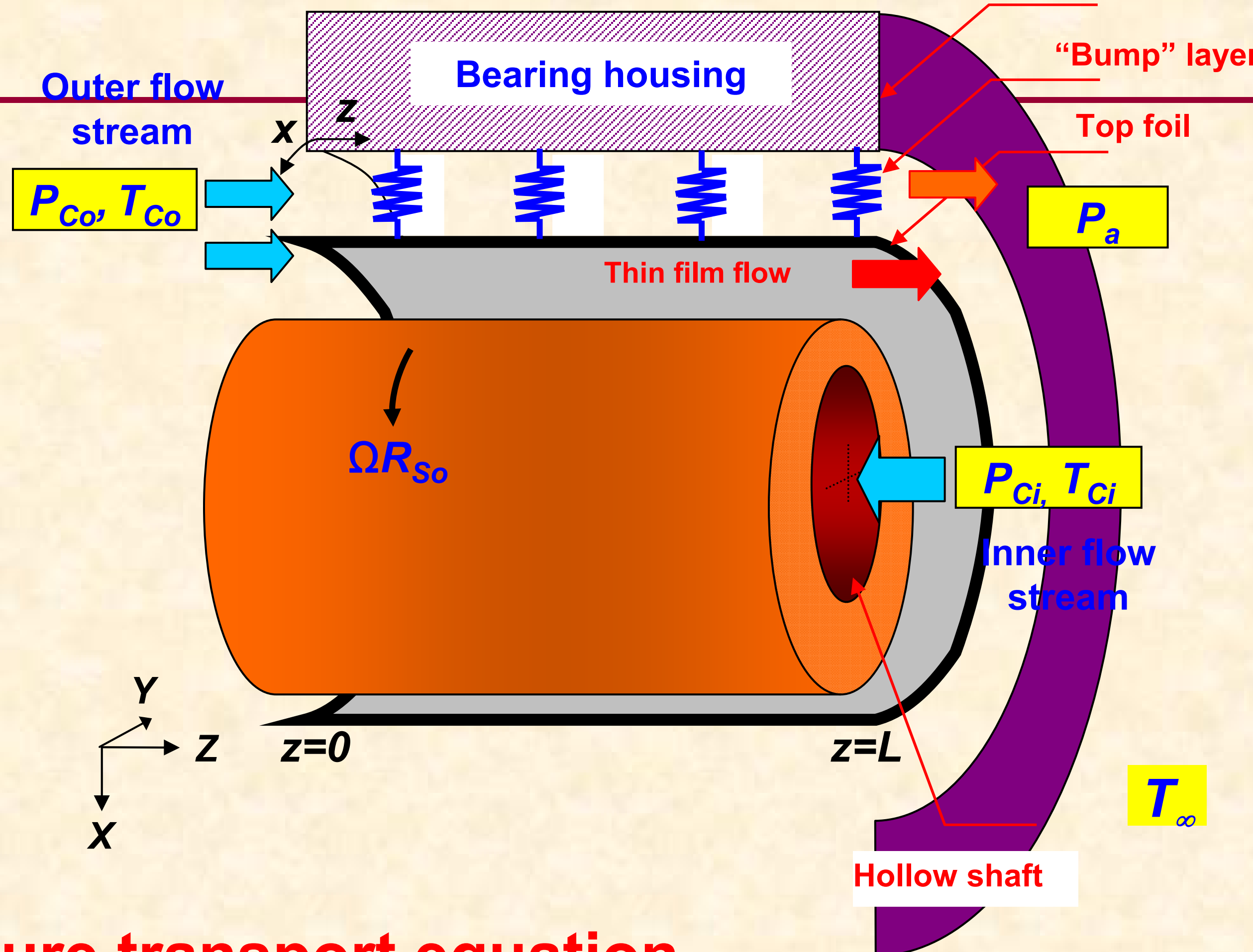
- Gas Specific heat (c_p) and thermal conductivity (κ_g) at an effective temperature

Reynolds equation in thin film

$$\frac{\partial}{\partial x}\left(\frac{h_f^3 P_f}{12\mu_f \Re_g T_f}\frac{\partial P_f}{\partial x}\right)+\frac{\partial}{\partial z}\left(\frac{h_f^3 P_f}{12\mu_f \Re_g T_f}\frac{\partial P_f}{\partial z}\right)=U_{m(z)}\frac{\partial}{\partial x}\left(\frac{P_f\, h_f}{\Re_g T_f}\right)$$

THD model

GFB with cooling flows
(inner and/or outer)



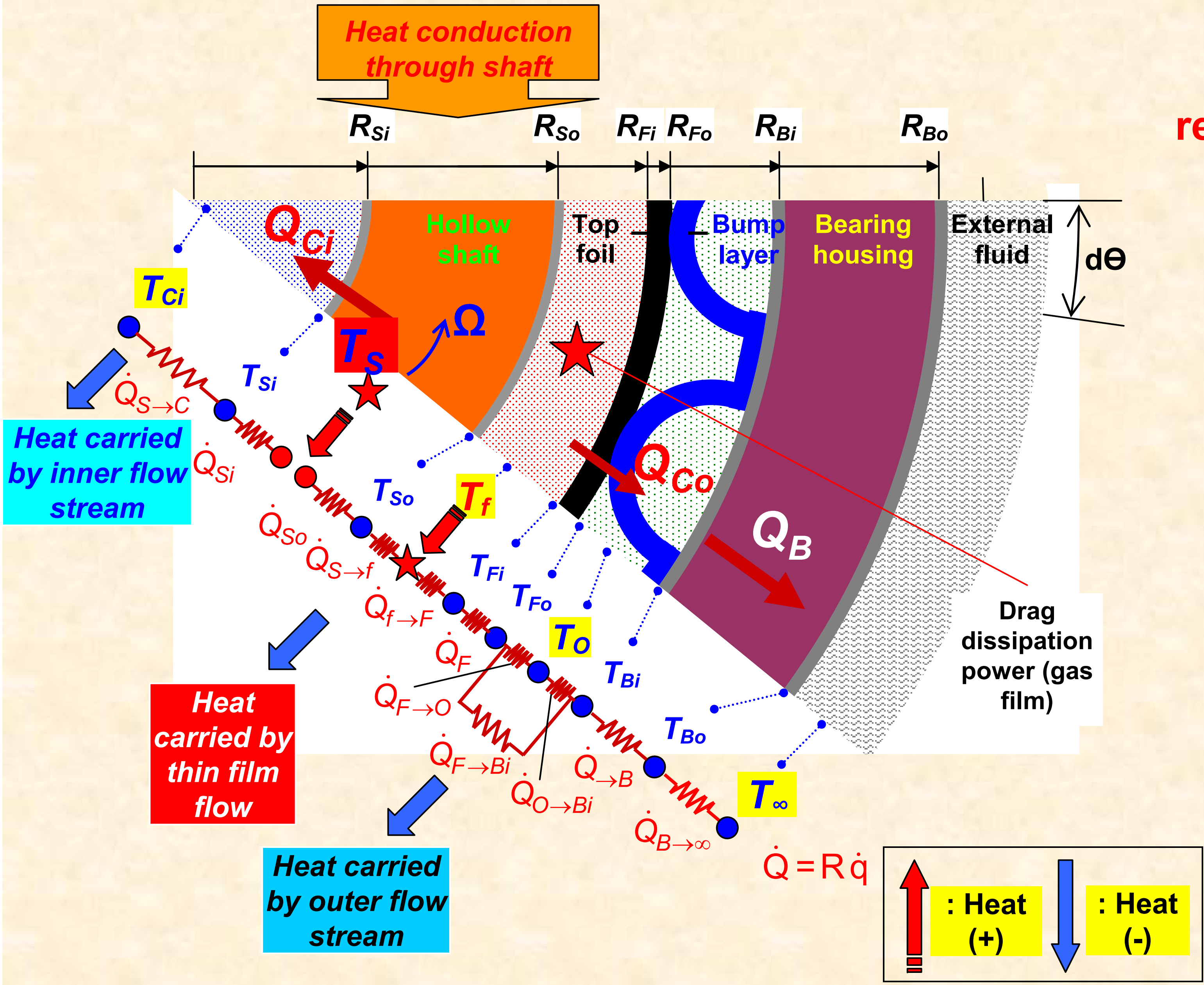
Bulk-flow temperature transport equation

$$c_{p_f}\left(\frac{\partial\left(\rho_f h_f U_f T_f\right)}{\partial x}+\frac{\partial\left(\rho_f h_f W_f T_f\right)}{\partial z}\right)+\bar{h}_{fF}\left(T_f-T_{F_i}\right)-\bar{h}_{Sf}\left(T_{S_o}-T_f\right)$$

$$=\left(U_f h_f \frac{\partial P_f}{\partial x}+W_f h_f \frac{\partial P_f}{\partial z}\right)+\frac{12\mu_f}{h_f}\left\{W_f^{\,2}+\frac{1}{3}U_m+\left(U_f-U_m\right)^2\right\}$$

Convection of heat by fluid flow + diffusion to bounding surfaces = compression work + dissipated energy

Heat flow paths in rotor - GFB system



Heat flows & thermal resistances in a GFB & hollow shaft

Heat conducted into bearing

Cooling gas streams carry away heat

THD Model Validation Published data

Generation I GFB with single top foil and bump strip layer

Parameters	Value / comment	
Bearing cartridge		
Bearing inner radius	25 mm	Ref. [7]
Bearing length	41 mm	Ref. [7]
Bearing cartridge thickness	5 mm	Assumed
Nominal radial clearance	20 μm	Assumed
Top foil and bump strip layer		
Top foil thickness	127 μm	Ref. [21]
Bump foil thickness	127 μm	Ref. [21]
Bump half length	1.778 mm	Assumed
Bump pitch	4.064 mm	Assumed
Bump height	0.580 mm	Assumed
Number of bumps x strips	39 x 1	Assumed
Bump foil Young's modulus	200 GPa	
Bump foil Poisson's ratio	0.31	
Bump foil stiffness	10.4 GN/m^3	

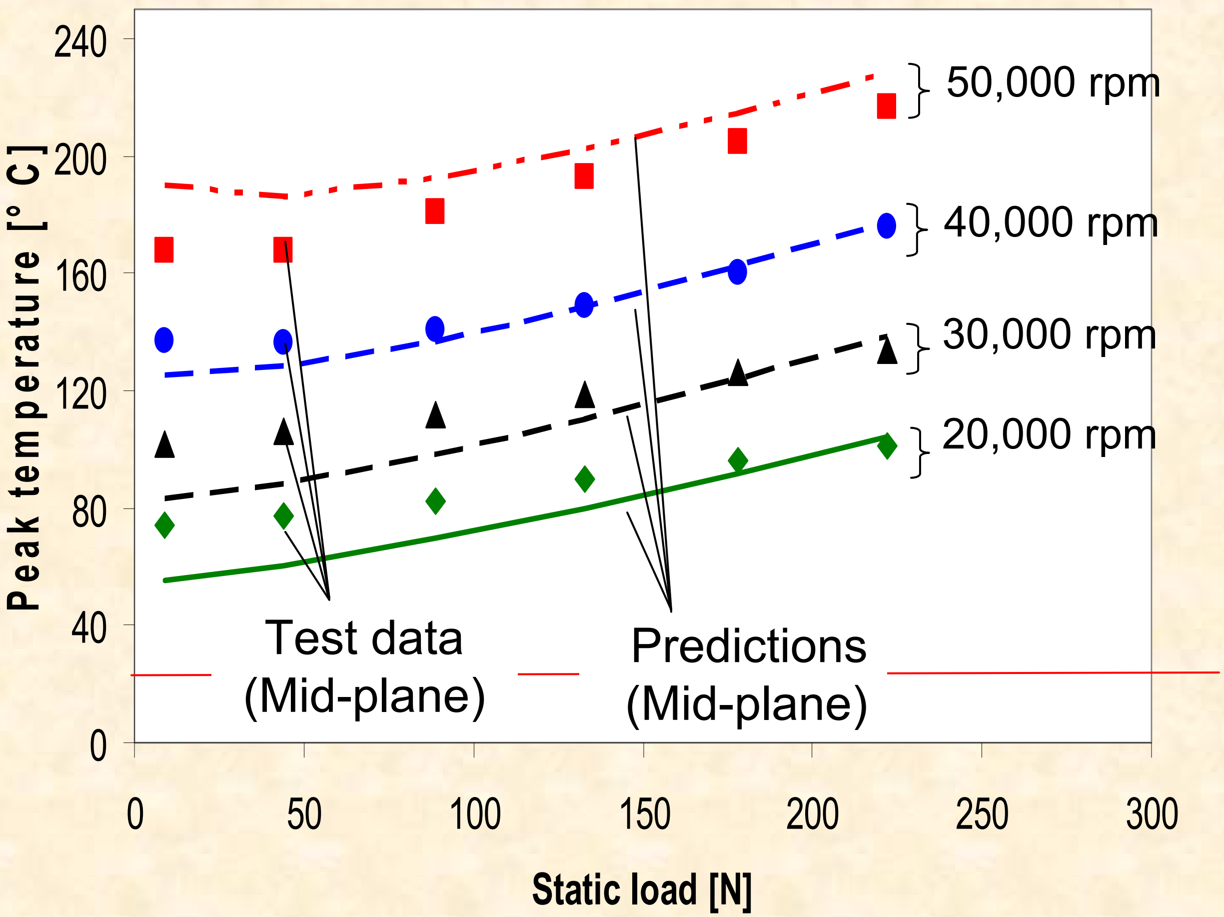
Parameters	Value
Gas properties at 21 °C	
Gas Constant	287 J/(kg-°K)
Viscosity	10^{-5} Pa-s
Conductivity	0.0257 W/m°K
Density	1.164 kg/m^3
Specific heat	1,020 J/kg°K
Ambient pressure	1.014 x 10^5 Pa

Gas viscosity & density & conductivity, foil Young's modulus, and **clearance** change with temperature.

Radil and Zeszotek, 2004
Dykas and Howard, 2004

Peak film temperature

Predictions & test data



T_{Supply}=21°C

Supply air (T_{Supply}), shaft (T_S), and bearing OD (T_B) temperatures at 21 °C.

Read more in ASME Paper GT2009-59919

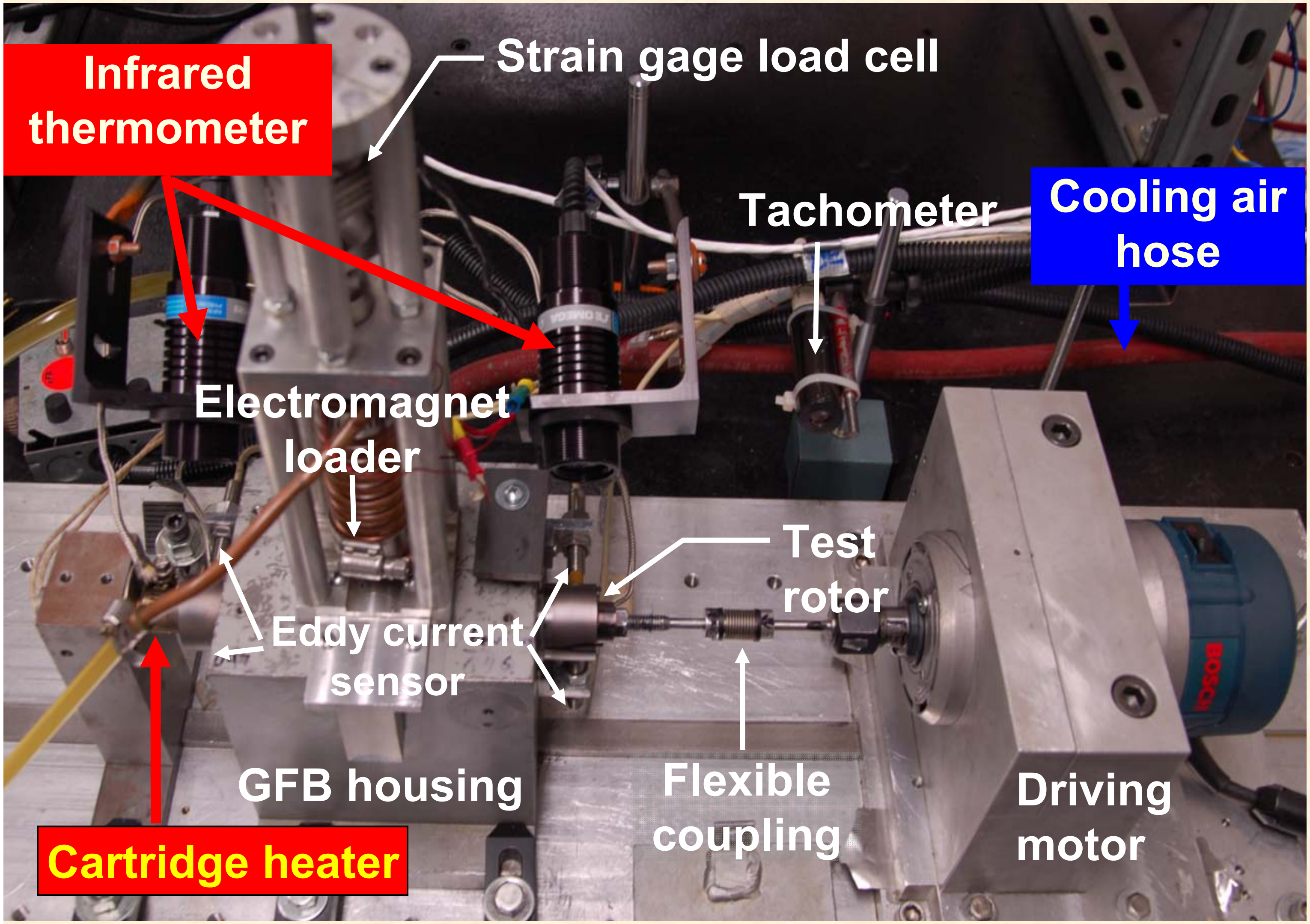
Predictions & test data **Radil and Zeszotek, 2004**

Film temperature + higher than ambient, even for small load of 9 N.

Good agreement preds with test data

2008 rotor-GFB test rig

Max. temp. 130 °C

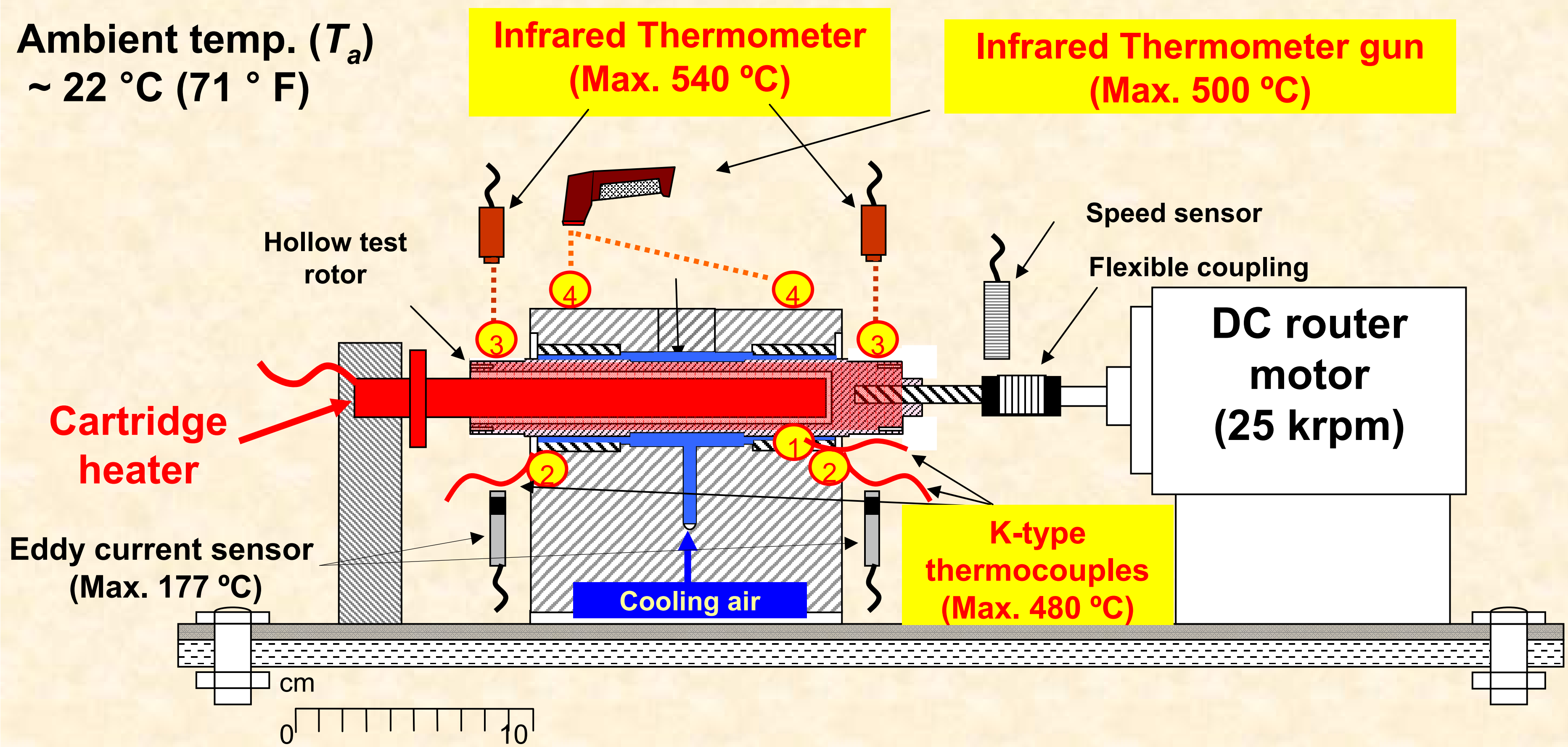


Drive motor (25 krpm).

Cartridge heater max. temperature: 300F

Air flow meter (Max. 100 L/min at 14 psig)

2008 hot rotor-GFB test rig



① Bearing sleeve temperature (at five locations along circumference)

② Bearing outer surface temperature (Drive and bearing and free end bearing)

③ Rotor surface temperature (Drive end and free end)

④ Bearing support (housing) surface temperature (Drive end and free end)

Numbers in circles show locations of temperature measurement.

THD Model Validation Bearings at TAMU

Parameter [mm]	Foster-Miller (2nd gen.)	KIST (1st gen.)	MiTi (2nd gen.)
Bearing cartridge			
Outer diameter	50.85	50.80	44.575
Inner diameter	39.36	37.95	37.921
Top foil and bump strip layer			
Top foil axial length	38.2	38.1	25.4
Top foil thickness	0.100	0.120	0.127
Bump foil thickness	0.100	0.120	0.102
Number of Bumps	25 × 5 axial	26 × 1 axial	24 × 3 axial
Bump pitch	4.581	4.300	4.640
Bump length	3.742	2.100	3.950
Bump height	0.468	0.540	0.510
Bump arc radius	5.581	4.161	4.079
Bump arc angle [deg]	68	59	58

Elastic Modulus 214 GPa,
Poisson ratio=0.29

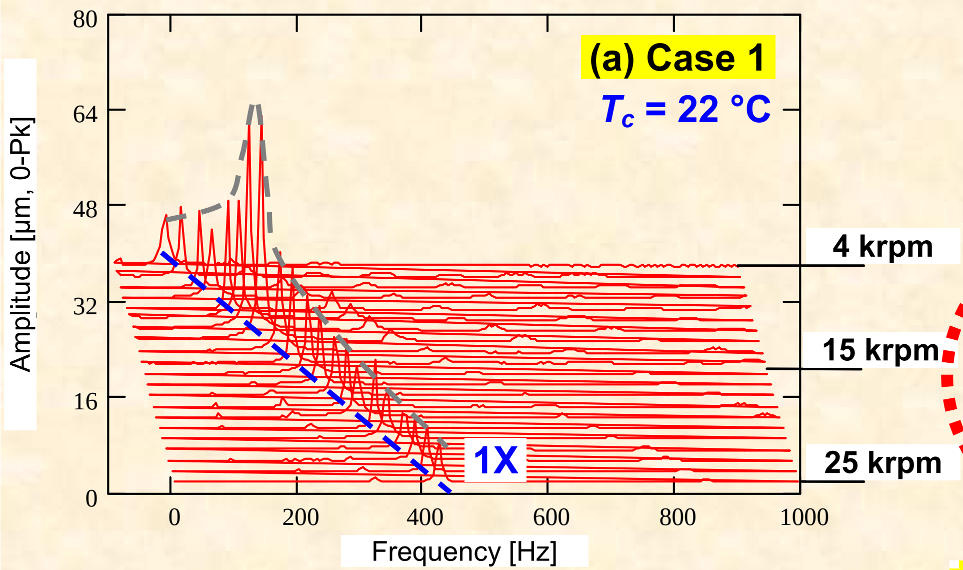
Foster-Miller FB with Teflon® coating (Generation II)

Waterfall plots: coastdown responses

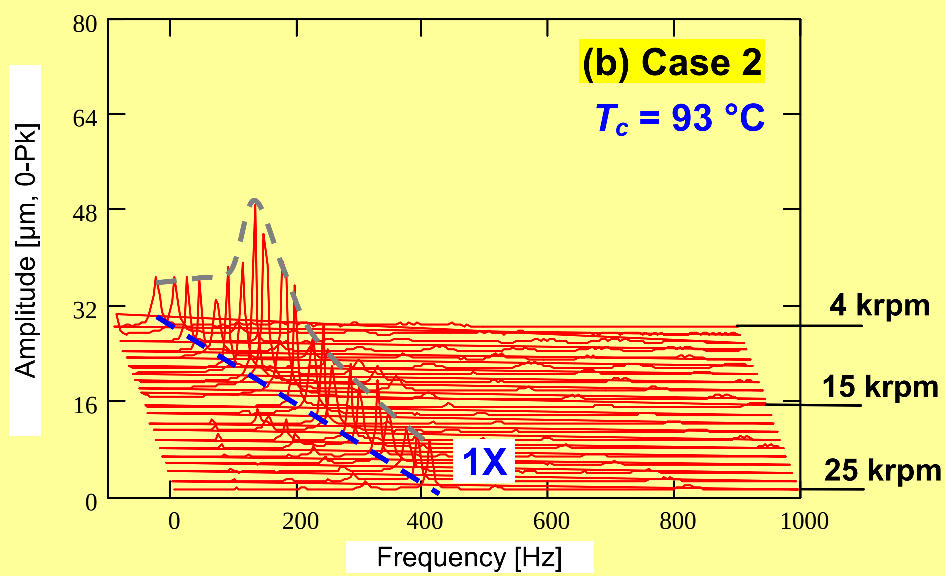
Case 1-3 without cooling flow, T_a ~ 22°C, and T_{hs} = 22°C, 93 °C, and 132°C

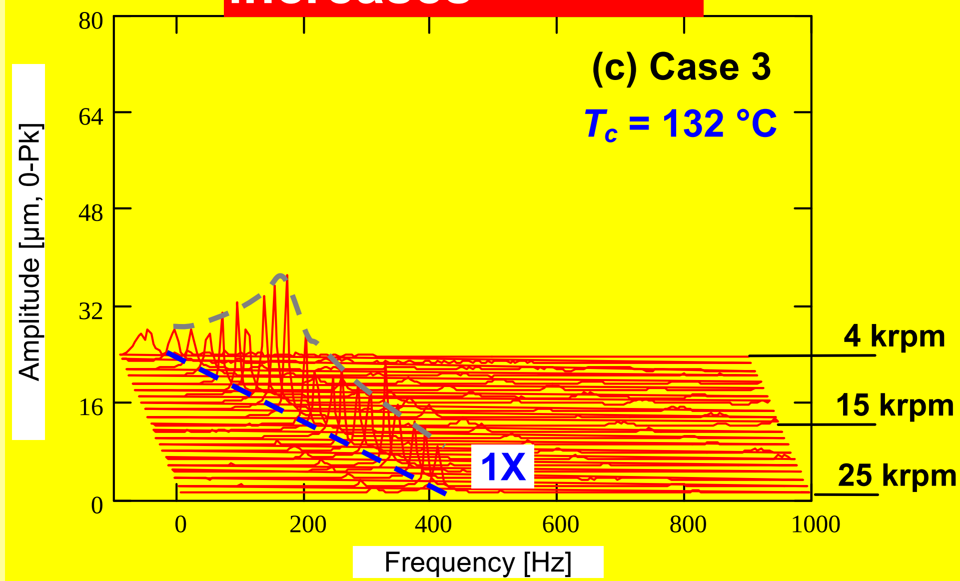
FE – vertical plane

1X component dominant, i.e., no subsynchronous motion, during coastdown tests



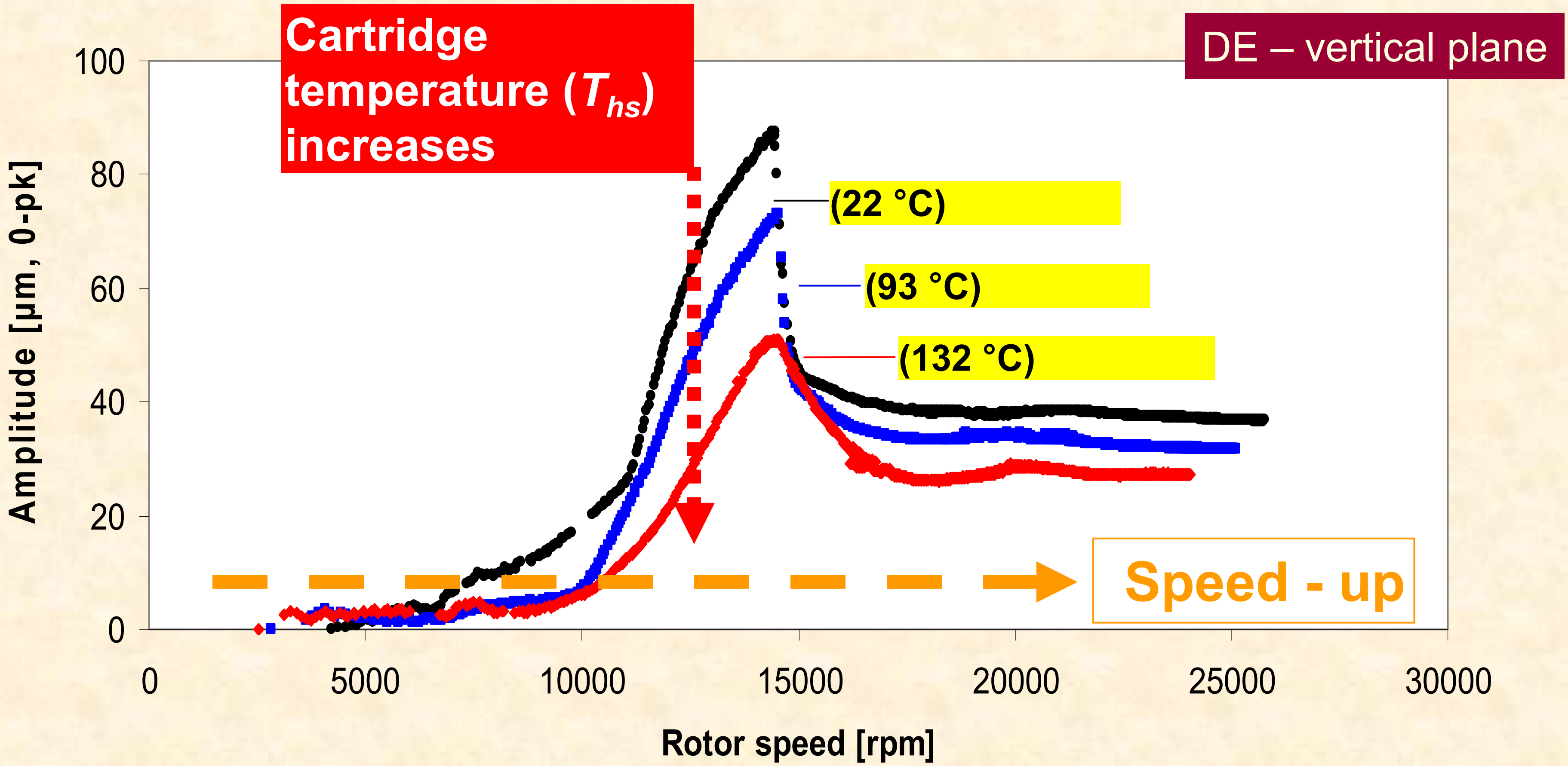
Cartridge temperature (T_{hs}) increases





Rotor speed up 1X response

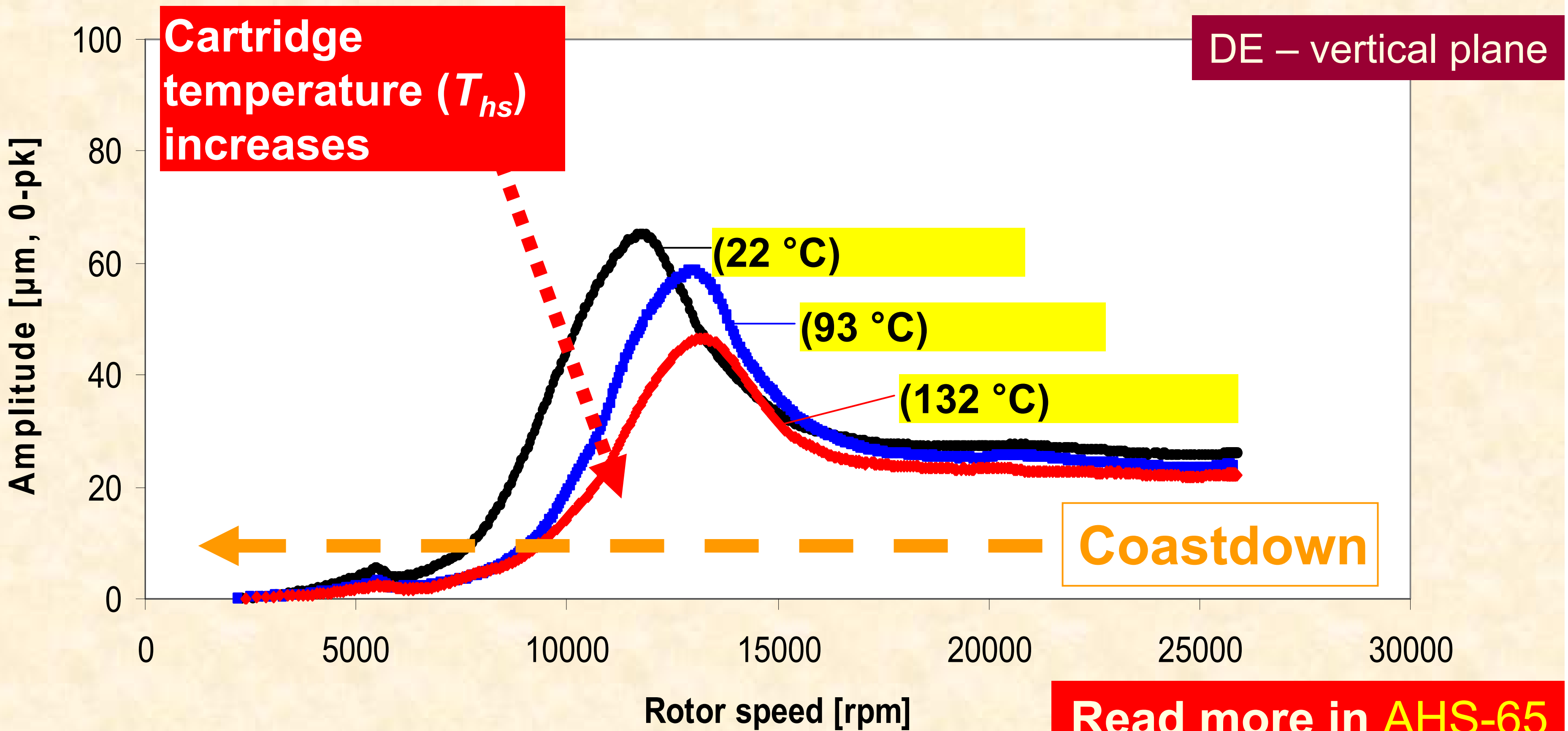
W/o cooling flow, $T_a \sim 22°C$, and $T_{hs} = 22°C, 93\ °C\ \&\ 132°C$



Above critical speed ~ 14.5 krpm, amplitude drops. A nonlinearity! As T_{hs} increases, the peak amplitude decreases.

Rotor coastdown 1X response

W/o cooling flow, T_a ~ 22°C, and T_{hs} = 22°C, 93 °C, and 132°C



Read more in AHS-65 2009 Paper

As T_{hs} increases, critical speed raises by ~ 2 krpm and the peak amplitude decreases. Nonlinearity absent!

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Research Objectives 2008-09

Model validation with TAMU FB test data

- Complete test rig using cartridge heater for high temperature operation (up to 360C)

- Measure rotordynamic performance during speed coastdown from 30 krpm and for increasing shaft temperatures

- Quantify effect of side flow on cooling bearings (max. 150 LPM per bearing)

- **Benchmark model predictions**

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Parameter [mm]	Foster-Miller (2nd gen.)	KIST (1st gen.)	MiTi (2nd gen.)
Bearing cartridge			
Outer diameter	50.85	50.80	44.575
Inner diameter	39.36	37.95	37.921
Top foil and bump strip layer			
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Top foil thickness	0.100	0.120	0.127
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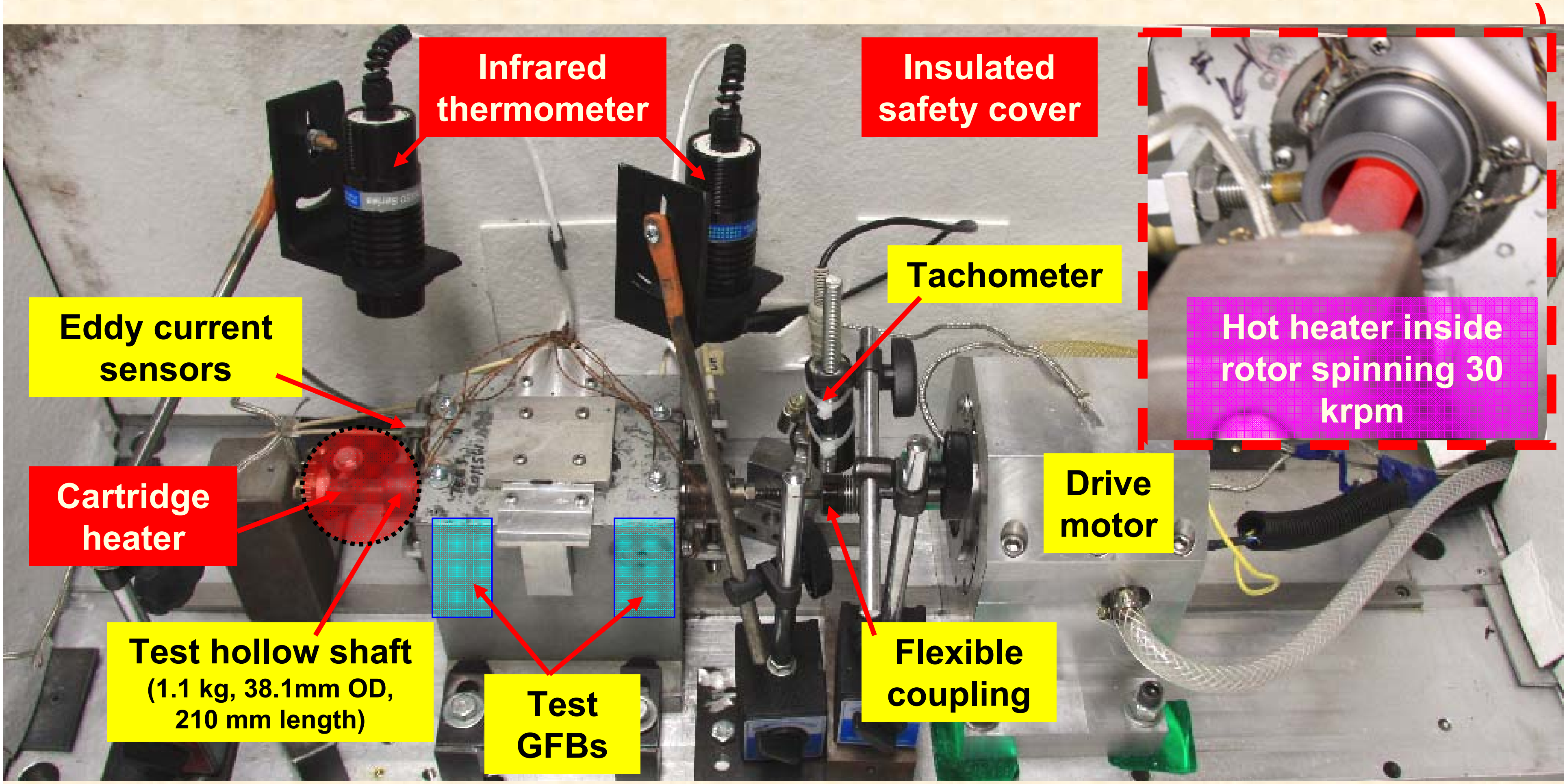
Bump arc radius	5.581	4.161	4.079
Bump arc angle [deg]	68	59	58

Elastic Modulus 214 GPa,
Poisson ratio=0.29

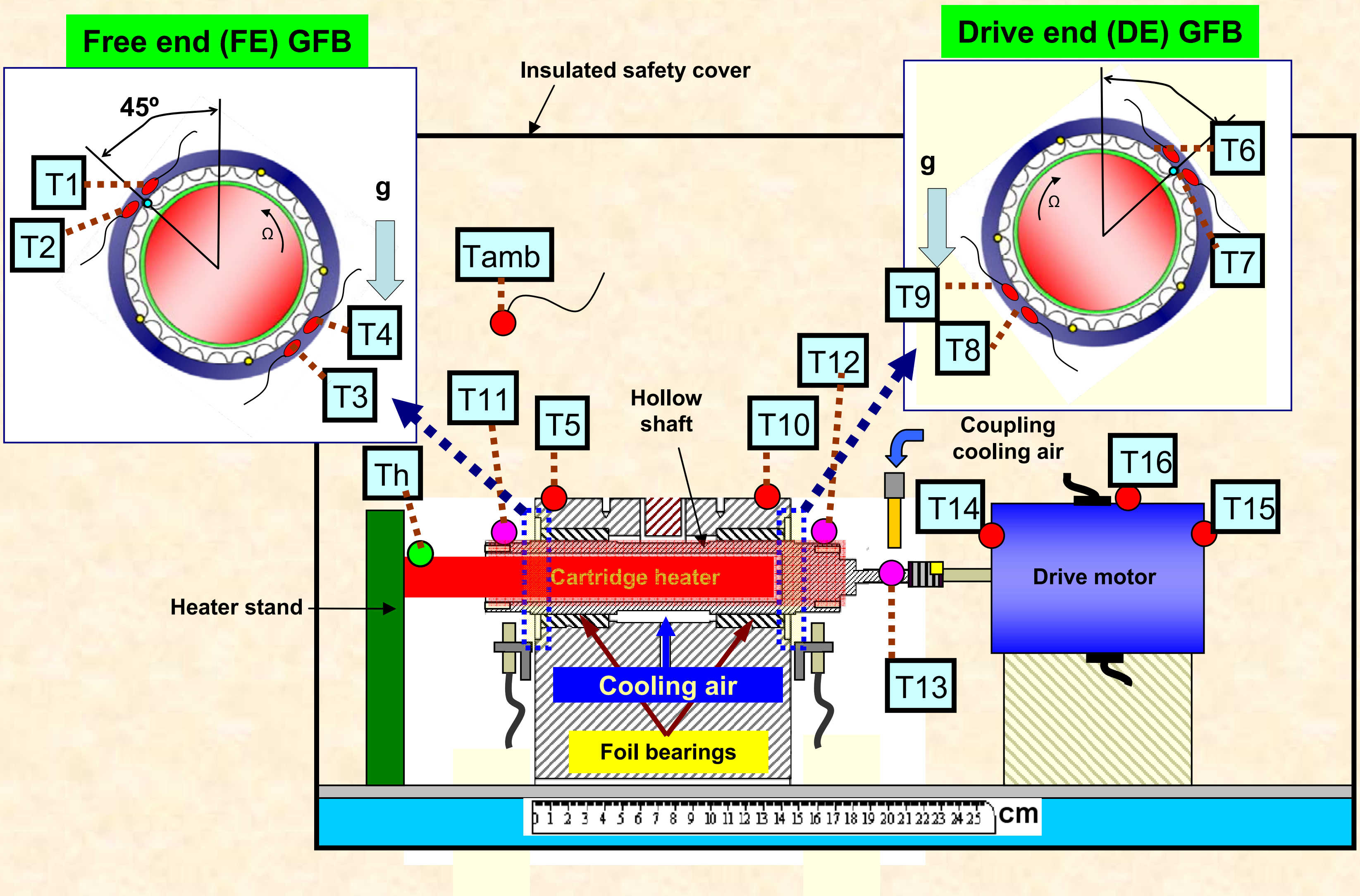
2009 hot rotor-GFB test rig

Max. 360 °C

Instrumentation for high temperature. Insulation casing
Gas flow meter (Max. 500 LPM). Drive motor (max. 65 krpm)



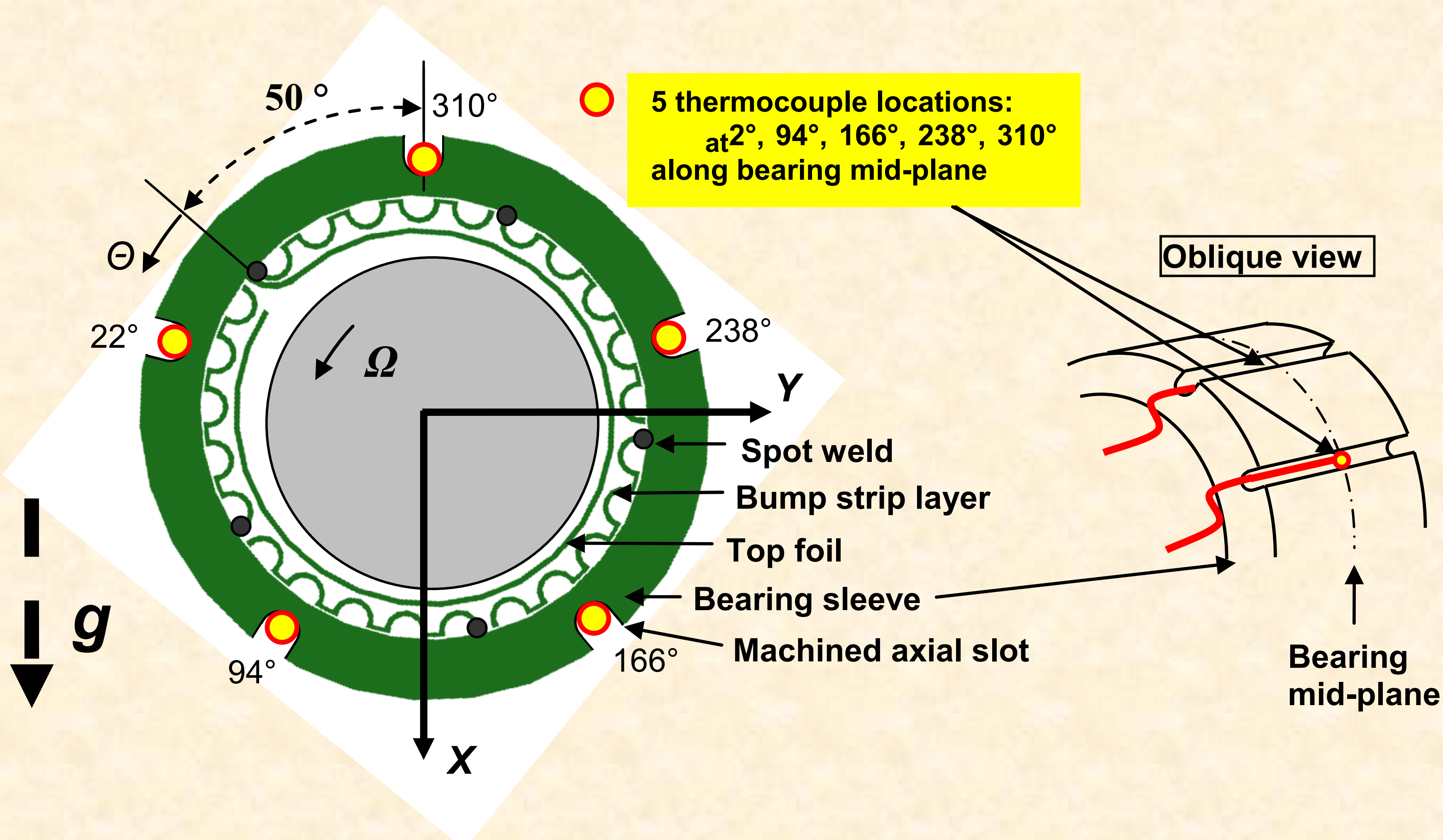
Thermocouples in test rotor-GFB rig



Overall 15 thermocouples for GFB cartridge outboard, Bearing support housing surface, Drive motor, Test rig ambient, and Cartridge heater temperatures

Two noncontact infrared thermometers for rotor surface temperature

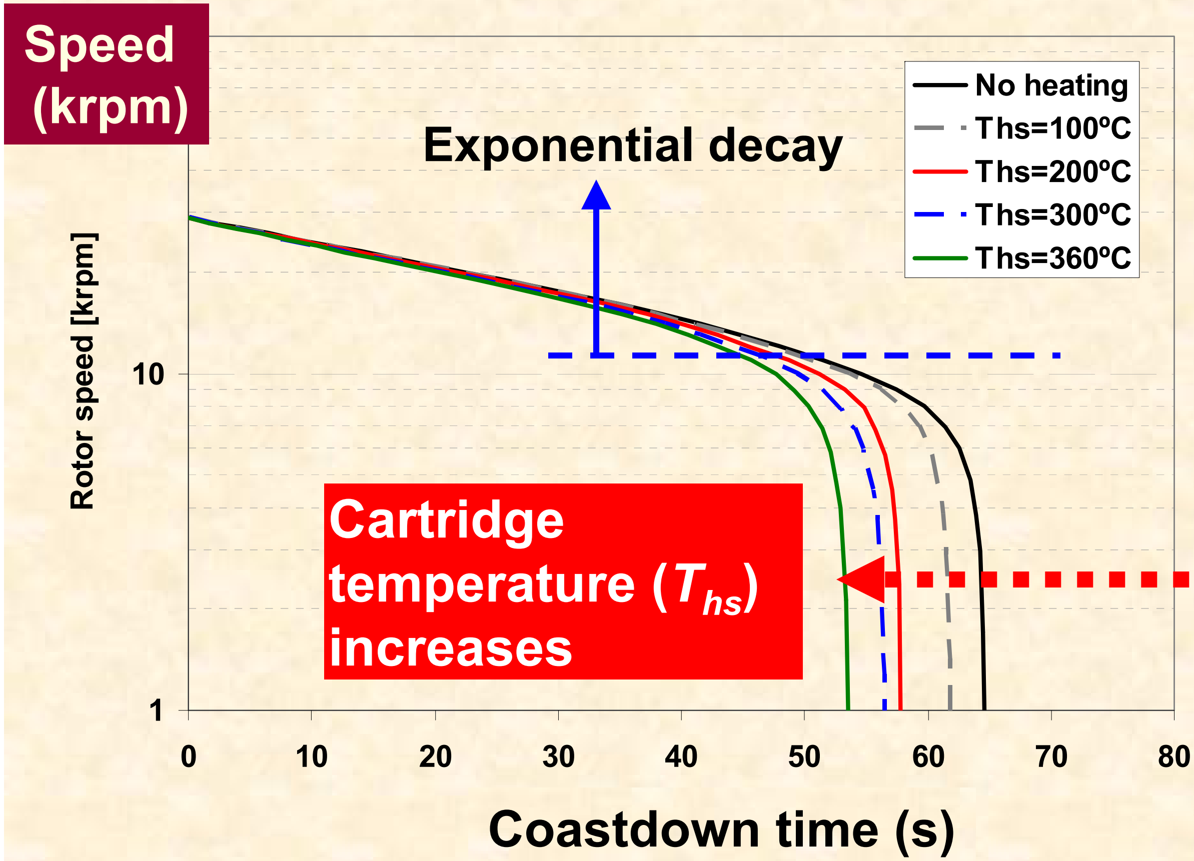
Thermocouples in test GFB



Foster-Miller FB uncoated (Generation II)

five (5) thermocouples placed within machined axial slots.

Time to coast down rotor Effect of shaft temperature



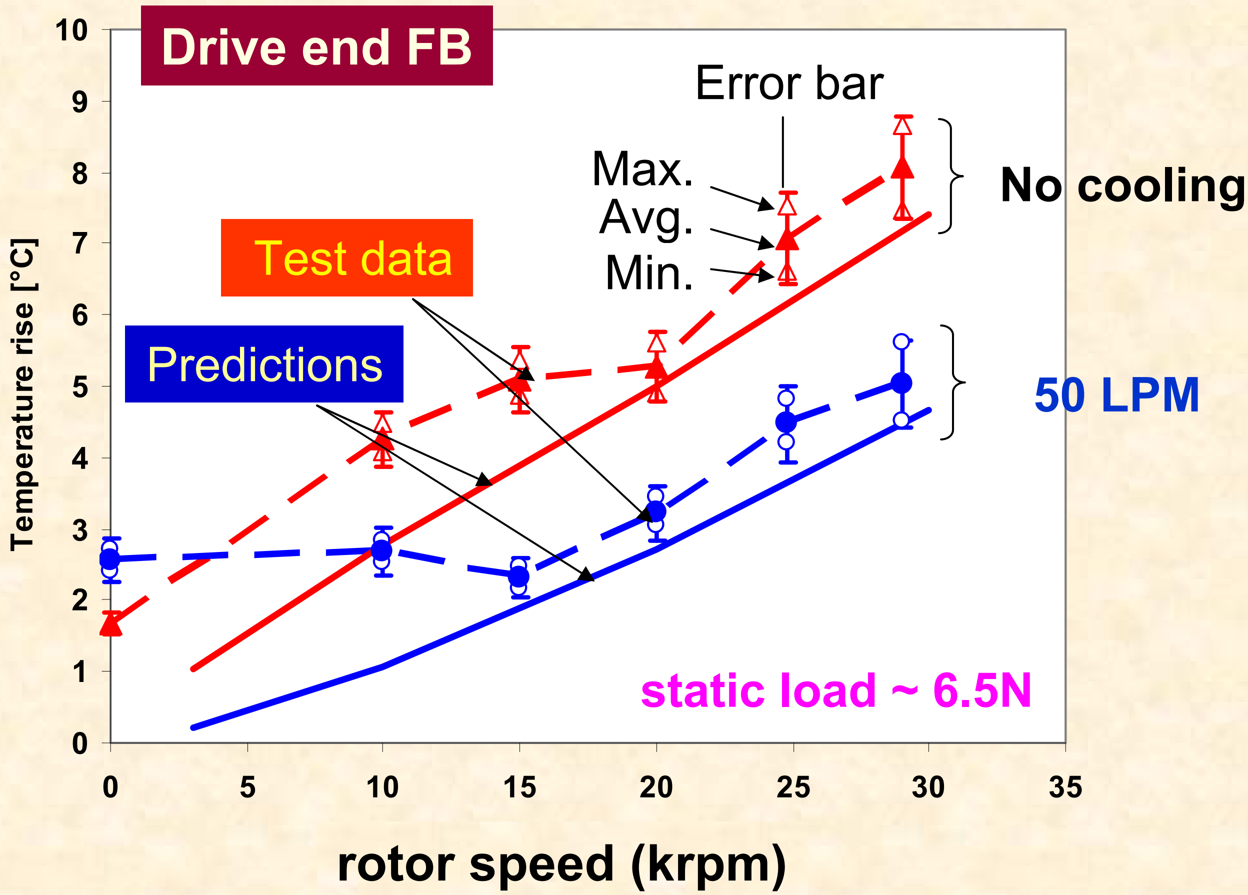
Baseline,
Heater up to 360C.
No forced cooling

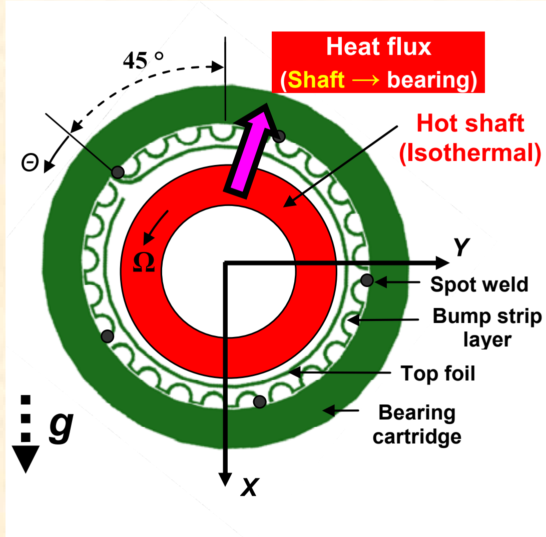
Long time to coastdown : very low viscous drag

(no contact between rotor and bearings)

Test Data **Coastdown time lesser as rotor heats (reduced clearance)**

Bearing outboard temperature predictions & test data



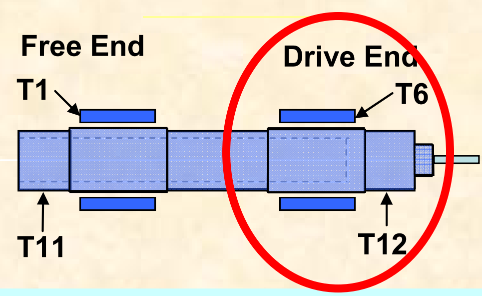


Room temperature 21 °C.

Heater OFF

w/o and w low cooling

Rotor speed : 30 krpm

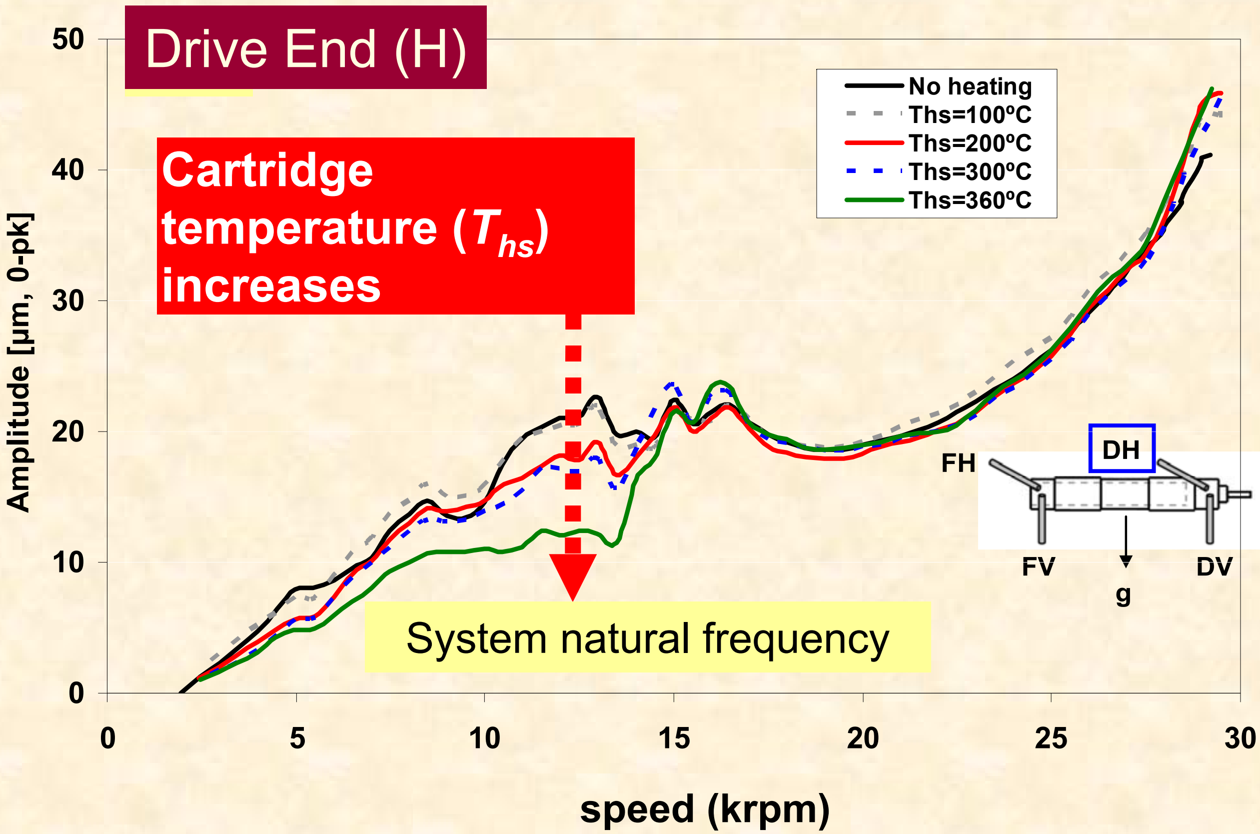


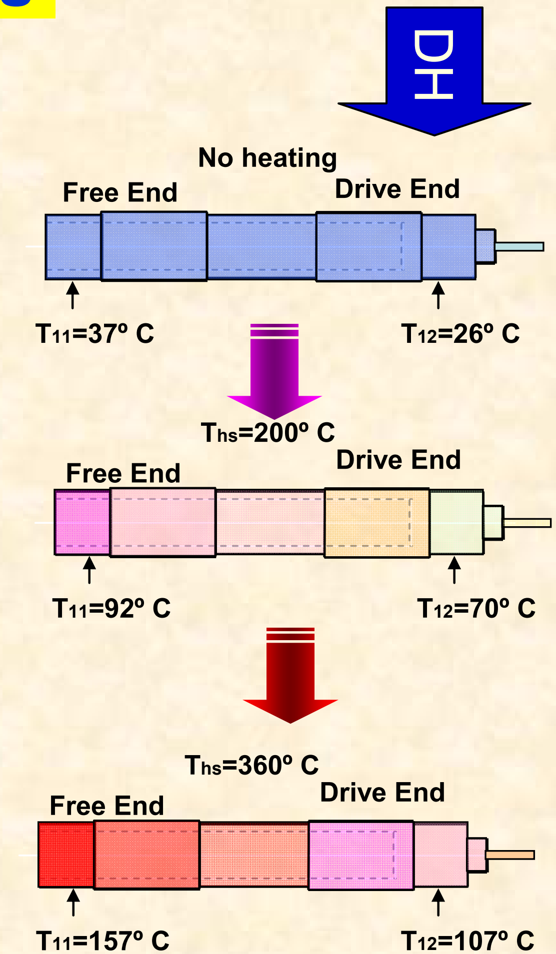
Test data & predictions

FB OD temperature rises with rotor speed and decreases with forced cooling stream ~ 50 LPM. Predictions agree with test data

1X response as rotor heats Tests

Baseline. Heater to 360C. No forced cooling



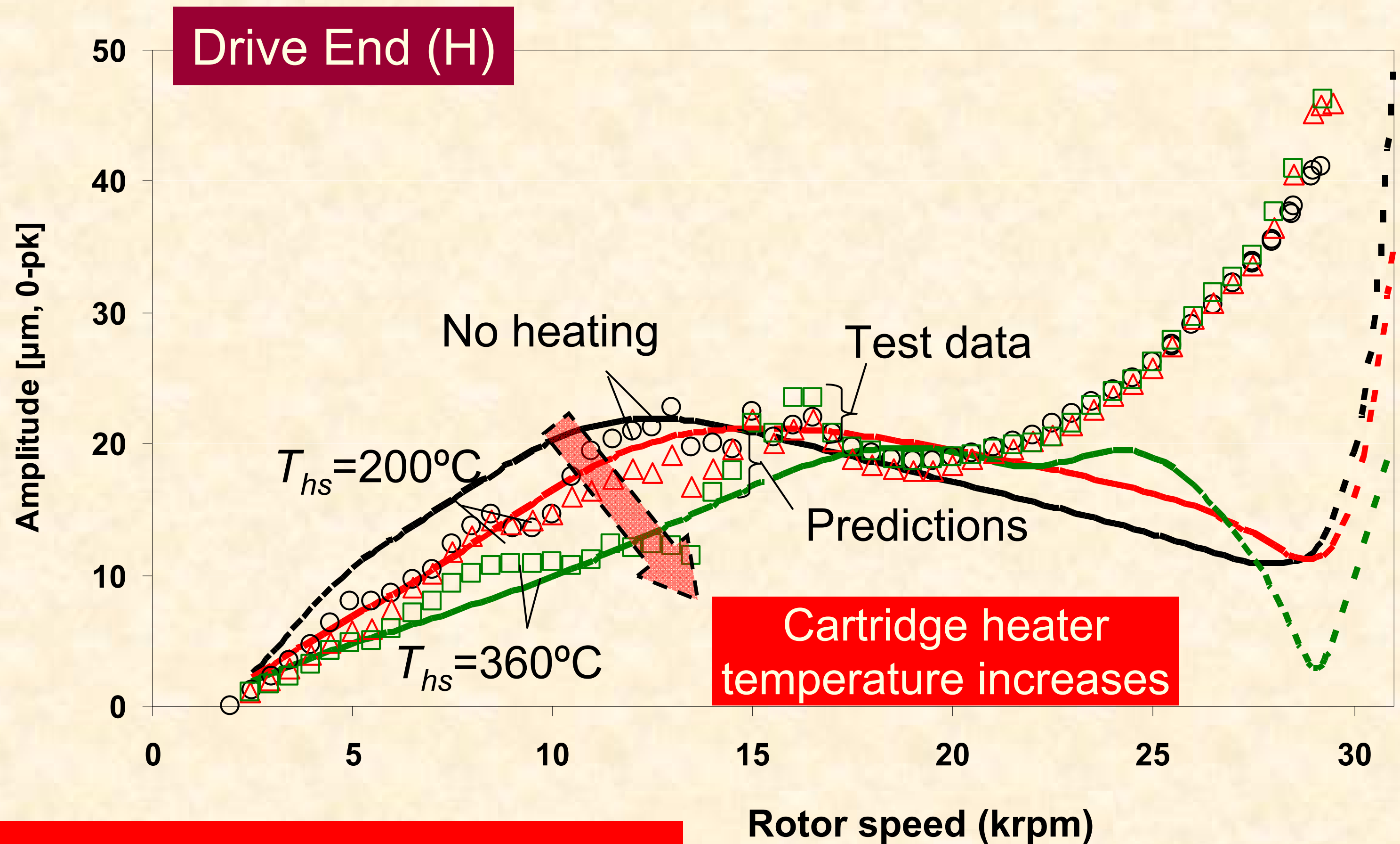


Test Data

As heater *T* increases to 360ºC, peak motion amplitude decreases in speed range 7 krpm to 15 krpm

1X rotor response predictions & tests

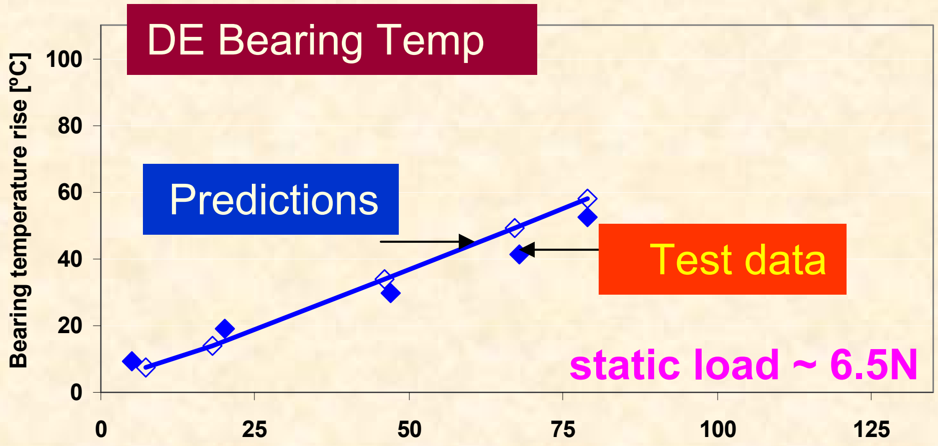
Baseline. Heater to 360C. No forced cooling

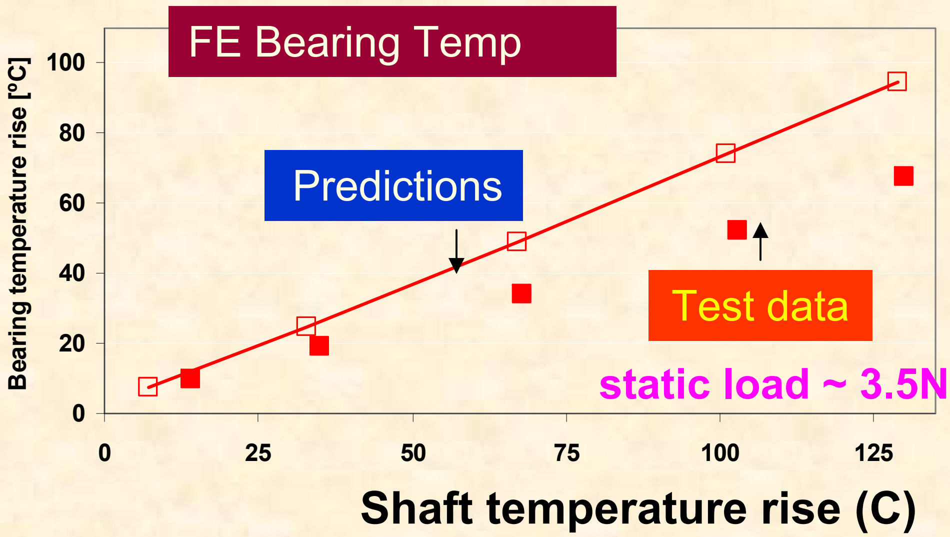


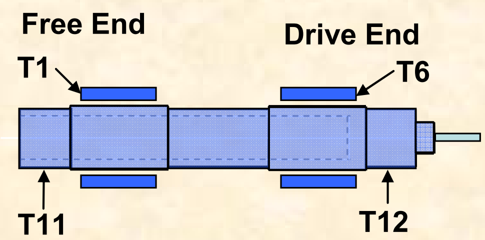
Test data & predictions

As heater temperature raises, rotor amplitude decreases for speed < 15 krpm & the critical speed increases from 14 krpm to 17 krpm

Bearing outboard temperature predictions & tests







Supply air (T_{Supply}) ~ 21 °C.

Heater up to 360C.

No cooling flow

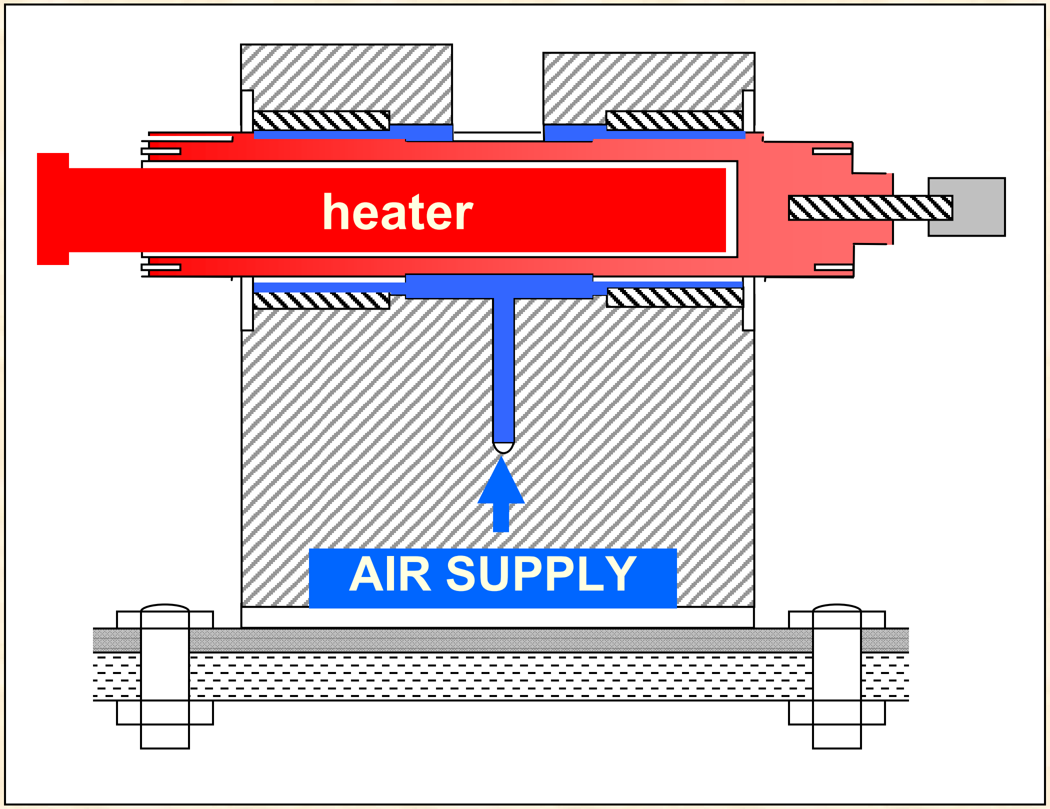
Rotor speed : 30 krpm

Test data & predictions

FB cartridge temperature increases linearly with shaft temperature

Cooling gas flow into GFBs

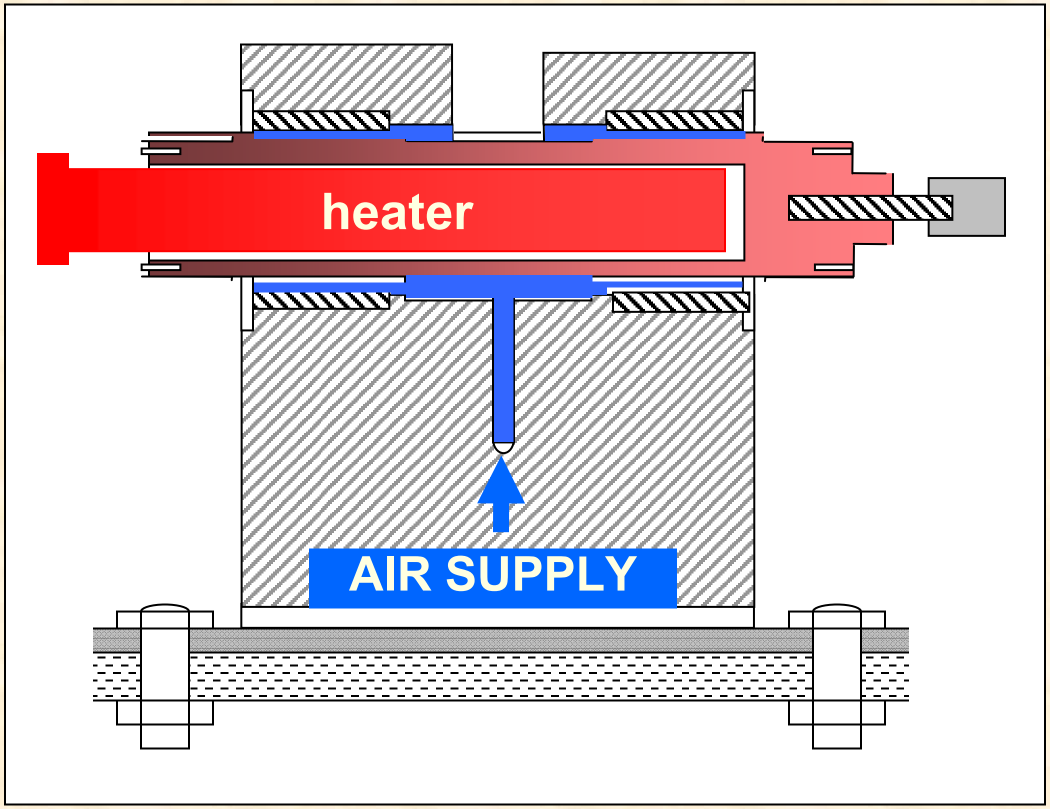
Gas pressure Max. 100 psi



Cooling flow needed for thermal management: to remove heat from shear drag or to reduce thermal gradients in hot/cold engine sections

Cooling gas flow into GFBs

Gas pressure Max. 100 psi



Heater warms unevenly rotor.

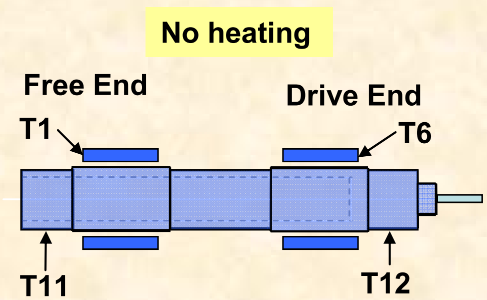
Side cooling cools unevenly rotor and also heater

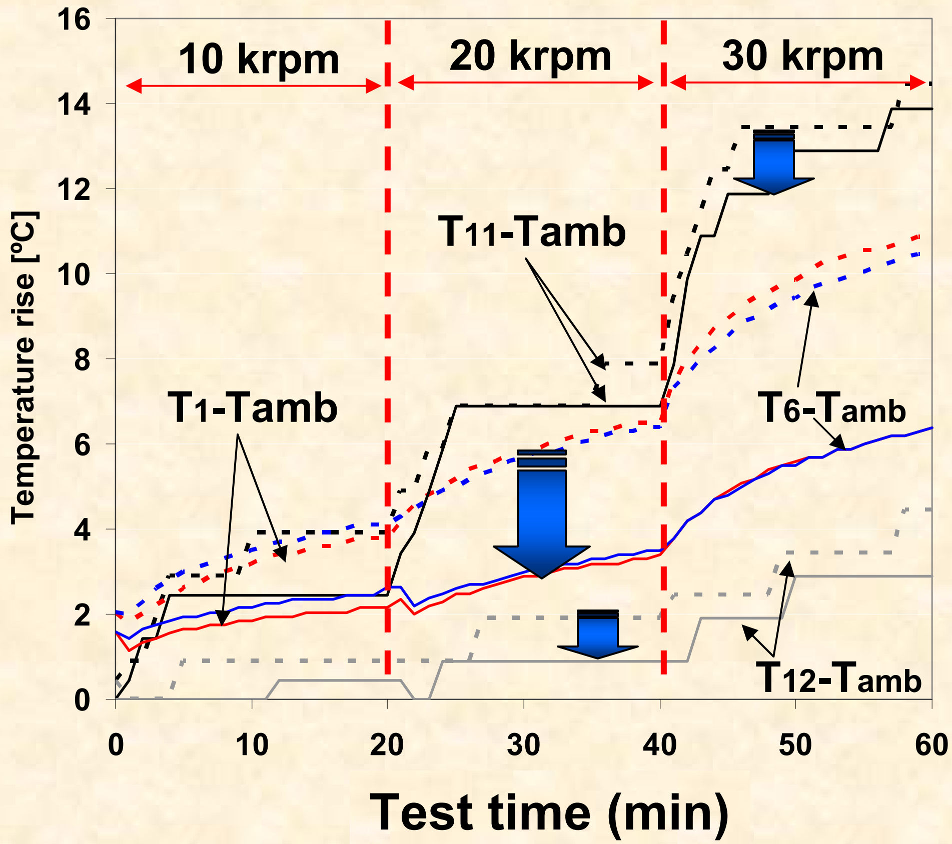
Heating of rotor

Effect of rotor speed and side cooling

Baseline imbalance, No side flow & 50 L/min

: Temp. drop due to 50L/min cooling flow





Bearing cartridge and rotor temperatures increase steadily with time

Rotor speed : 10, 20, 30 krpm

Rotor speed makes rotor and bearings hotter

Cooling flow removes heat from shear dissipation in rotor, most effective at high speed

Test Data **Heater OFF**

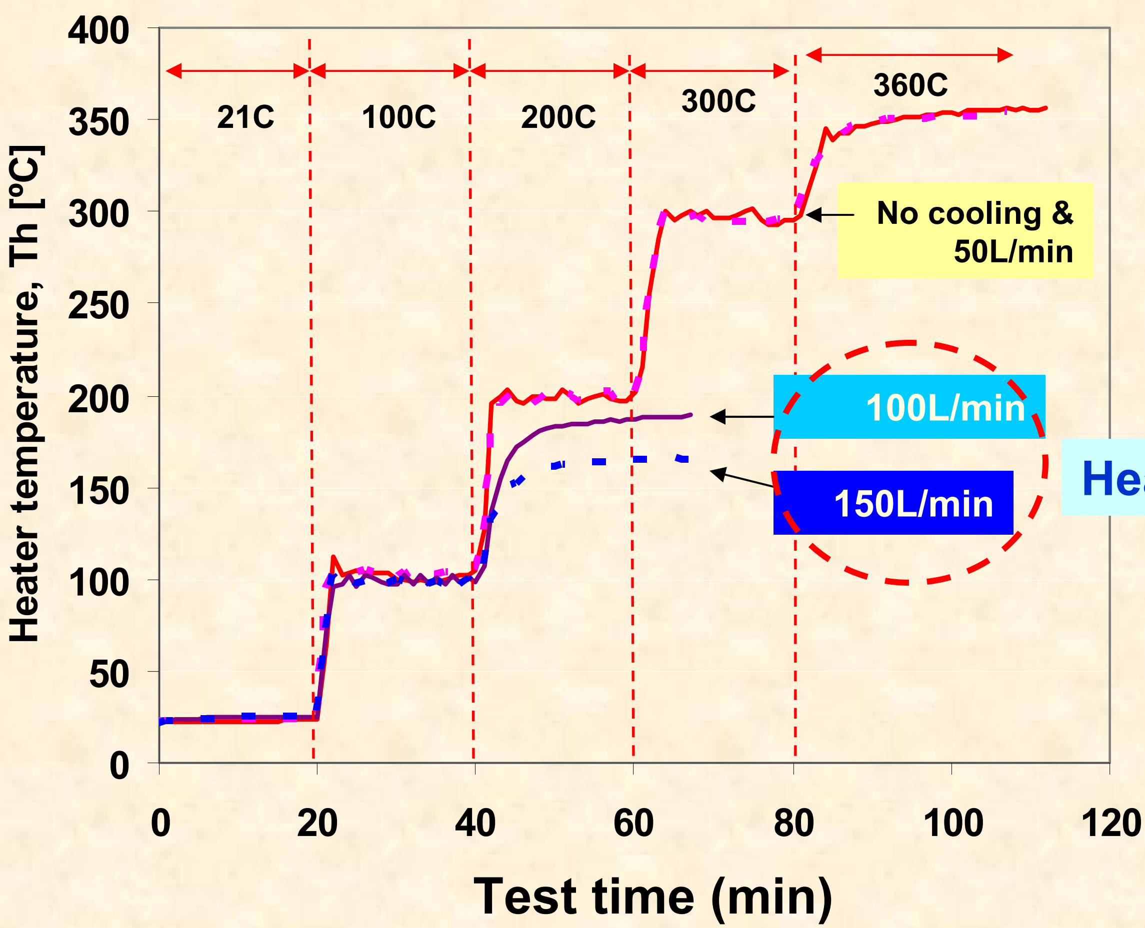
Heater temperature Effect of cooling flow

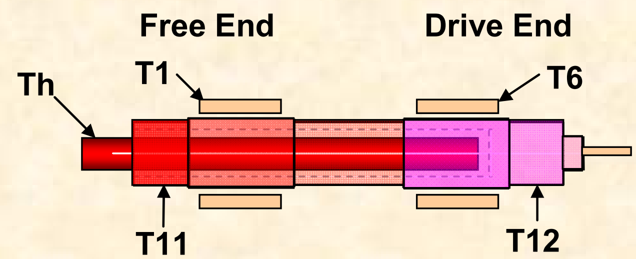
Heater temperature increases

Heater up to 360C

w/o & w cooling flows

Rotor speed : 30 krpm

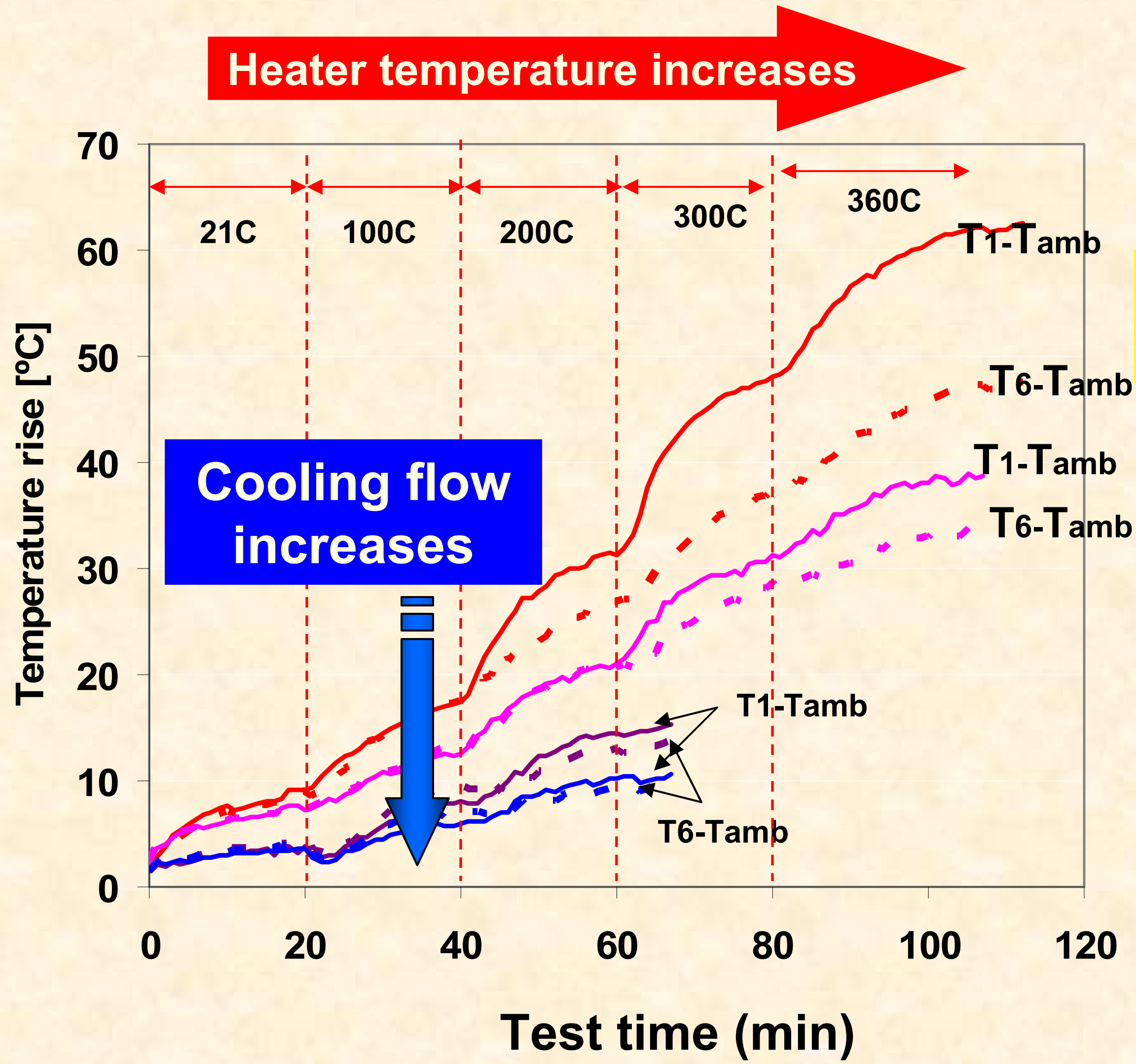


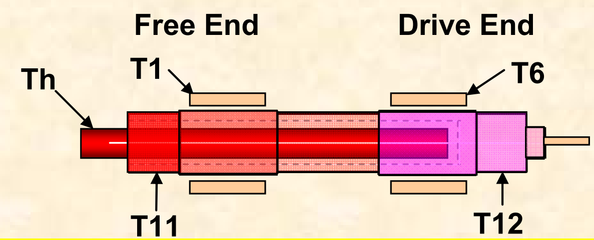


Heater power is limited

Test Data Cooling>100 LPM cools both rotor & HEATER!

Bearings OD temperatures Effect of cooling flow





High temp. (heater up to 360C). Cooling flow up to 150 L/min

rotor speed : 30 krpm

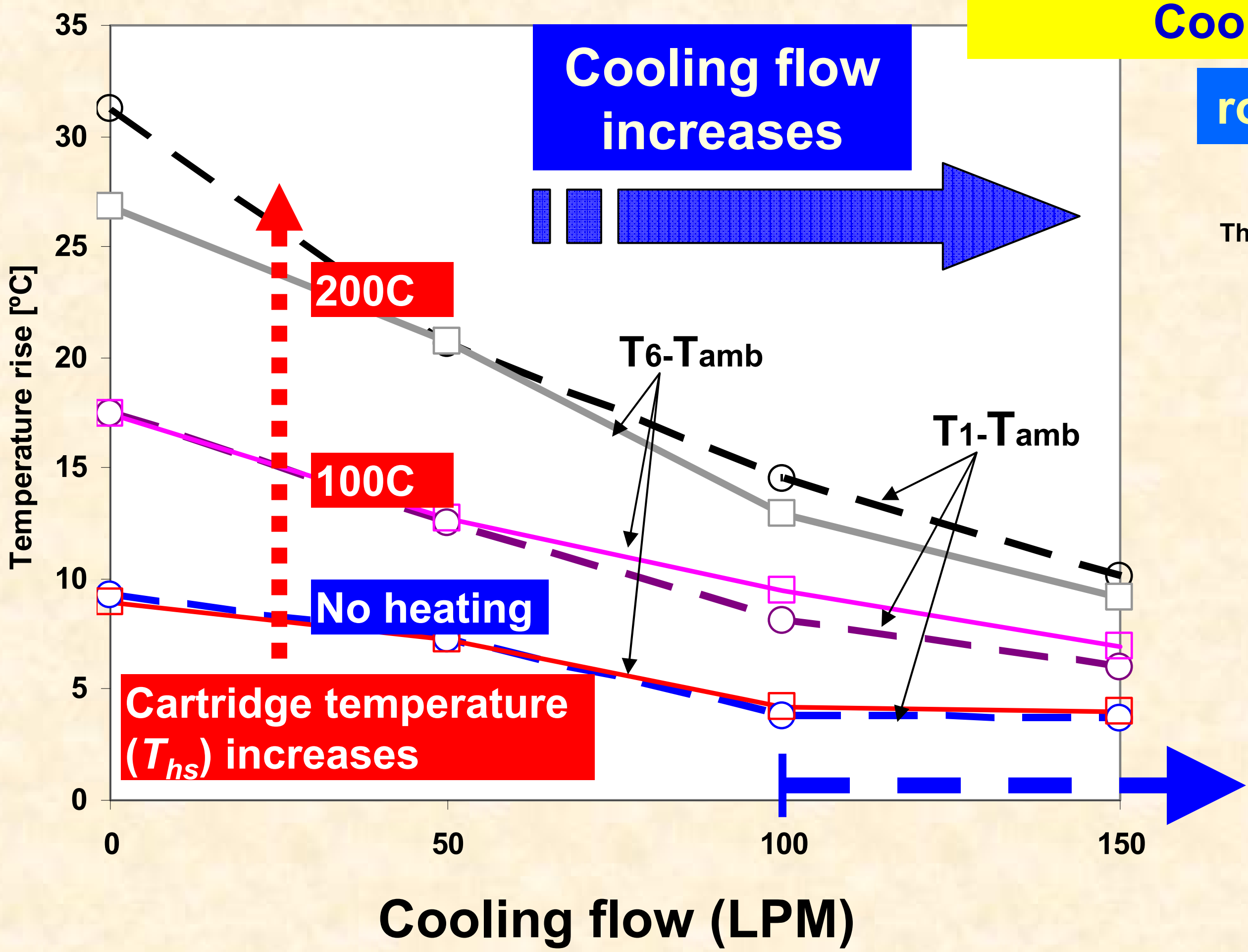
Cooling effective > 100 LPM and when heater at highest temperature

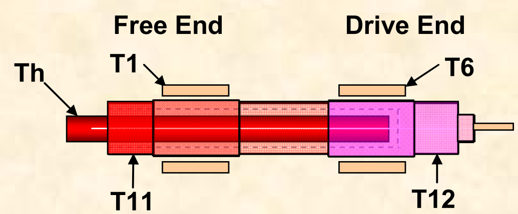
Test Data

Bearing cartridge temperature — Effect of cooling flow

High temp. (heater up to 360C). Cooling flow to 150 LPM

rotor speed : 30 krpm



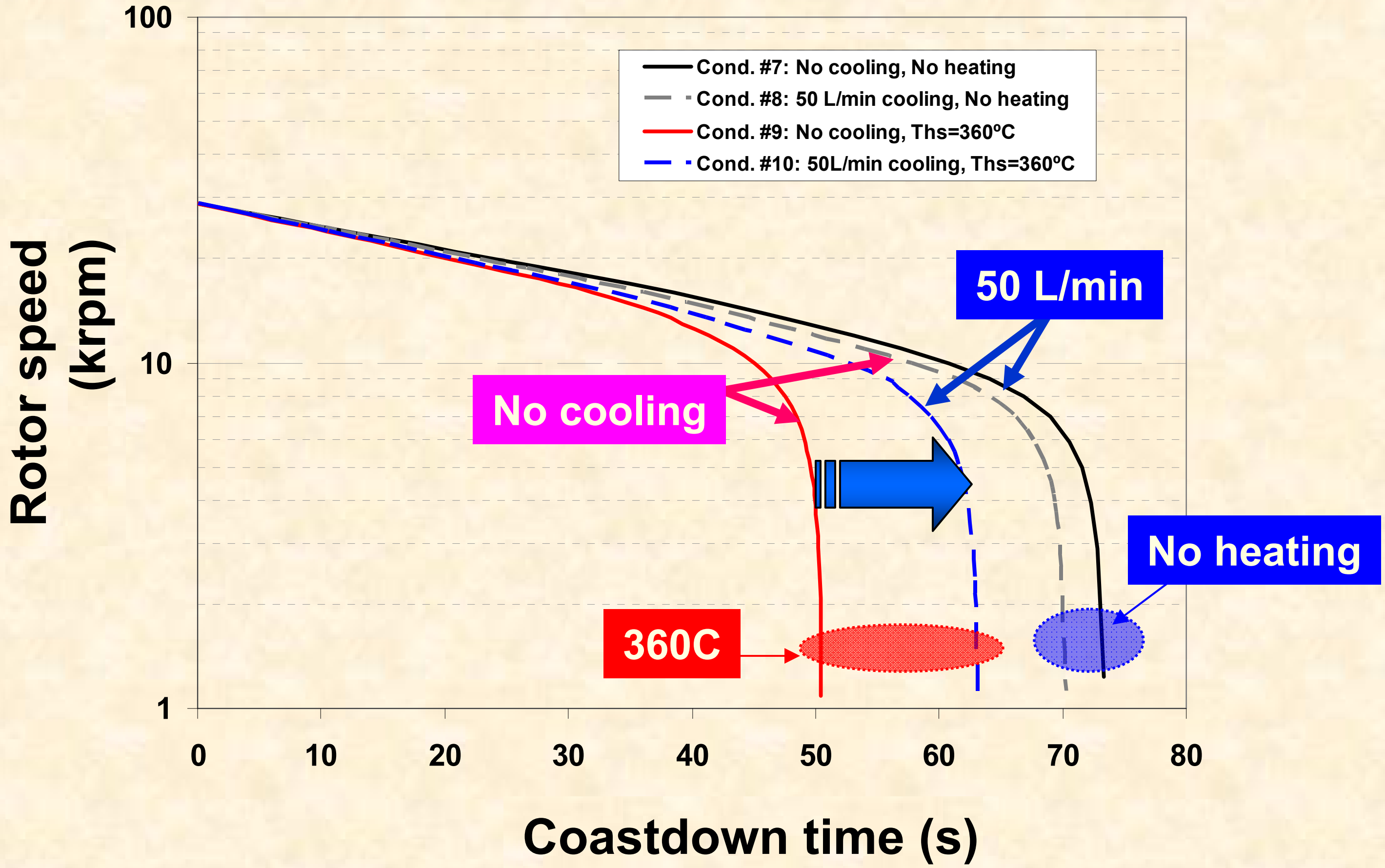


Turbulent flow > 100 LPM

Test Data

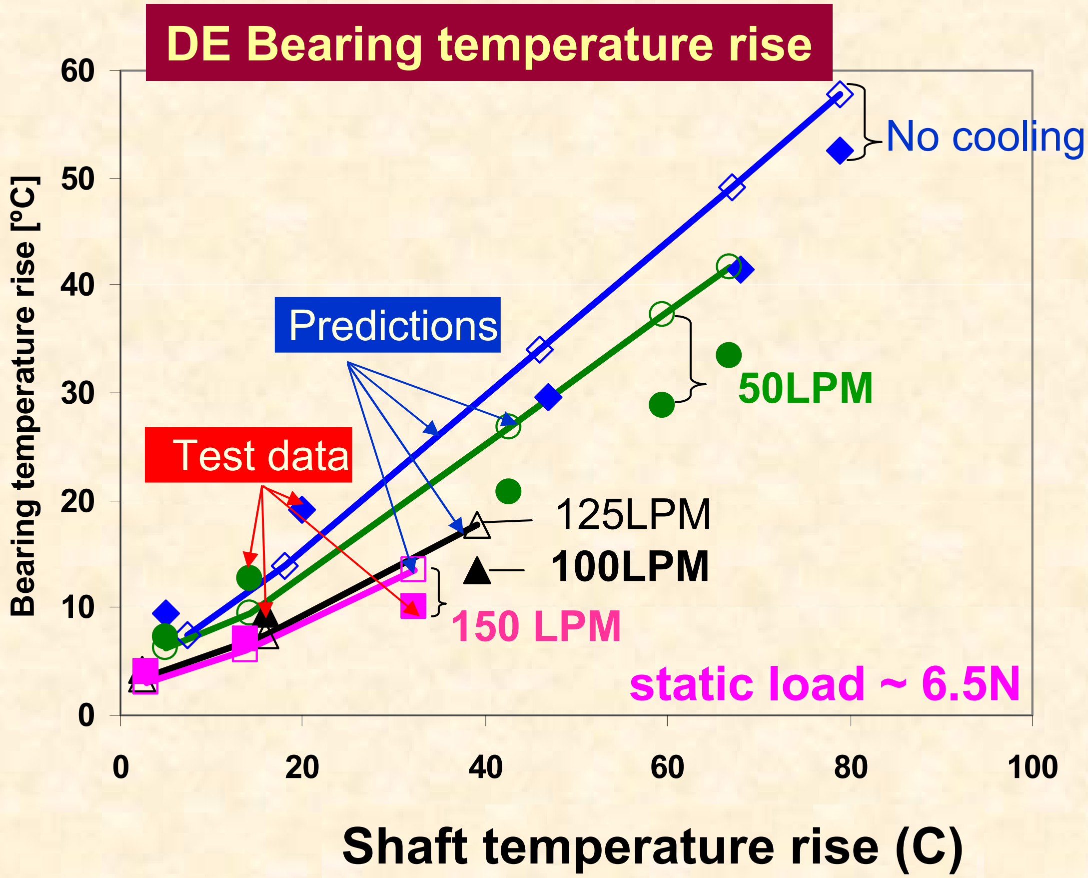
Bearing OD temperature decreases with cooling flow

Time to coast down rotor Effect of cooling flow



Test Data Coastdown time down by 20% (13 s) with cooling at 50 LPM

Bearing cartridge temperature Predictions & tests

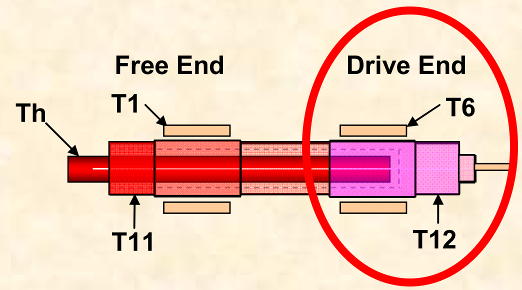


Supply air (T_{Supply}) ~ 21 °C.

Heater up to 360C.

w/o & w cooling flow

Rotor speed : 30 krpm



Test data & predictions

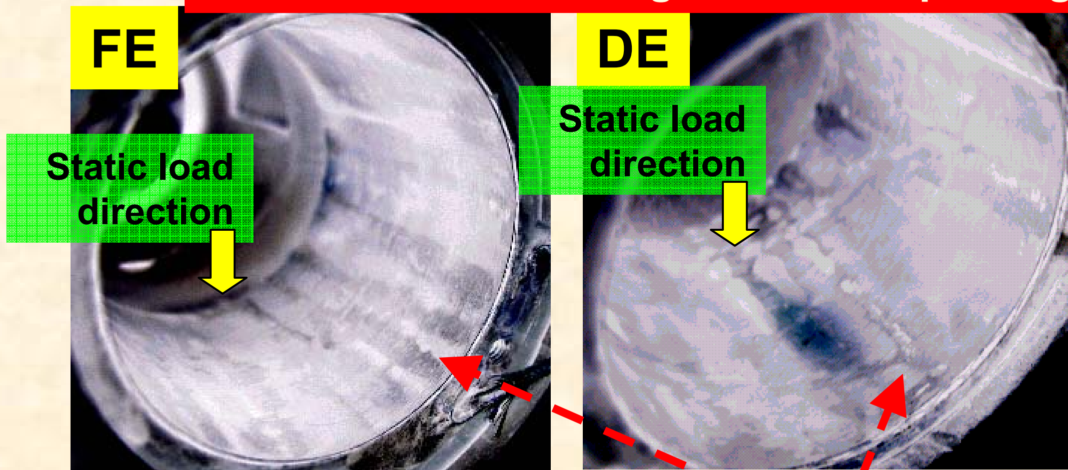
As cooling flow rate increases, FB cartridge temperature decreases. Predictions agree with test data.

Post-test condition of rotor and GFBs

Before operation

UNCOATED top foil !

After extensive heating with rotor spinning



Wear marks on top foils are at side edges

Before operation

After extensive hearing with rotor spinning tests

Rotor shows polishing marks at bearing locations. Deep wear marks at outboard edges

Test data & predictions

- Amplitudes of rotor synchronous motion proportional to added imbalances.
- For operation with hot shaft, amplitude of rotor motion drops while crossing (rigid body mode) critical speed.
- As rotor and bearing temperatures increase, air becomes more viscous and bearing clearances decrease; hence coastdown time decreases.
- Thermal management with axial cooling streams is beneficial at high temperatures and with large flow rates ensuring turbulent flow conditions.
- Test foil bearings continue to survive high temperature & high vibration operation!

Topic

- Statement of Work & Sources for Presentation
- Objectives and accomplished work in 07-08
  - Computational model. Validation with published data.
  - Rotordynamic measurements at TAMU
- Objectives and accomplished work in 2008-09
  - Description of test rig and foil bearings at TAMU
  - Effect of temperature on bearing temperatures, coastdown speed and rotor motions
  - Effect of cooling flow on bearing and shaft temperatures. Validation of computational model
- **The computational code**
  - **Graphical User Interface. Further predictions**
- GFB thermal management tests and preds.
- Closure & added value

The computational program

• Windows XP OS and MS Excel 2003 (minimum requirements)

• Fortran 99 Executables for FE underspring structure and gas film analyses. Prediction of forced – static & dynamic- performance.

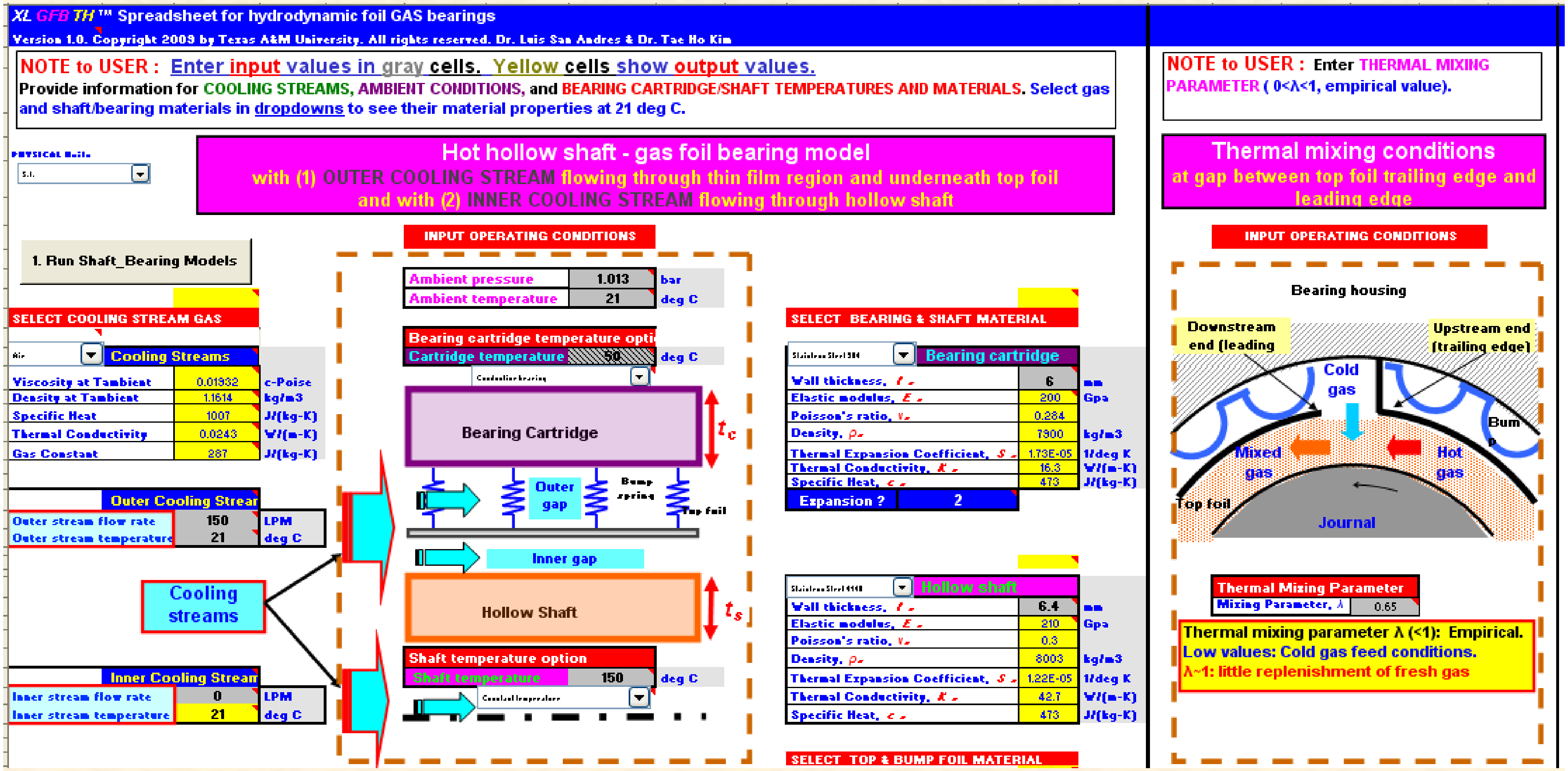
• Excel® Graphical User Interface (US and SI physical units). Input & output (graphical)

• Compatible with XLTRC^2 and XLROTOR codes

Code: XL_GFB_THD

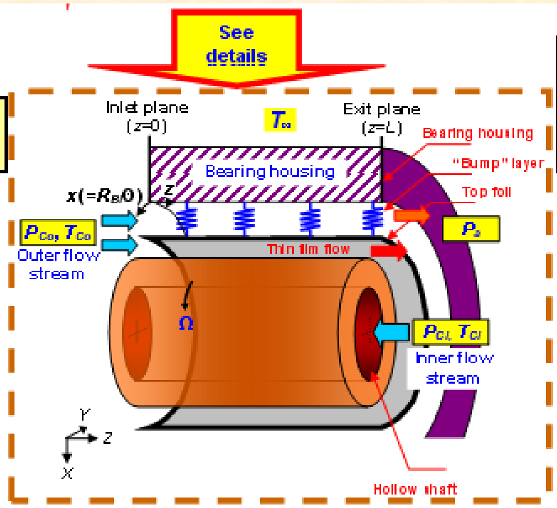
Delivered on June 2009

Graphical User Interface



Worksheet: Shaft & Bearing models (I)

Graphical User Interface



Modify Tables 1 and 2 below, if need more materials for gas, bearing, shaft and foil

Inconel X750	Top & Bump foil	
Elastic modulus, E_T	214	Gpa
Poisson's ratio, ν_T	0.29	
Density, ρ_T	8303	kg/m3
Thermal Expansion Coefficient, δ_T	1.21E-05	1/deg K
Thermal Conductivity, K_T	16.9	W/(m-K)
Specific Heat, c_T	440	J/(kg-K)

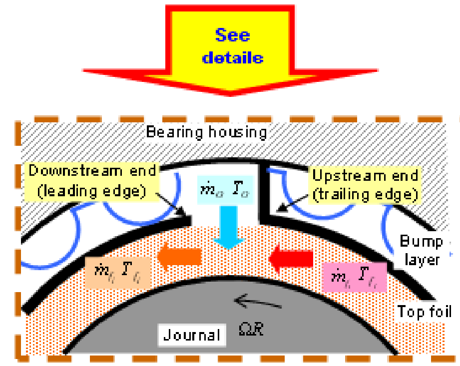


Table 1. DATABASE (COMMON COOLING GAS MATERIAL PROPERTIES)

	Viscosity at Tambient: Viscosity (T)=A * T deg K + B			Thermal Conductivity at Tambient: Tconducticity (T)=C * T deg K + D			Specific Heat t Tambient: Specific heat (T)=E * T deg K +			Gas Constant	Density	Temperature
	21 deg C	A	B	21 deg C	C	D	21 deg C	E	F	21 deg C	21 deg C	
	μ-Poise			W/(m-K)			J/(kg-K)			J/(kg-K)	kg/m3	deg C
Air	1.93E-02	4.00E-08	#####	0.0243	7.00E-05	0.0042	1007	0.1602	953.08	287	1.1614	21
Helium	2.00E-02	4.00E-08	#####	0.149	7.00E-05	0.0042	5188	0.1602	953.08	2077	0.1637	21
Carbon Dioxide	7.03E-02	4.00E-08	#####	0.087	7.00E-05	0.0042	839	0.1602	953.08	189	1.98	21
Nitrogen	1.80E-02	4.00E-08	#####	0.026	7.00E-05	0.0042	1040	0.1602	953.08	297	1.1468	21
Other												

Table 2. DATABASE (COMMON SHAFT & BEARING CARTRIDGE & FOIL MATERIALS)

	Elastic modulus at Tambient: Emodulus (T)=A * T degK + B			Poisson's ratio	Density	Thermal Expansion Coefficient	Thermal Conductivity	Specific Heat
	21 deg C	A	B	21 deg C	21 deg C	21 deg C	21 deg C	21 deg C
	Gpa			-	kg/m3	1/degK	W/(m-K)	J/(kg-K)
Inconel 718	209	-0.0605	226	0.284	8220	1.30E-05	11.4	473
Inconel X750	214	-0.0677	234	0.29	8303	1.21E-05	16.9	440
Stainless Steel 304	200	-0.0729	221	0.284	7900	1.73E-05	16.3	473
Stainless Steel 4140	210	-0.064924	234	0.3	8003	1.22E-05	42.7	473
Other								

README / TYPICAL notation / Shaft_Bearing_Models / Top Foil and Bump Models / FOIL_GAS_BEARING / RadialTemperature / Temperature field / CenterlineFILM

Worksheet: Shaft & Bearing models (II)

Graphical User Interface

XL GFB TH™ Spreadsheet for hydrodynamic foil GAS bearings

NOTE to USER : Run "Bump_stiffness", EnterBump type, and then run "FE_2D Analysis" to build the SUB-FOIL STRUCTURE MODEL of GAS FOIL BEARINGS. Enter input values in gray cells. Yellow cells show output

Box 8

BUMP GEOMETRY

2. Run Bump_Stiffness

Enter INPUT values		
Bump foil thickness, t_b	1.02E-04	meters
Bump pitch, s_0	4.57E-03	meters
Bump half length, l_0	2.03E-03	meters
Bump height, h_b	3.81E-04	meters
Bump foil elastic modulus, E_b	214.00	GPa
Bump foil Poisson's ratio, ν_b	0.29	
Friction coefficient, μ_f	0.0	

Bump type - STIFFNESS PER UNIT AREA (Iordanoff's formula, 1999)

1. Bump stiffness (free-free end bump), K_{ff}	3.17E+09	N/m3
2. Bump stiffness (welded-free end bump), K_{fp}	7.15E+09	N/m3
Structural loss factor, γ	0.2	

3. Run FE_2D Analysis

TOP FOIL MATERIAL PROPERTIES

Enter INPUT values		
Foil Arc Diameter (= Rotor outer diameter, D)	3.81E-02	meters
Foil Arc Length, L_t	350	deg
Top foil width (=Axial length), W_t	3.81E-02	meters
Number of Bumps, N_b	25	
Foil thickness, T_t	1.02E-04	meters
Material elastic modulus, E_t	214.00	GPa
Material Poisson's ratio, ν_t	0.29	
Stiffening factor (circumferential direction), S_{fc}	4	
No. of elements between bumps, N_{int}	3	
No of elements in the cir. direction, N_{ex}	75	
No of elements in the axial direction, N_{ey}	12	

Box 10

Box 9

Enter BUMP Type

BUMP STIFFNESS MATRIX

Bump No.	Bump type
1	1
2	1
3	1
4	1
5	2
6	1
7	1
8	1
9	1
10	2
11	1
12	1
13	1
14	1
15	2
16	1
17	1
18	1
19	1
20	2
21	1
22	1
23	1
24	1
25	2

BUMP GEOMETRY

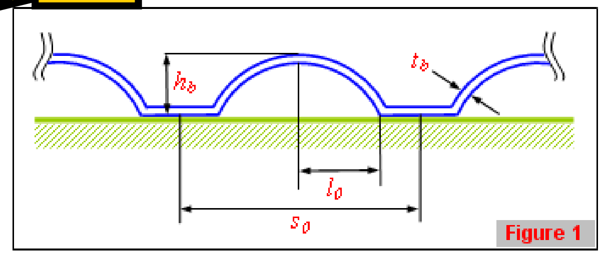


Figure 1

TOP FOIL MODEL

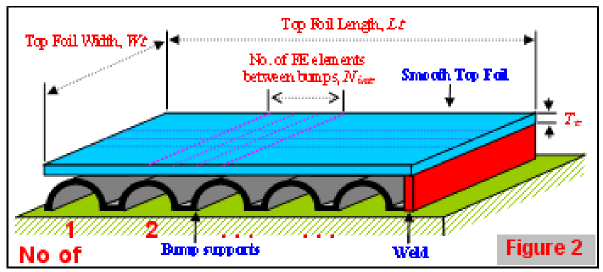
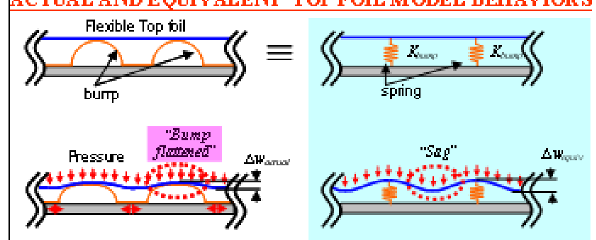


Figure 2

ACTUAL AND EQUIVALENT TOP FOIL MODEL BEHAVIORS



(a) Foil deflections in actual GFBs (b) Foil deflections for equiv. model

$\Delta W_{actual} \leq \Delta W_{equiv}$

Figure 3

README / TYPICAL notation / Shaft Bearing Models / Top Foil and Bump Models / FOIL GAS BEARING / RadialTemperature / Temperature field / Cent

Worksheet: Top Foil and Bump Models

Graphical User Interface

XL GFB TH ™ Spreadsheet for hydrodynamic foil GAS bearings

TEES project # 32525/39600/ME funded by NASA

Interface to Program:XLGFBHT

EDIT "DUMP.TXT" after program execution to VERIFY calculations & convergence.

Title: Foster-Miller Tech 2nd GEN GAS FOIL BEARING

Thermohydrodynamic MODEL

Bearing Geometry		
Rotor Outer Diameter	3.81E-02	meters
Axial Length	3.81E-02	meters
Radial Clearance	3.50E-05	meters
Number of shims	0	
TOP Foil - arc length	350.00	deg
TOP Foil - leading edge	45.00	deg
Shim thickness	0.00	meters
Offset (Ang location of shi	0.00	deg

Lubricant Fluid Properties		
Ambient Pressure	1.013	bar
Ambient Temperature	21.0	deg C
Properties at 21 deg C		
Viscosity at Tambient	1.93E-02	c-Poise
Density at Tambient	1.16	kg/m3
Specific Heat	1007.0	J/(kg-K)
Thermal Conductivity	0.02	W/(m-K)
Gas Constant	287	J/(kg-K)

CONVERGENCE PARAMETERS	
Max Iterations - film lan	500
error press-temp film la	1.00E-05

GRID RATIO (circ/Axi	0.98
No. Circ. Grid Points	76
No. Axial Grid Points	13

Frequency Analysis Option		
Constant Shaft Rpm	30000	rpm

Synchronous Analysis

Select - Analysis Type: Vary Load

Select - Option Foil Detach: DETACH

4. Run FOIL_GAS_BEAR

NOTE to USER : Enter input values in gray cells.
"Run FOIL_GFB_BEAR" predicts GAS FOIL BEARING static and dynamic force performance and temperature field.
Yellow cells show output values.

Enter INPUT values / FOIL Bearing force coefficients / IMPEDANCES (R: real, I: im

ex_input µm	ey_input µm	Load-X N	Load-Y N	Speed rpm	Kstructure N/m	Kxx [N/m]	Kxy N/m	Kyx N/m	Kyy N/m	Cxx [N-s/m]	Cxy N-s/m	Cyx N-s/m	Cyy N-s/m	R-XX N/m	R-XY	R-YX
		6.50	0.00	10000	2.37E+06	3.33E+06	6.12E+05	-1.18E+06	3.33E+06	3227.6	-1868.5	879.9	3676.8	3.33E+06	6.12E+05	-1.18E+06
		6.50	0.00	20000	2.37E+06	5.53E+06	-6.09E+05	-1.29E+06	5.39E+06	1619.4	-1110.6	621.5	1913.9	5.53E+06	-6.09E+05	-1.29E+06
		6.50	0.00	30000	2.37E+06	6.93E+06	-1.14E+06	-1.18E+06	6.72E+06	938.6	-661.0	457.3	1188.0	6.93E+06	-1.14E+06	-1.18E+06
		6.50	0.00	40000	2.37E+06	7.98E+06	-1.31E+06	-1.10E+06	7.76E+06	606.0	-439.6	356.4	815.2	7.98E+06	-1.31E+06	-1.10E+06

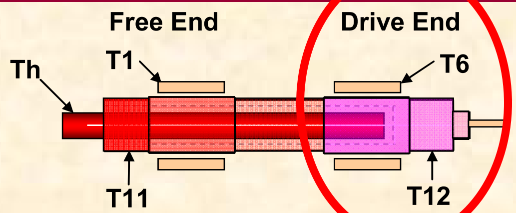
FOIL Bearing static load performance parameters

Speed [rpm]	e_structure µm	ex µm	ey µm	Eccentricity µm	eccentricity ratio	Angle deg	Minimum film [µm]	Peak film temperatur [degC]	Fx Reaction N	Fy Reaction N	Force N	Specific load bar	Max pressure [bar]	Torque N-m	Power Loss [kW]	Keq N/m
10000.00	38.41	1.38	3.00	3.30	0.09	65.33	15.48	124.3	-6.5	-0.1	6.5	0.04	1.11	1.68E-03	1.76E-03	2.93E+06
20000.00	38.41	0.16	1.64	1.65	0.05	84.26	16.76	125.1	-6.4	0.0	6.4	0.04	1.13	3.26E-03	6.82E-03	5.17E+06
30000.00	38.41	-0.10	1.14	1.14	0.03	-84.80	16.76	126.4	-6.5	0.1	6.5	0.04	1.18	4.81E-03	1.51E-02	6.71E+06
40000.00	38.41	-0.22	0.82	0.85	0.02	-74.68	16.32	128.4	-6.5	0.1	6.5	0.05	1.23	6.39E-03	2.68E-02	7.87E+06

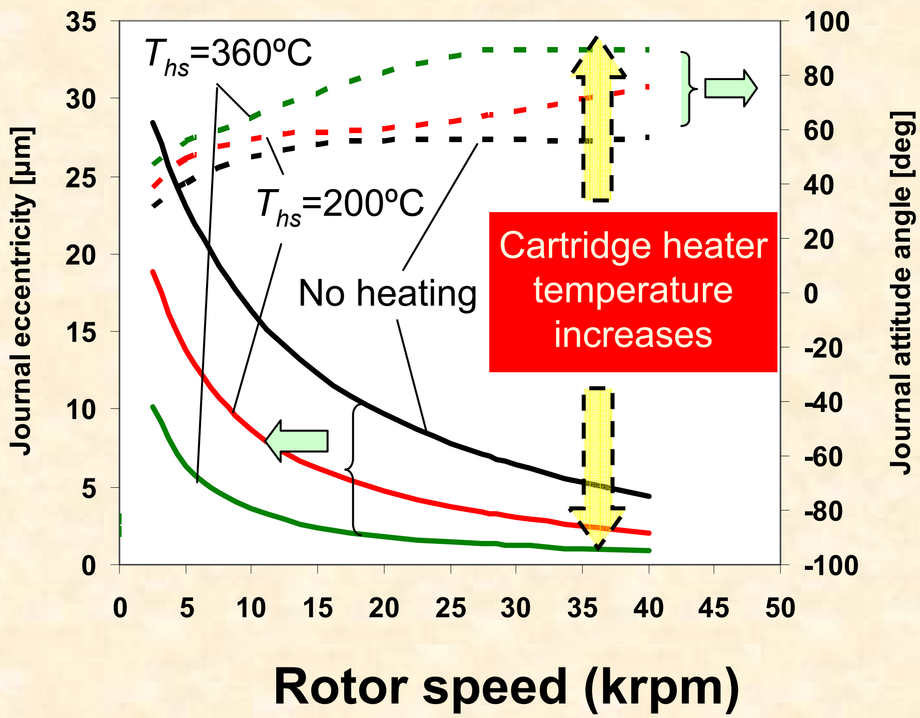
Worksheet: Foil Bearing (Operation and Results)

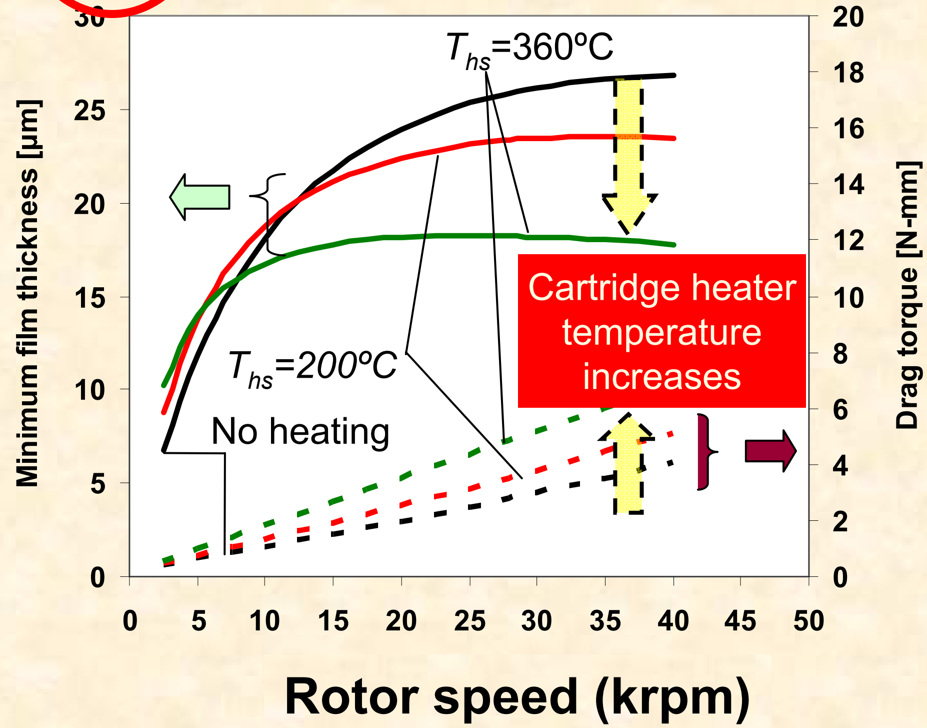
Static load parameters Predictions

Drive End FB



static load ~ 6.5 N
No cooling flow





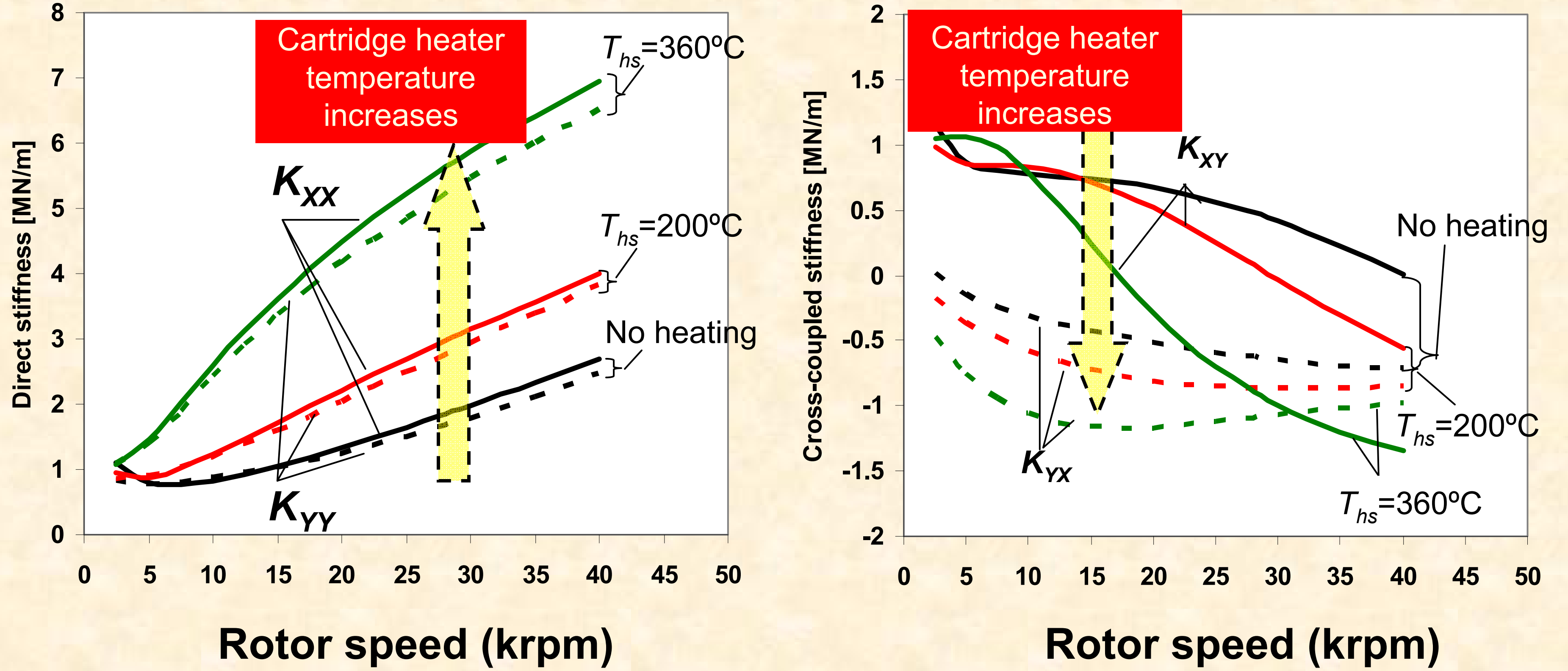
Predictions

As temperature increases, journal attitude angle and drag torque increase but journal eccentricity and minimum film thickness decrease due to reduction in operating clearance

Bearing stiffnesses Predictions

Drive End FB

static load ~ 6.5 N
No cooling flow



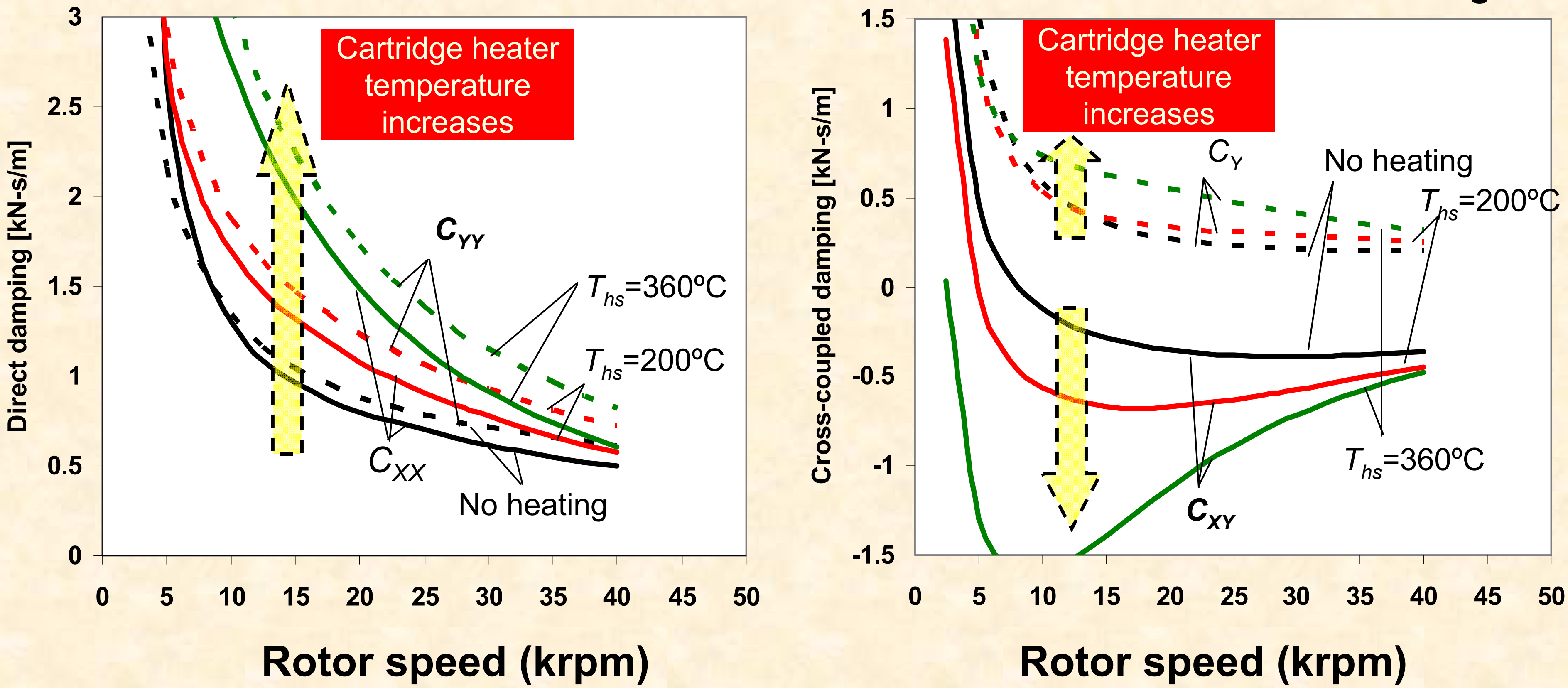
Predictions

As temperature increases, stiffnesses (K_{XX}, K_{YY}) increase significantly, while difference (K_{XY}-K_{YX}) increases slightly at low rotor speeds and decreases at high rotor speeds

Bearing damping Predictions

Drive End FB

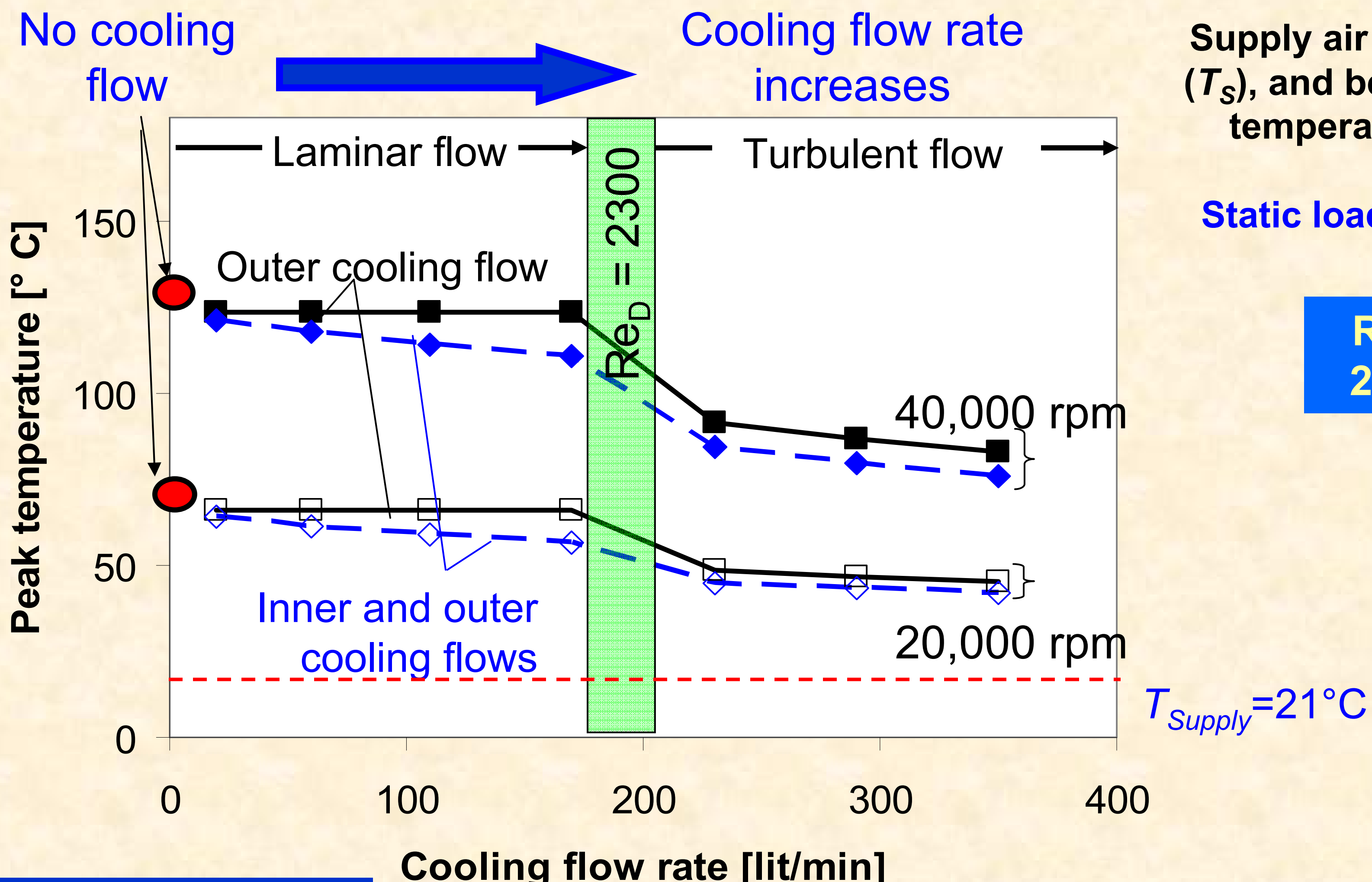
static load ~ 6.5 N
No cooling flow



Predictions

As temperature increases, damping (C_{XX}, C_{YY}) increase. Cross damping (C_{XY},C_{YX}) change little above 30 krpm.

Predictions on effect of cooling flow



Supply air (T_{Supply}), shaft (T_S), and bearing OD (T_B) temperatures at 21 °C.

Static load =89 N (180°).

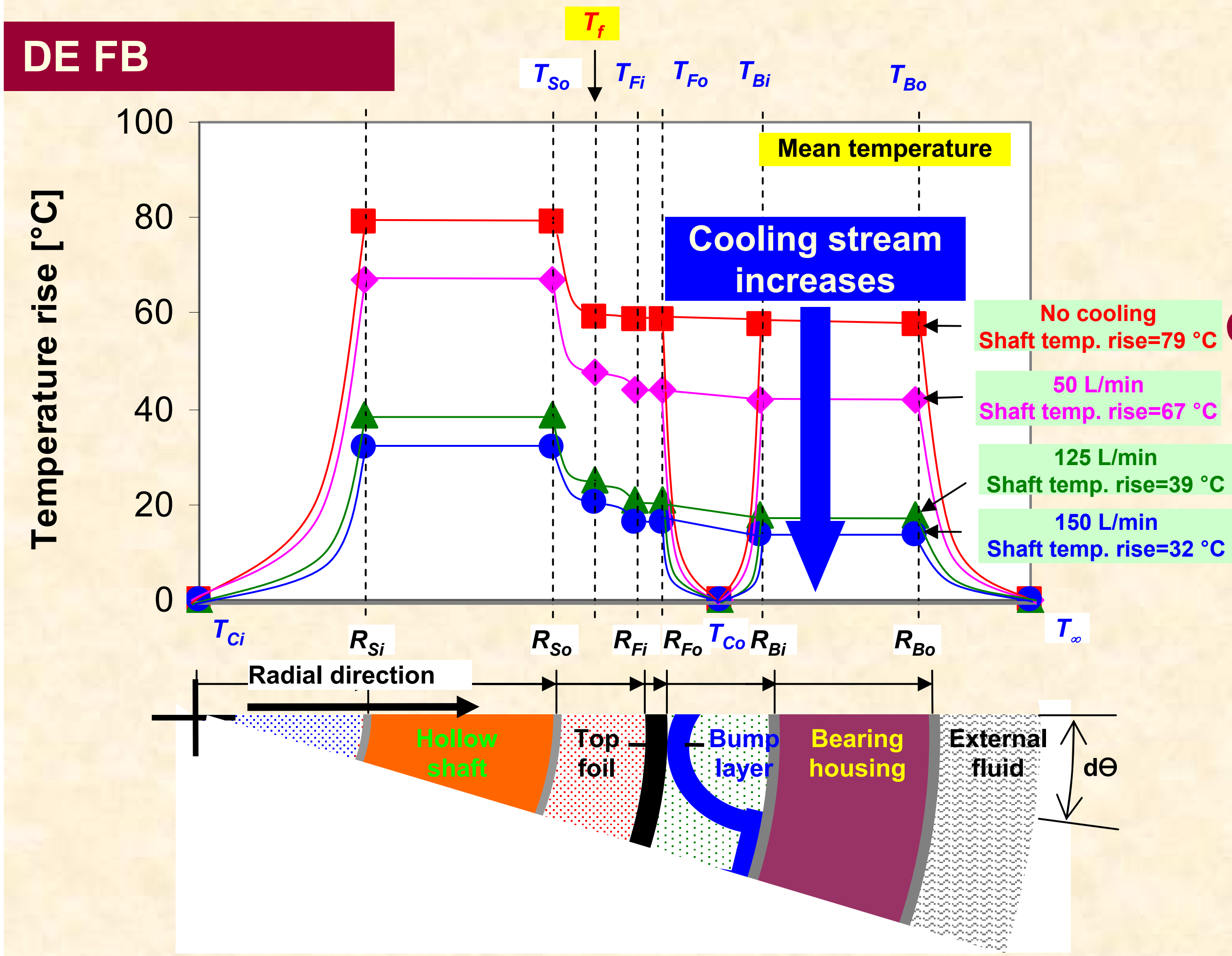
Rotor speed : 20 & 40 krpm

Predictions

Peak temperature drops with strength of cooling stream. Sudden drop at ~ 200 lit/min b/c of transition from laminar to turbulent flow

Predictions radial temperature Model

DE FB



Natural convection on exposed surfaces of bearing OD and shaft ID

w/o & w cooling flow

Rotor speed : 30 krpm

Model predictions

With forced cooling, GFB operates 50 °C cooler. Outer cooling stream is most effective in removing heat

Topic

- Statement of Work & Sources for Presentation
- Objectives and accomplished work in 07-08

  Computational model. Validation with published data.

  Rotordynamic measurements at TAMU
- Objectives and accomplished work in 2008-09

  Description of test rig and foil bearings at TAMU

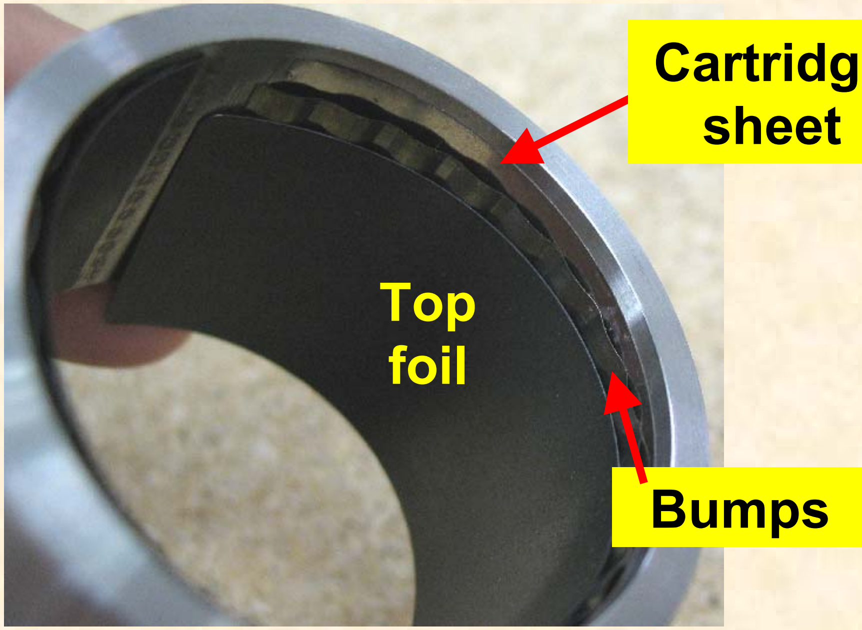
  Effect of temperature on bearing temperatures, coastdown speed and rotor motions

  Effect of cooling flow on bearing and shaft temperatures. Validation of computational model
- The computational code

  Graphical User Interface. Further predictions
- **Work with MiTi Bearings**
- Cl

San Andrés, L., Ryu, K., and Kim, T.H., 2011, "Identification of Structural Stiffness and Energy Dissipation parameters in a 2nd Generation Foil Bearing; Effect of Shaft Temperature", ASME J. Eng. Gas Turbines Power, vol. 133 (March) , pp. 032501

MiTi Korolon® foil bearing



FB nominal dimensions

Parameter [Dimension]	Symbol	Value
Cartridge inner diameter [mm]	D	37.98
Cartridge outer diameter [mm]	D_O	44.64
Axial bearing length [mm]	L	25.40
Number of bumps	N_B	24× 3
Bump pitch [mm]	s	4.318
Bump length [mm]	$2l_o$	3.302
Bump foil thickness [mm]	t	0.102
Bump height [mm]	h	0.394
Top foil thickness [mm]	t_T	0.127
Bump arc radius [mm]	r_B	5.08
Bearing Top foil inner diameter [mm]	D_T	38.135
Shaft diameter [mm]	D_J	36.84
Nominal radial clearance [mm]	C	0.160

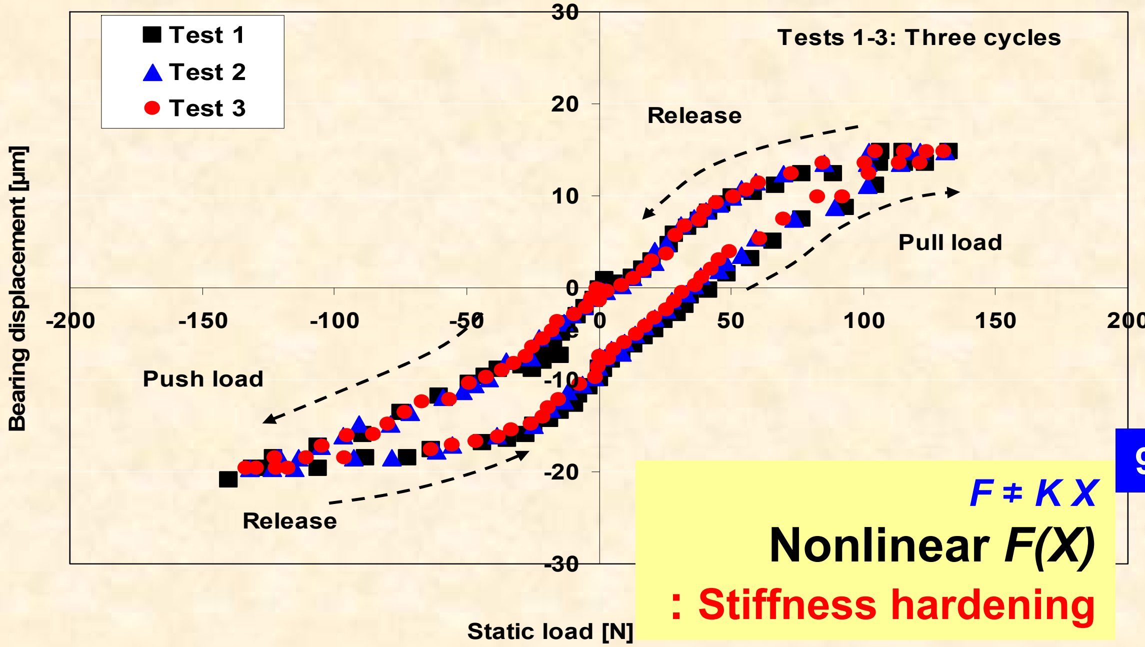
Two generation II GFBs

Three (axial) bump strip layers, each with 24 bumps.

Korolon® 800 coating (up to 800°F) on top foil surface.

MiTi® FB deflection versus static load

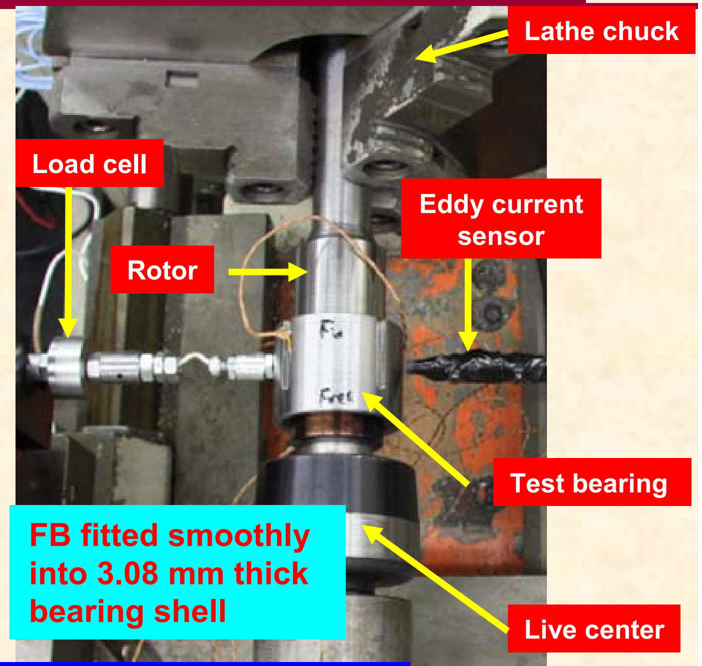
Room temperature tests



$F \neq K X$

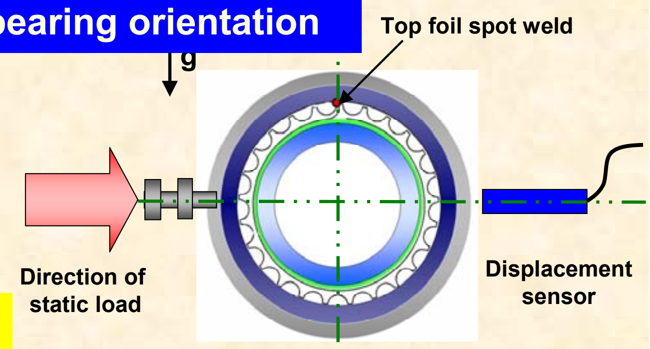
Nonlinear *F(X)*

: Stiffness hardening



FB fitted smoothly into 3.08 mm thick bearing shell

90° bearing orientation



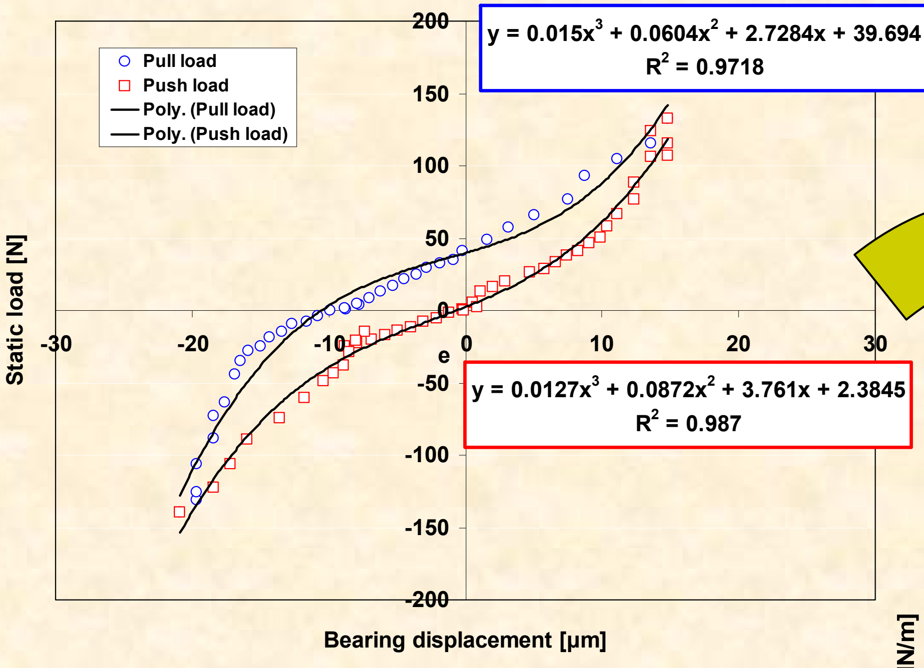
Shaft OD 36.56 mm: Highly preloaded FB

Large hysteresis loop : Mechanical energy dissipation

due to dry-friction between top foil contacting bumps and bump strip layers contacting bearing cartridge sheet

MiTi® FB structural stiffness

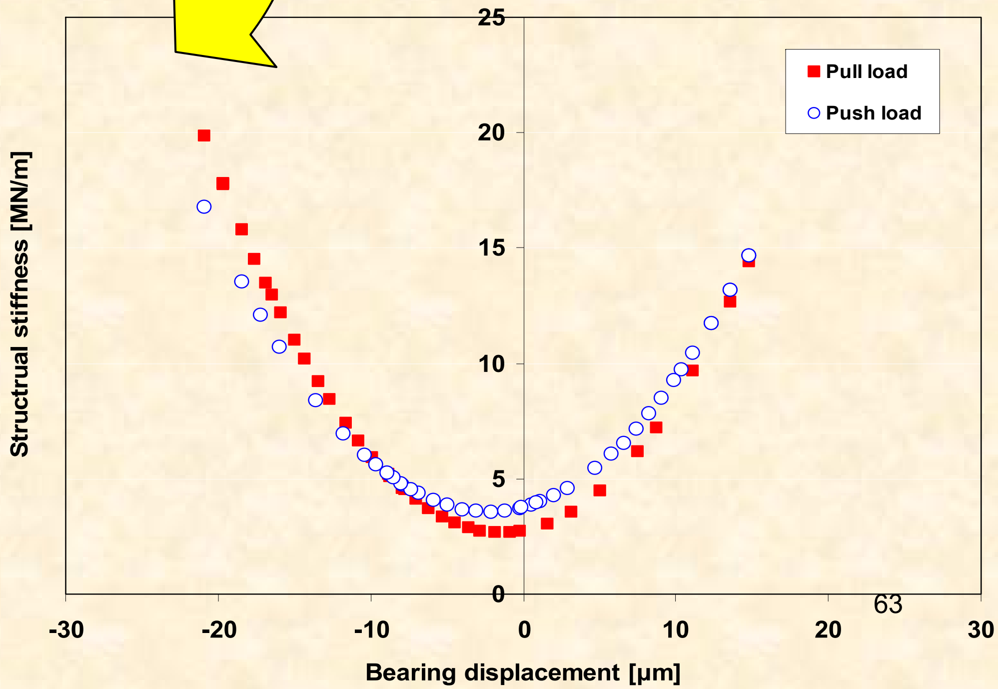
Room temperature tests



Cubic polynomial curve fit over span of applied loads

$$F=F_0+K_1X+K_2X^2+K_3X^3$$

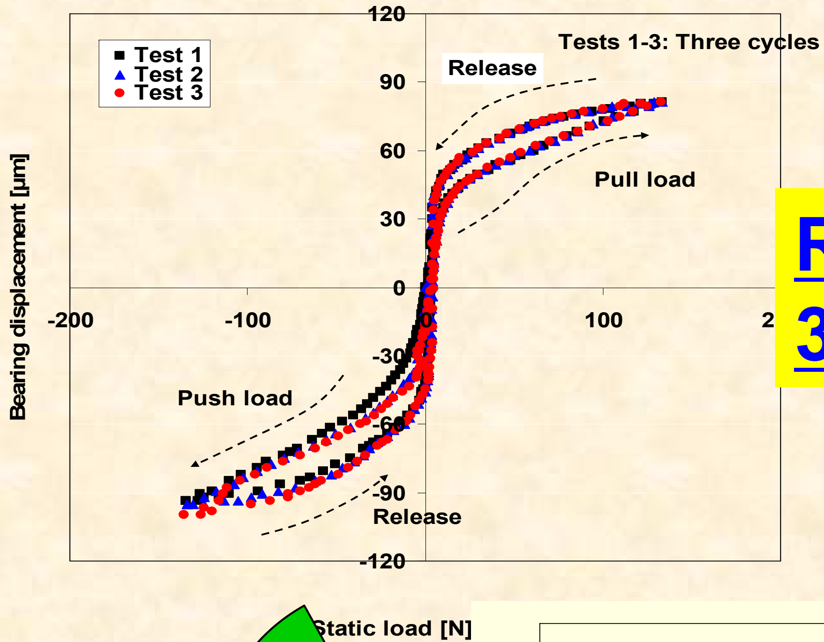
$$K=K_1+2K_2X+3K_3X^2$$



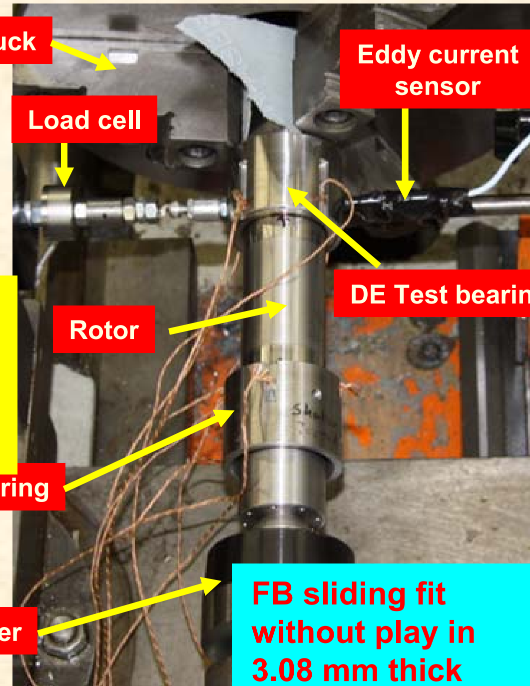
Distinctive hardening effect as FB deflection increases

MiTi® FB deflection versus static load

Room temperature tests

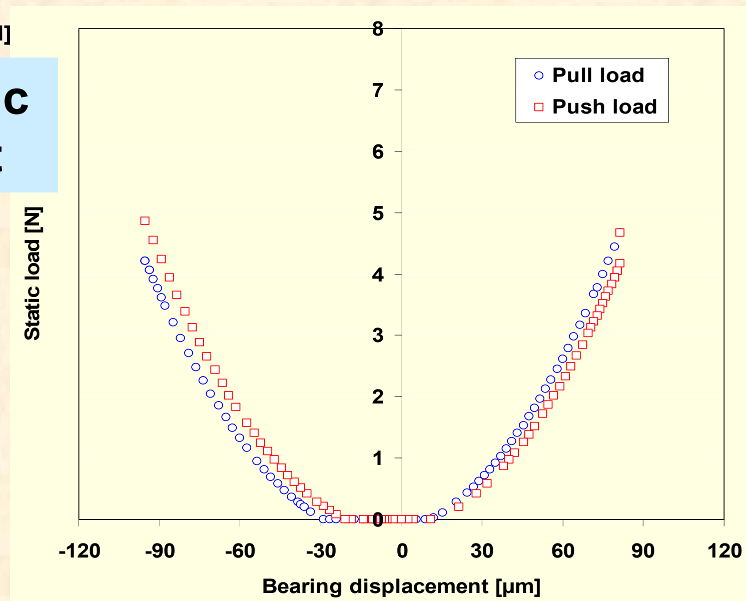


ROTOR OD 36.46 mm

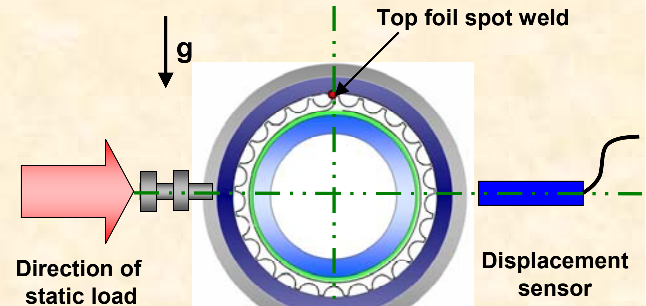


FB sliding fit without play in 3.08 mm thick bearing shell!

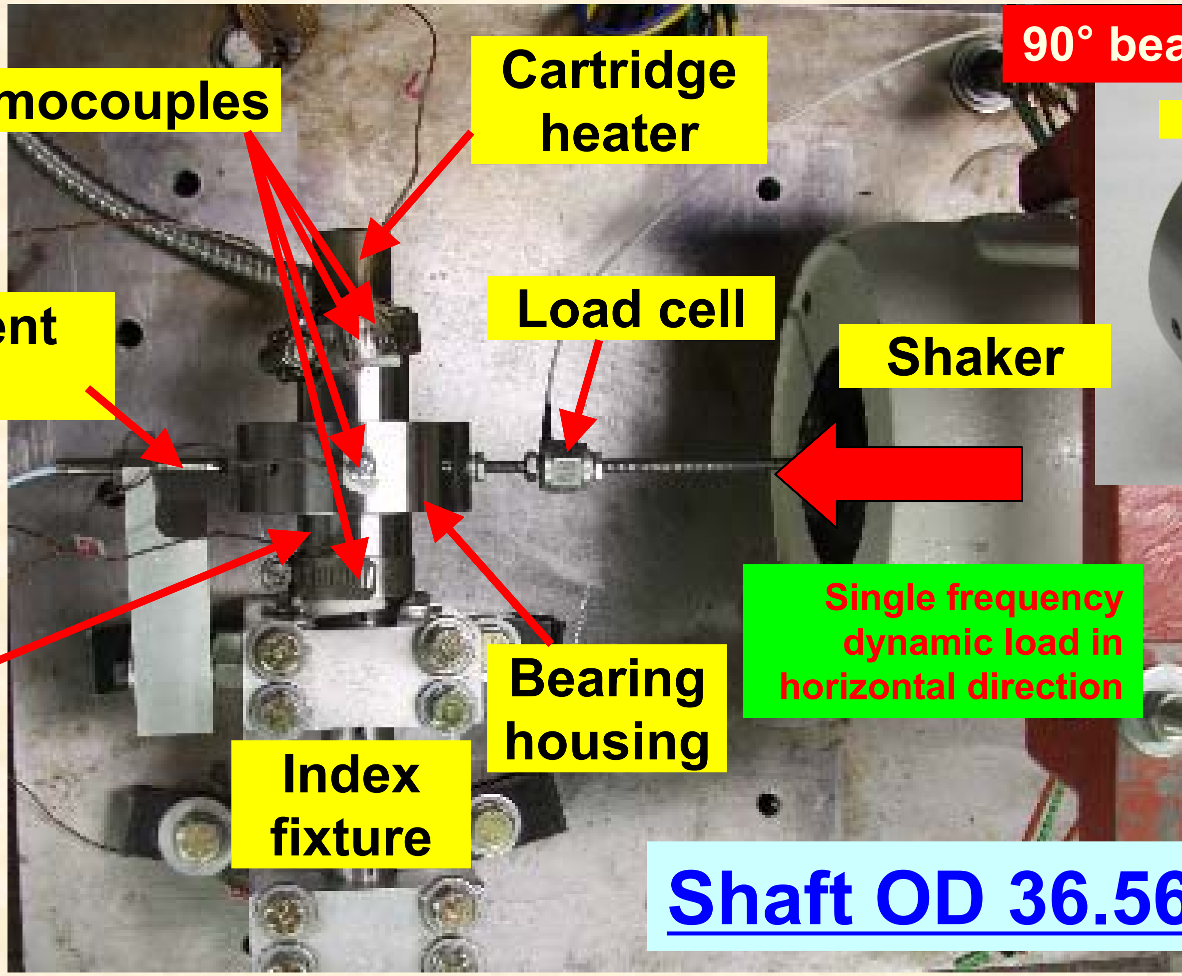
Identified from Cubic polynomial curve fit



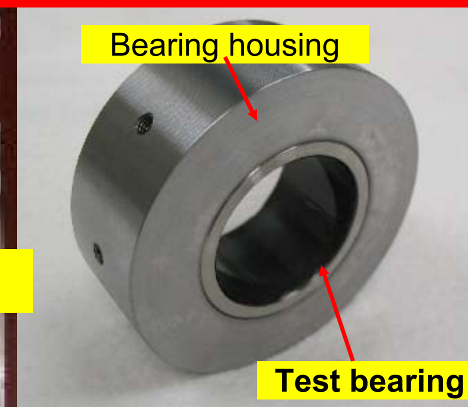
90° bearing orientation



MiTi® FB test setup for dynamic loads



90° bearing orientation



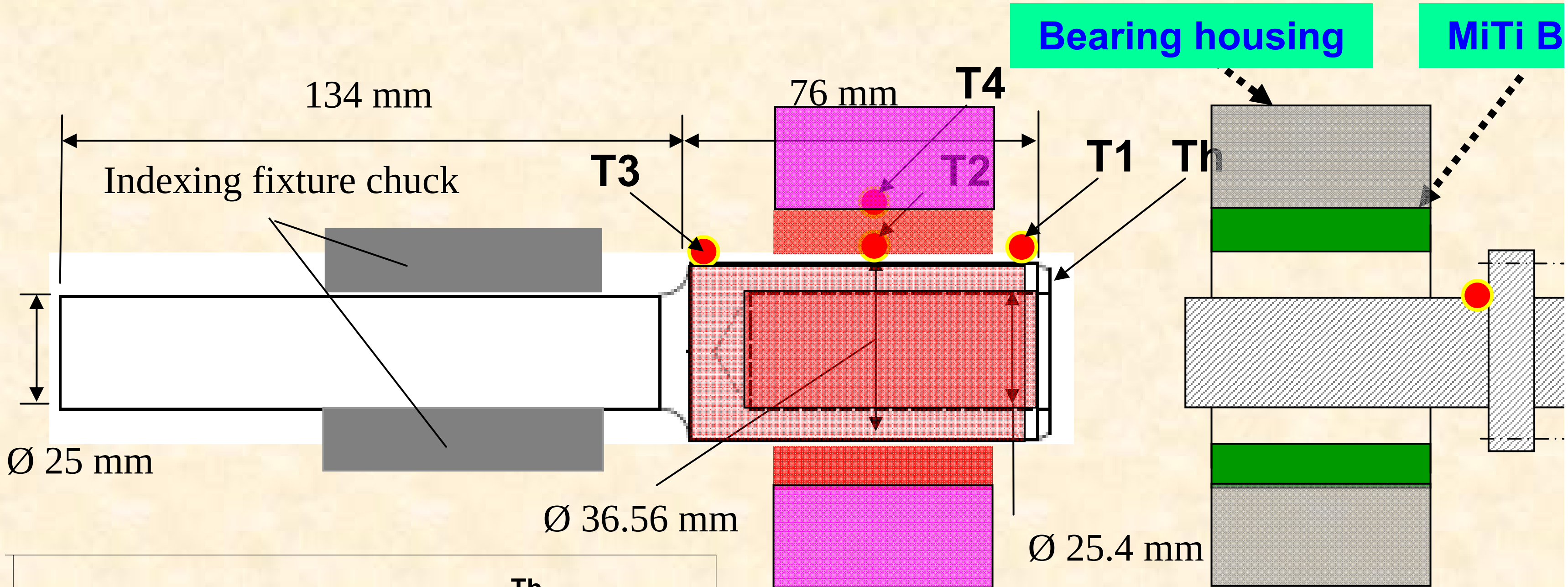
FB press fitted onto 15.5 mm thick bearing housing!

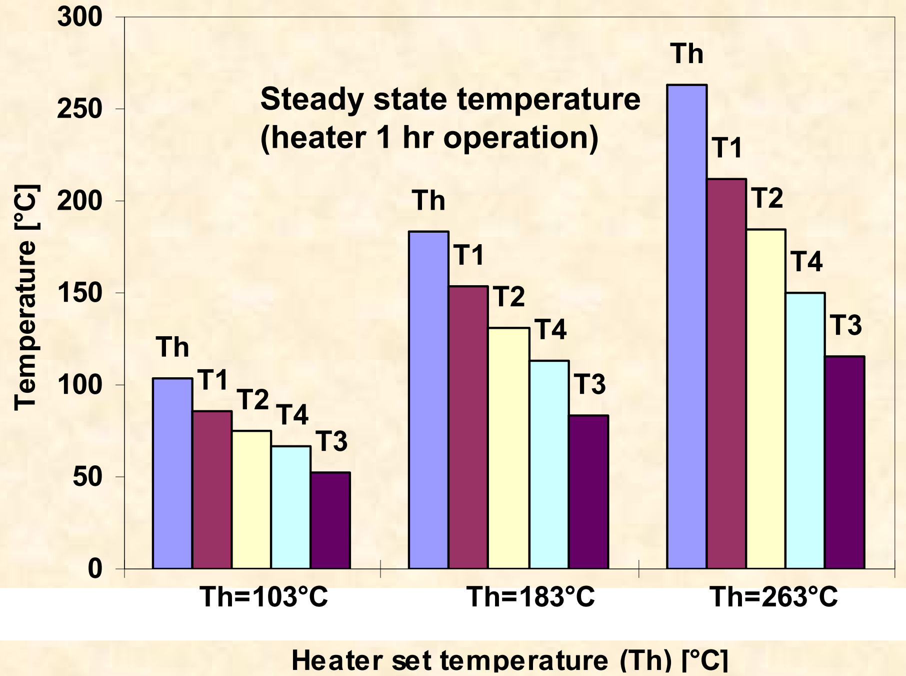
Shaft OD 36.56 mm

FB Displacement controlled [μm]	7.4, 11.1, 14.8, and 18.5
Frequency Range, Hz	50-200 (increment: 25 Hz)
Shaft Temperature, °C	23, 103, 183, and 263
Bearing Mass *M*, kg	0.785 (load cell + attachment hardware)

Uncoated rigid, non-rotating, hollow shaft supported on FB.

Shaft heating using electric heater





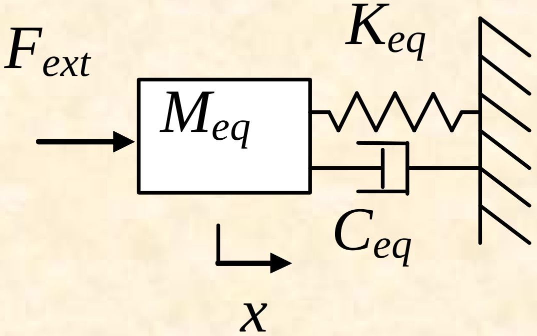
Significant temperature gradient along shaft axis.

Cartridge heater warms unevenly shaft

Parameter Identification (no shaft rotation)

Equivalent Test System: 1DOF

K **stiffness,** ***C**$_{eq}$* **viscous damping OR** γ **loss factor**



$$M\,\ddot{x} + K\,x + C_{eq}\,\dot{x} = F_{(t)}$$

$$F(t) = F_O e^{i\omega t} \qquad x(t) = \bar{X}\,e^{i\omega t}$$

Harmonic force & displacements

$$Z = \frac{F_O}{\bar{X}} = (K - \omega^2 M) + i\,\omega C_{eq}$$

Impedance Function

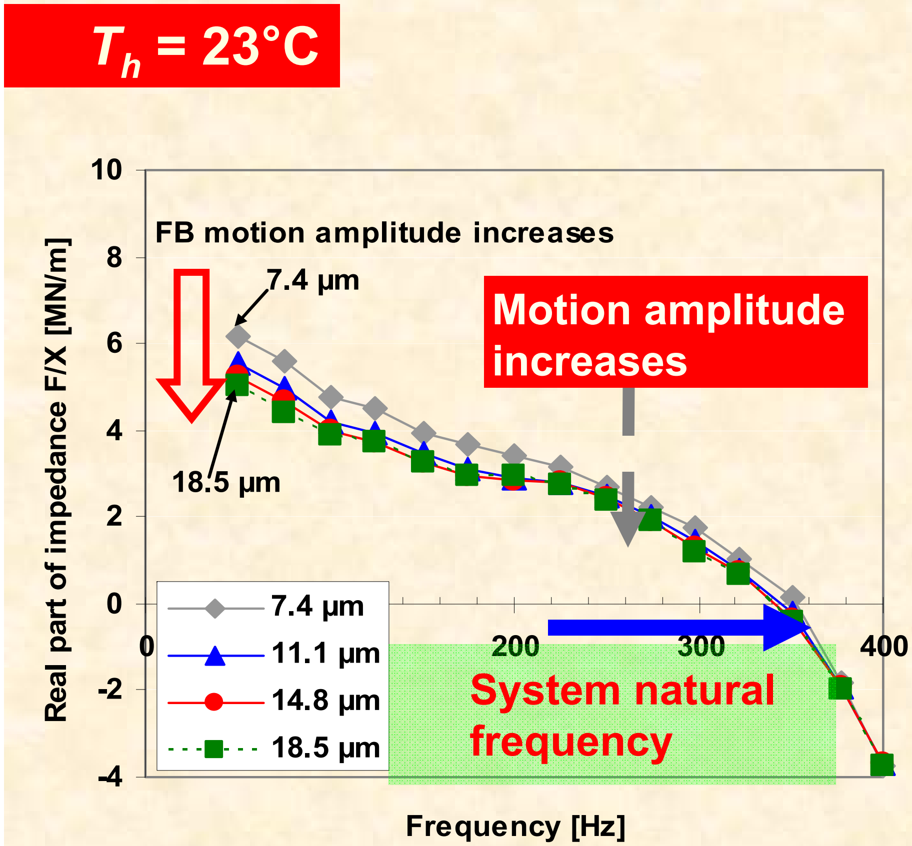
$$E_{dis} = \pi\omega C_{eq}\left|\bar{X}\right|^2$$

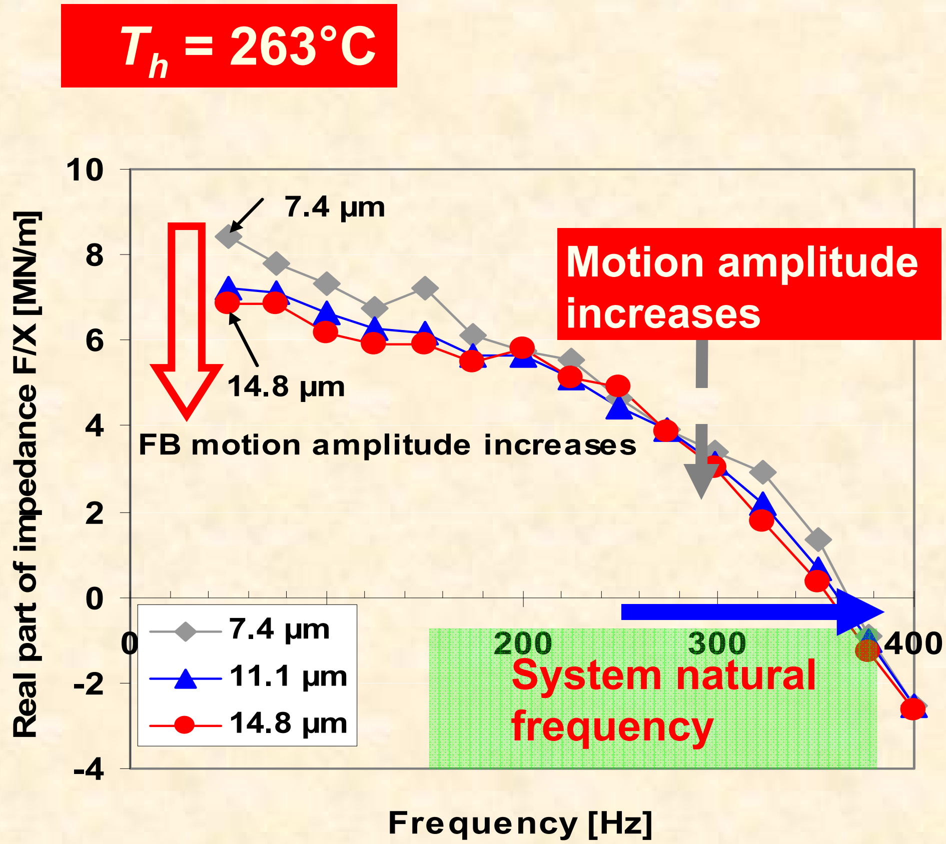
$$E_{dis} = \pi\,\gamma K\left|\bar{X}\right|^2$$

Viscous Dissipation or dry-friction Energy

Effect of temperature on dynamic stiffness

$$\mathrm{Re}\left(\frac{F_O}{\bar{X}}\right) = (K - \omega^2 M)$$



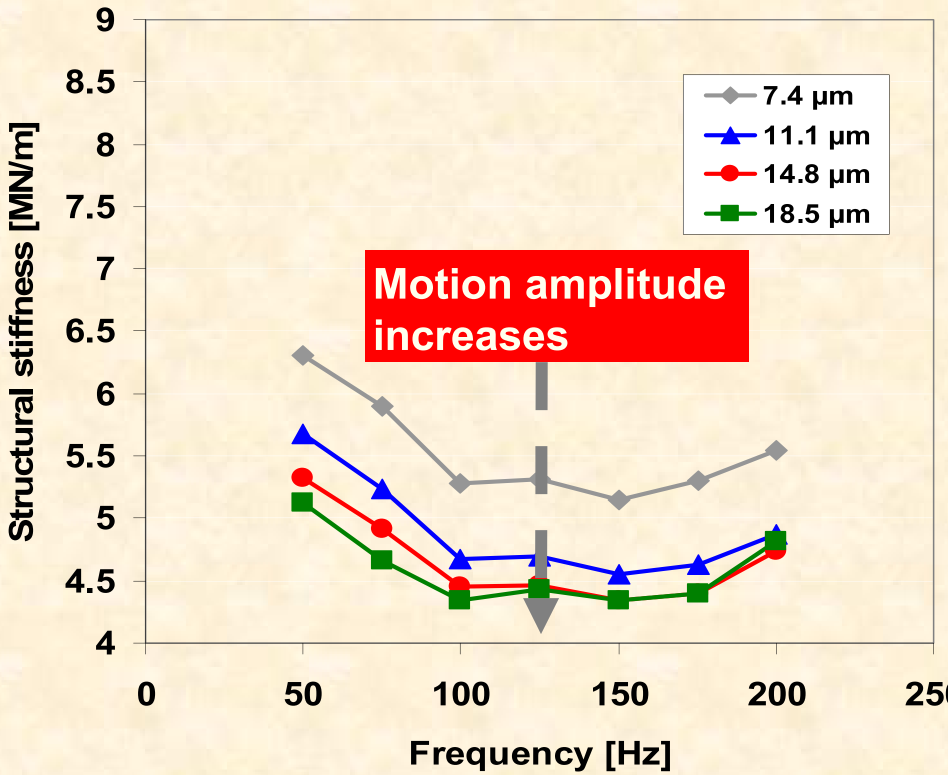


Real (*F/X*) decreases with FB motion amplitude & increases with shaft temperature.

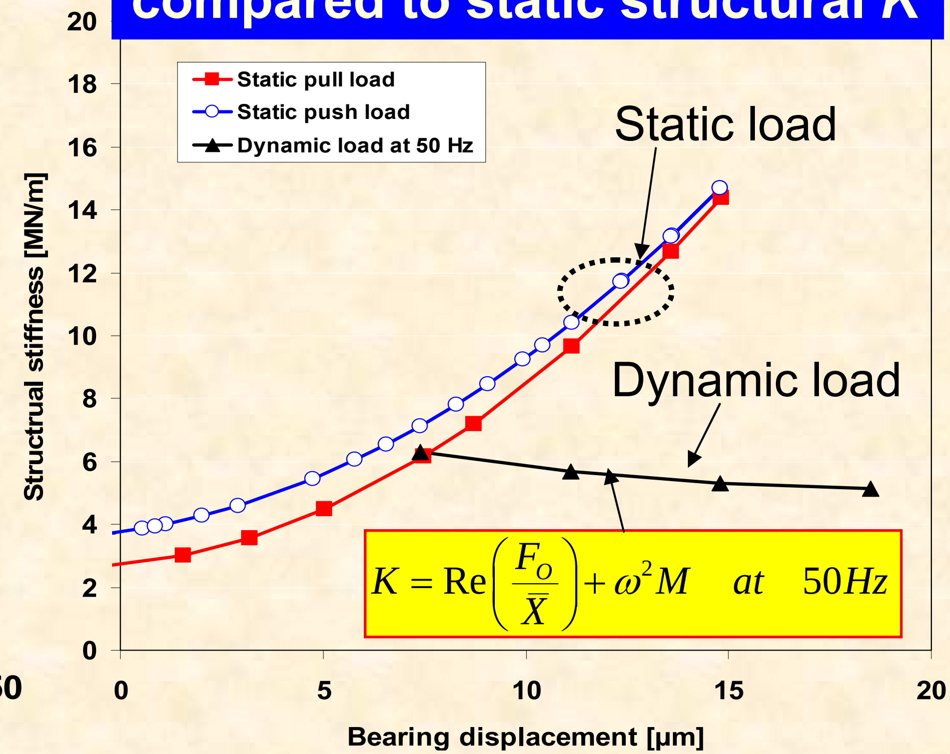
Effect of temperature on structural stiffness

$$K = \mathrm{Re}\left(\frac{F_O}{\bar{X}}\right) + \omega^2 M$$

T_h = 23°C, 90° bearing orientation



Dynamic structural *K* compared to static structural *K*



At larger FB deflections, static K is larger than dynamic K

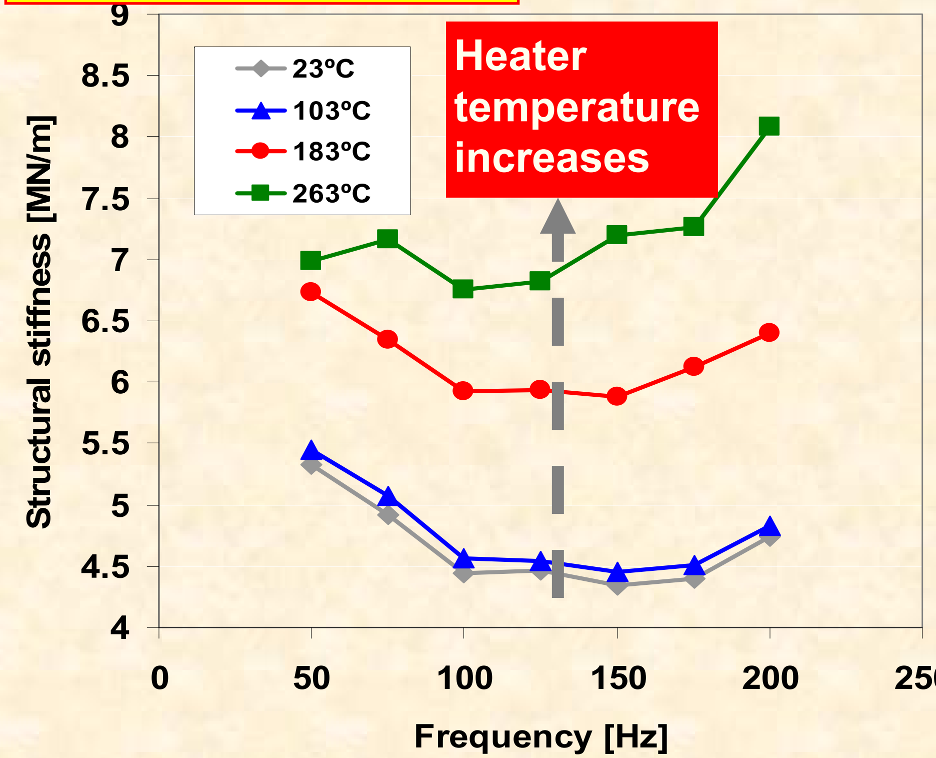
Highly preloaded FB: *K* decreases as FB motion amplitude increases due to decrease in # of active bumps

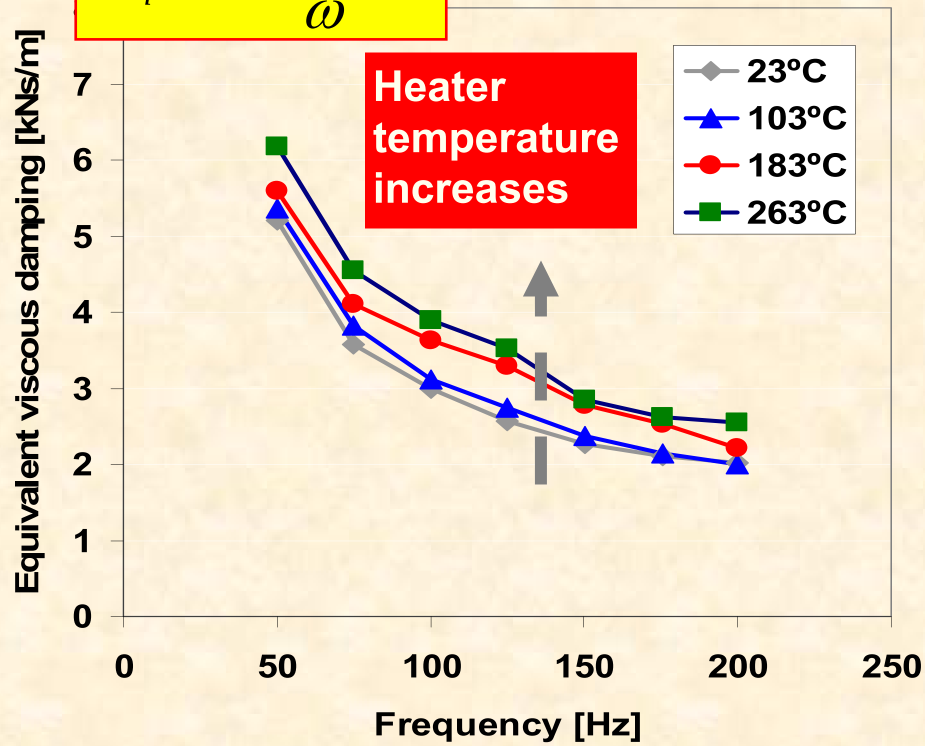
Effect of temperature on stiffness & damping

$$K = \mathrm{Re}\left(\frac{F_O}{\bar{X}}\right) + \omega^2 M$$

$$C_{eq} = \frac{\mathrm{Im}\left(\frac{F_O}{\bar{X}}\right)}{\omega}$$

FB motion amplitude: 14.8 μm





FB stiffness and viscous damping increase with shaft temperature and decrease with excitation frequency.

TEST FB cartridge OD is constrained within bearing housing.

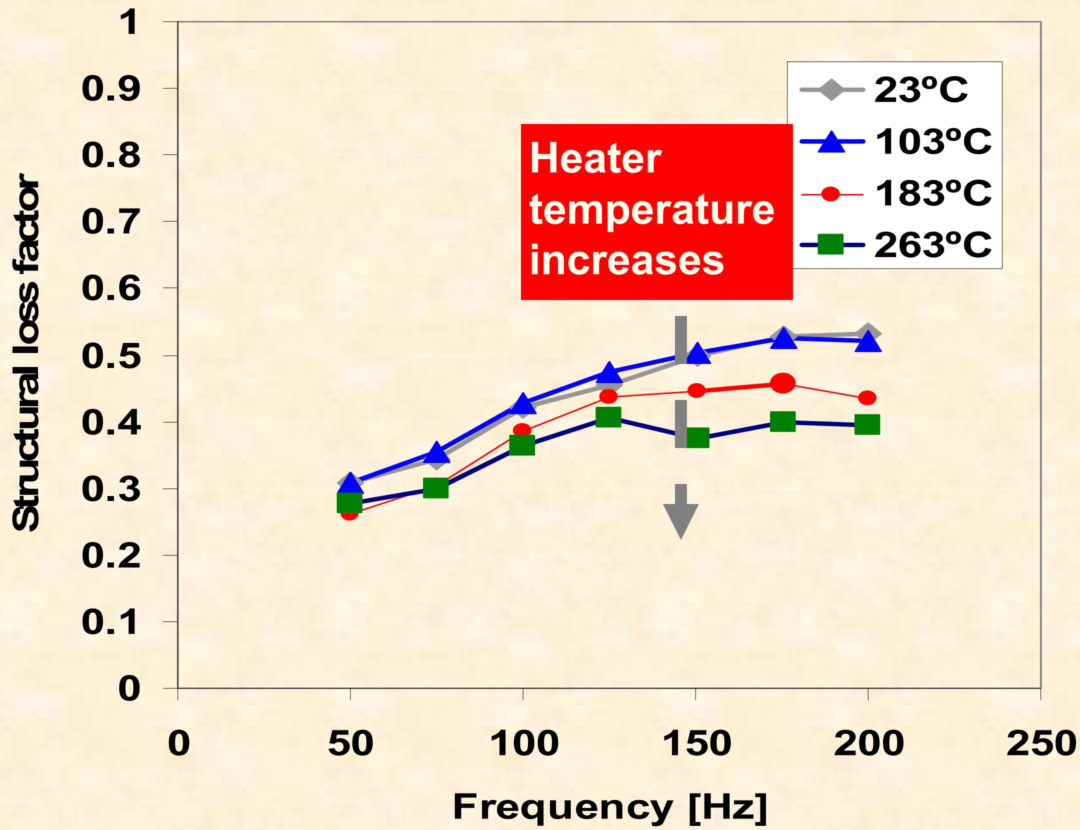
FB radial clearance decreases as shaft temperature raises!

Effect of temperature on loss factor γ

Structural (material) loss factor best represents energy dissipation in FB

FB motion amplitude: 14.8 μm

$$\gamma = \frac{C_{eq}\omega}{K}$$



The FB loss factor increases with excitation frequency and decreases slightly with shaft temperature. More damping expected in rotordynamic measurements

Texas A&M University
Mechanical Engineering Department

Nov. 23, 2010

EFFECT OF COOLING FLOW ON THE OPERATION OF A HOT ROTOR-GAS FOIL BEARING SYSTEM

Ph. D. Final Exam

Keun Ryu

Chair of Advisory Committee: Dr. Luis San Andrés

This material is based upon work supported by NASA GRC and the TAMU Turbomachinery Research Consortium (TRC)

Objective

Quantify effect of cooling flow and shaft temperature on the rotordynamic performance of a GFB supported rotor.

Investigate adequate thermal management strategies using forced cooling flow into the GFBs

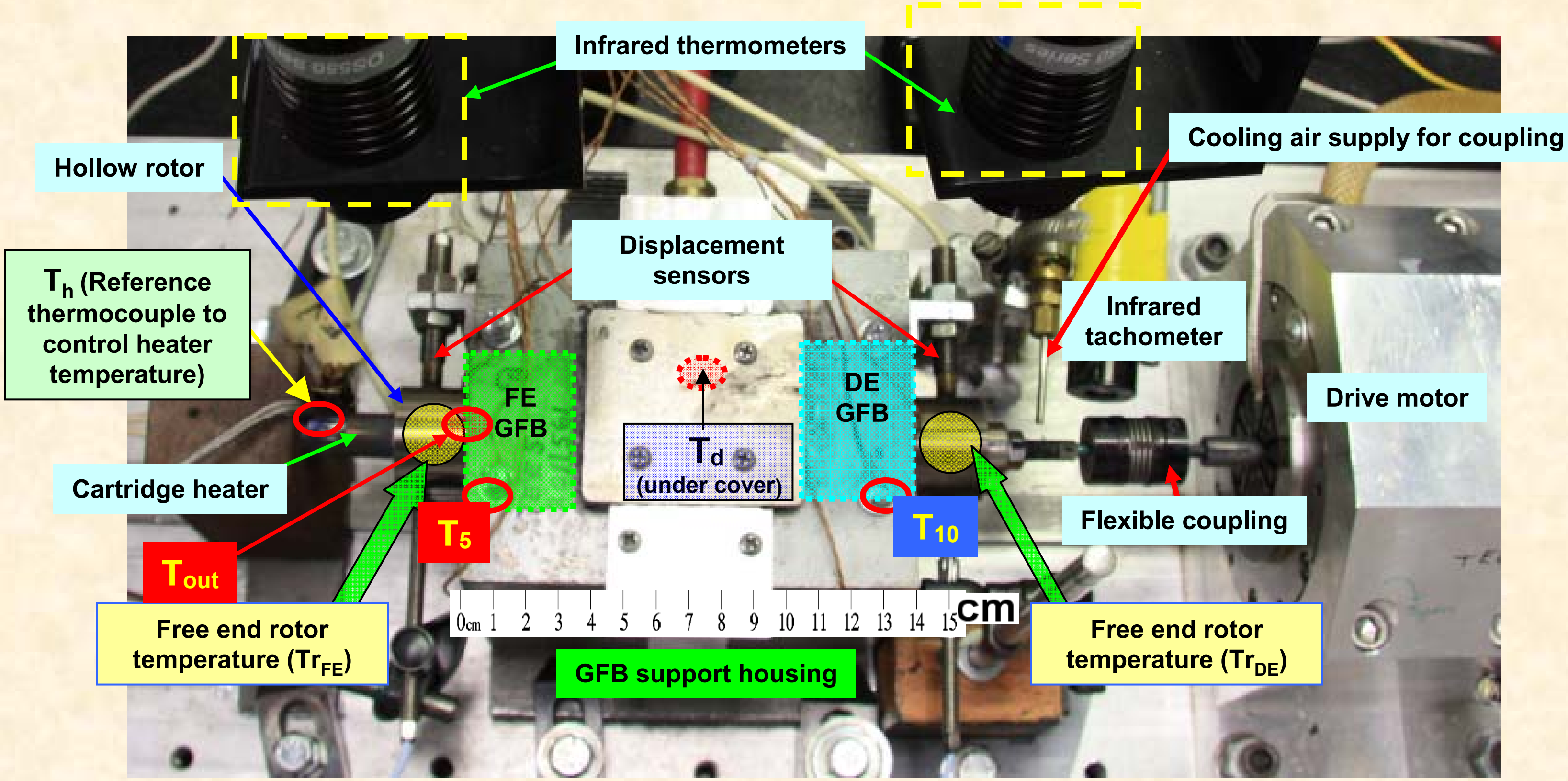
Research tasks

- **Revamp a GFB rotordynamic test rig for operation at high speed and extreme temperatures**
- **Measure temperature of bearings and rotor and the motions of rotor for increasing rotor speeds, shaft temperatures, and cooling flow rates**
- **Quantify effect of gas flow on cooling bearings (max. 500 L/min)**
- **Compare the experimental results (rotor responses and bearing temperatures) to predictions from an in-house computational program**

TAMU Hot rotor-GFB test rig

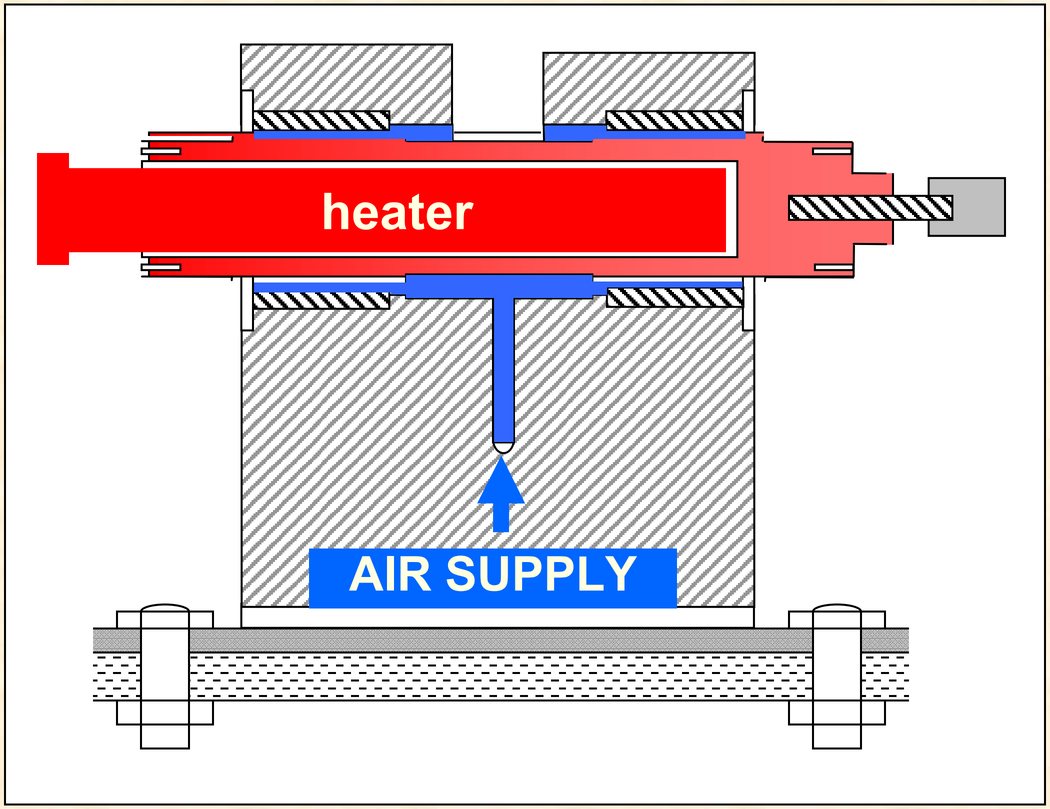
Instrumentation for high temperature

Gas flow meter (Max. 500 LPM). Drive motor (max. 50 krpm)



Cooling gas flow into GFBs

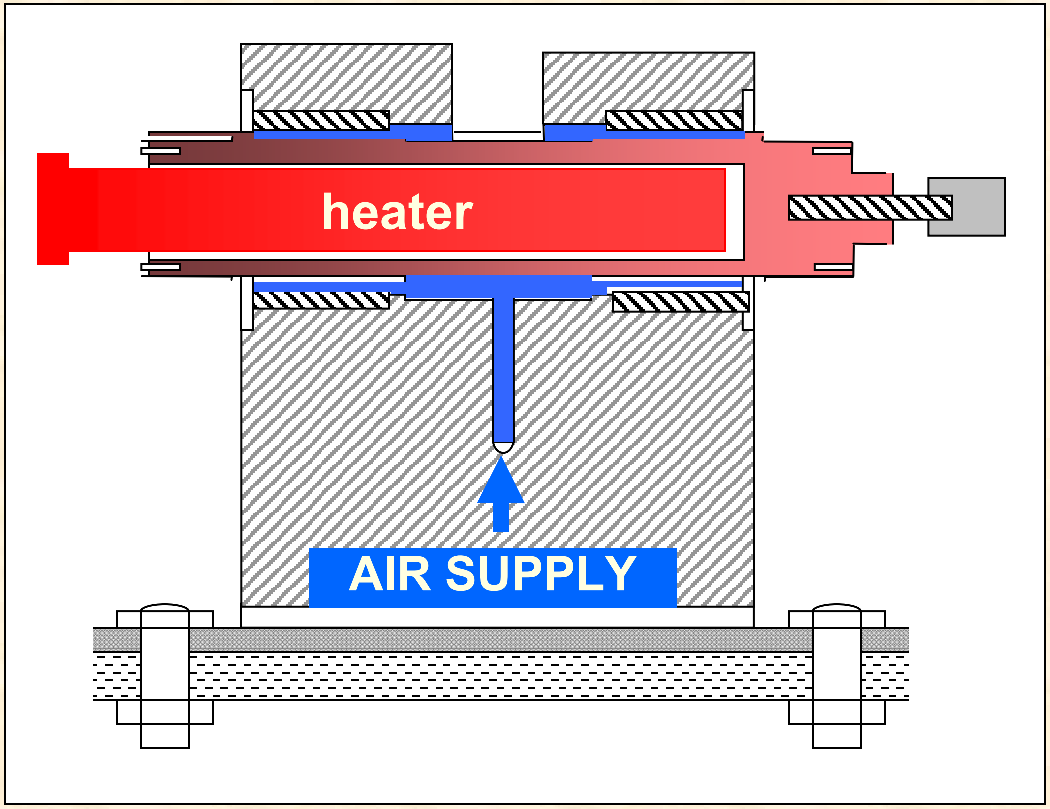
Gas pressure Max. 100 psi



Cooling flow needed for thermal management: to remove heat from shear drag or to reduce thermal gradients from hot to cold engine sections

Cooling gas flow into GFBs

Gas pressure Max. 100 psi



Heater warms unevenly test rotor.

Side cooling flows cool unevenly the rotor.

Overview – Thermal management

Component-level tests

- **Ruscitto et al (1978):** Perform load capacity tests on **1st gen. GFB** up to 45 kprm (1.7 MDN) and static load 111 N with **110 L/min cooling flow at 315C** bearing temperature.

- **DellaCorte (1998):** **No cooling flow**. **3rd gen. GFB** up to 70 krpm (2.4 MDN) at 700C. Bearing load capacity and torque decrease with temperature because of reduced bearing preload.

- **Dykas (2006)**: Investigates thermal management in **foil thrust bearings**. Cooling flow rates, to **450 L/min**, increase bearing load capacity at high rotor speeds. Inadequate thermal management can give thermo-elastic distortions affecting load capacity of test FB

- **Radil et al (2007):** Evaluate effectiveness of **three cooling methods (axial cooling, direct and indirect shaft cooling**) for thermal management in a hot GFB environment

Overview – Thermal management

System-level tests

- **LaRue et al (2006):** Oil-free Turbocharger. Thermal management achieved by **cooling the TC rotor and FBs**.

- **Lubell et al (2006):** Commercial oil-free micro-turbines. Cooling air flows axially through **hollow rotor ID** remarkably decrease rotor temperature.

- **Heshmat et al (2005):** Demonstrates hot (650C) GFB operation in a turbojet engine to 60 krpm. Cooling flow rates to 570 L/min still give **large axial thermal gradients (13ºC/cm)**

- **San Andrés et al (2009)**: Forced cooling flow has limited effectiveness at low rotor temperatures. At high test temperatures, **large cooling flows (turbulent) remove heat more efficiently**.

Gases have limited thermal capacity,
hence (some) bearings demand large cooling flows to remove heat from hot rotor sections.

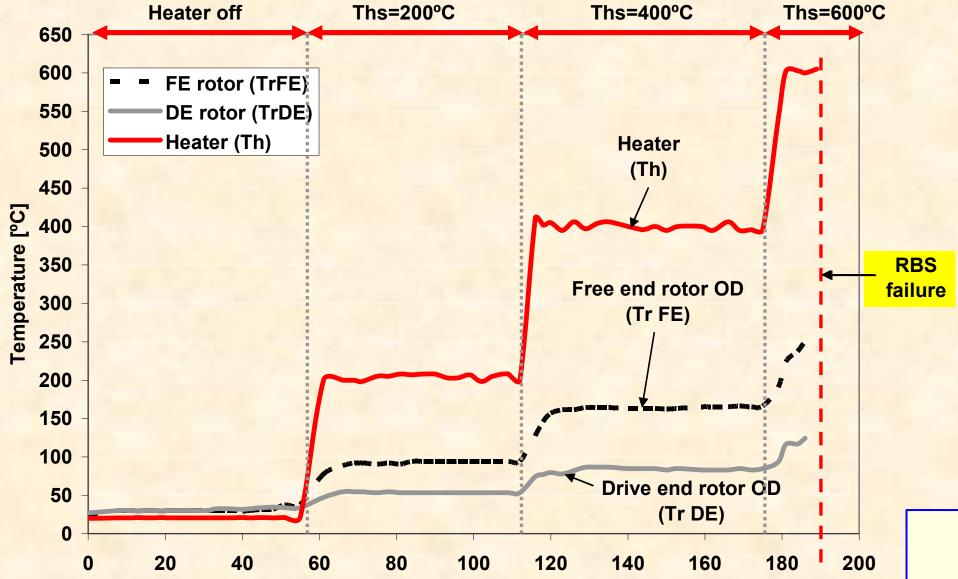
Why thermal effects are important?

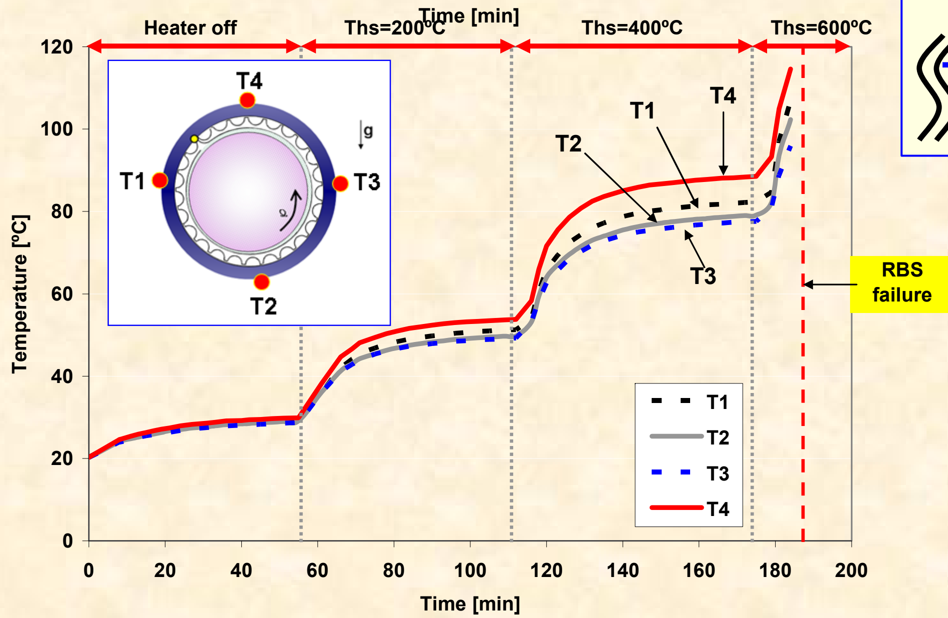
Gas bearings (when airborne) are nearly friction free, hence the show small (drag) power loss and temperature raise. With **hot rotors** the "lubricant" in the bearings must also **cool** components. But gases have small thermal capacity and conductivity, and hence, get **hot!** Rises in temperature change material properties (solids and gas), and most importantly, **change bearing clearance**!

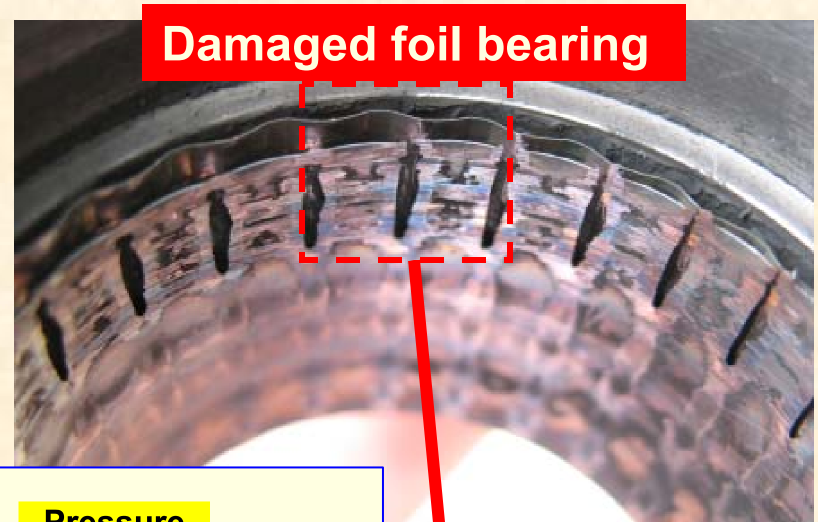
Lesson from previous demonstration

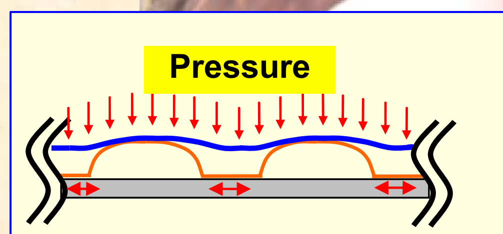
12/2009: HT GFB test

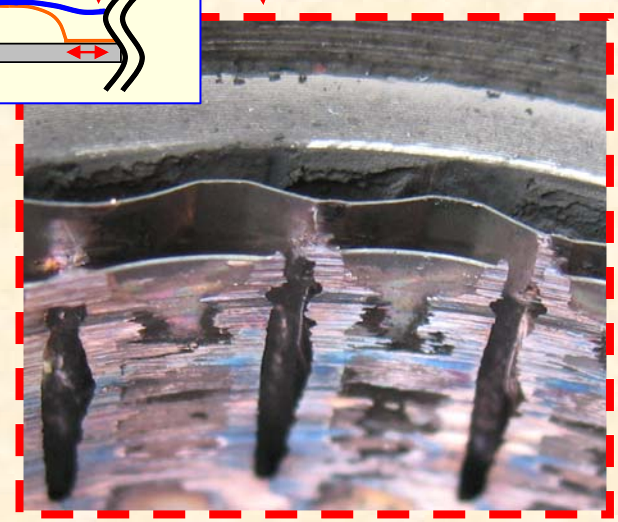
NO COOLING FLOW!!





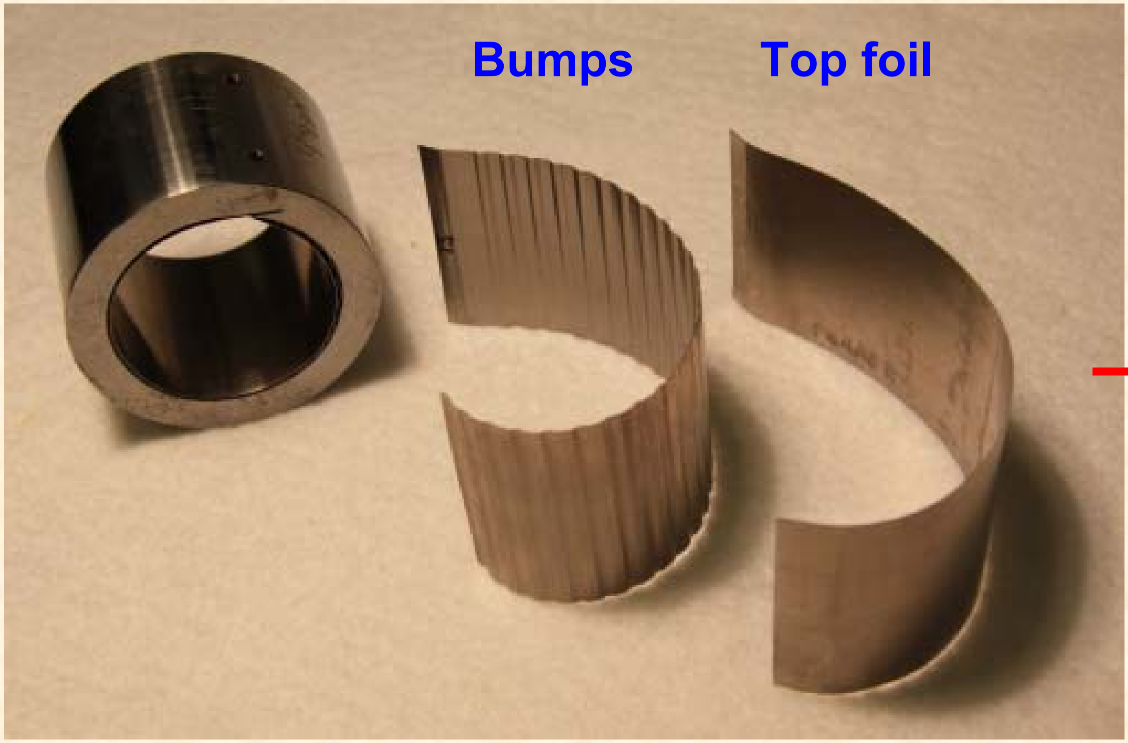


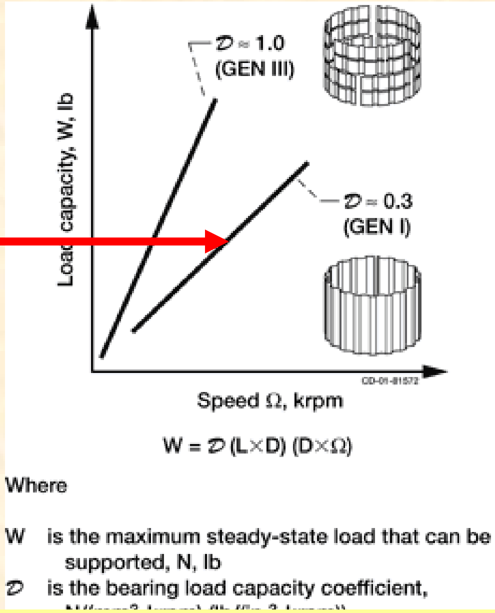




Test Gas Foil Bearing

2010: GFBs donated by KIST





Reference: **DellaCorte (2000)**
Rule of Thumb

Test Gas Foil Bearing (Bump-Type)

1st Generation. Diameter: 36.63 mm

Foil material: Inconel X-750

UNCOATED TOP Foil !

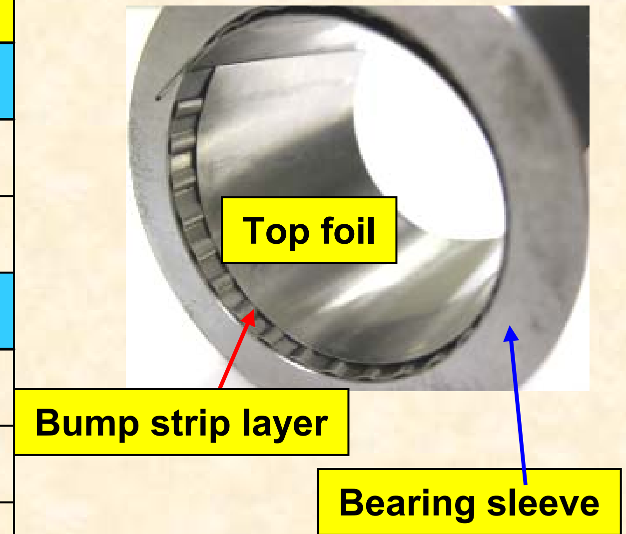
Hollow rotor (Inconel 718): 1.360 kg. Length: 200.66 mm. OD 36.51 mm and ID 17.9 mm. HT Coating up to 400C

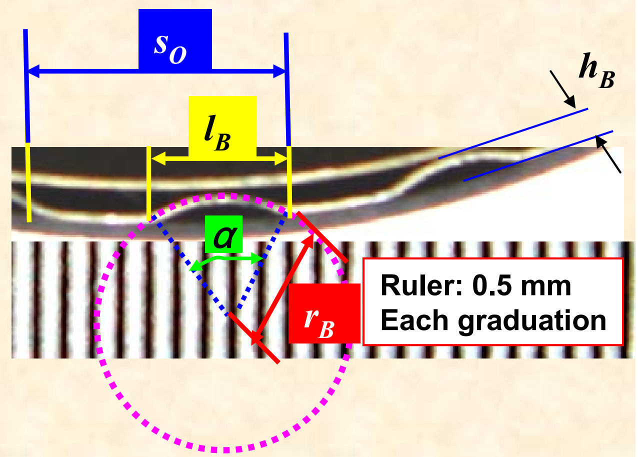
Foil Bearing Dimension

KIST FB uncoated (Gen. I)

	Parameter [mm]
Bearing cartridge	
Outer diameter	50.8
Inner diameter	37.95
Top foil and bump strip layer	
Top foil axial length	38.1
Top foil thickness	0.12
Bump foil thickness	0.12
Number of Bumps	26 × 1 axial
Bump pitch, s_o	4.4
Bump length, l_B	2.5
Bump height, h_B	0.50
Bump arc radius, r_B	2.25
Bump arc angle [deg]	67

Foil material: Inconel X-750

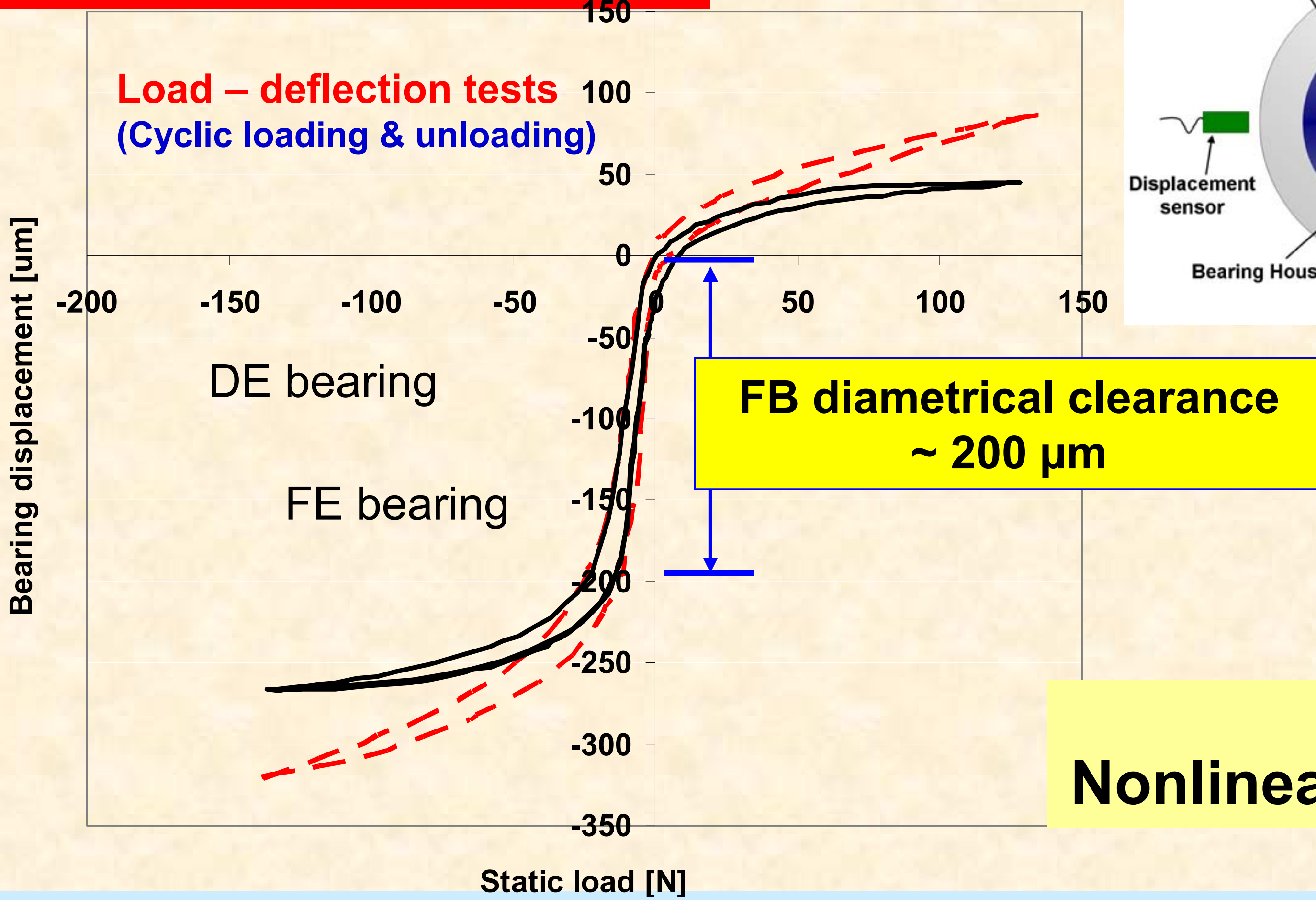


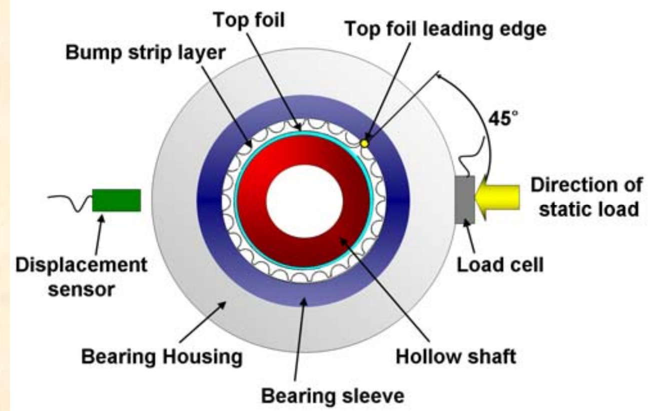


FB deflection versus static load

Room temperature tests: Estimation of bearing clearance

45° bearing orientation





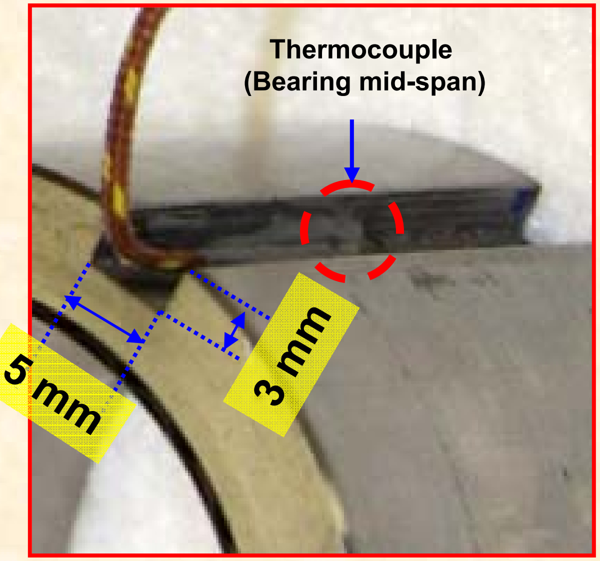
$F \neq K X$

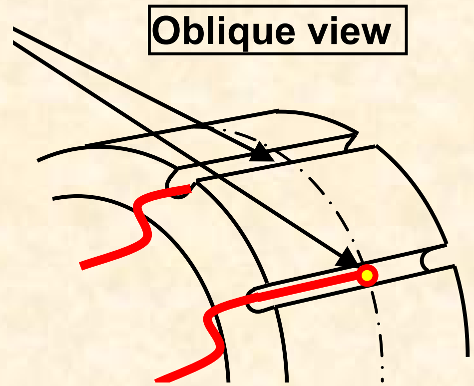
Nonlinear *F(X)*

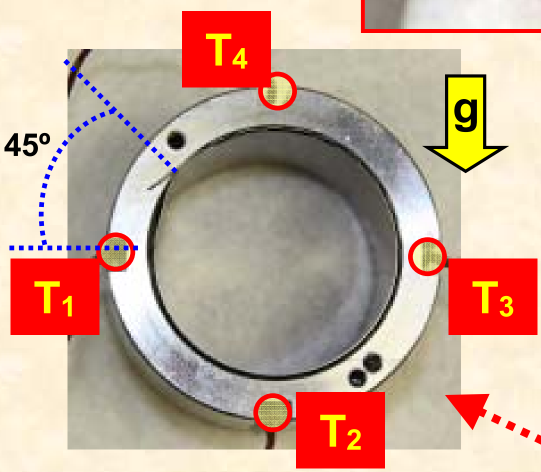
Hysteresis loop : **Mechanical energy dissipation**

due to dry-friction between top foil contacting bumps and bump strip layers contacting bearing cartridge

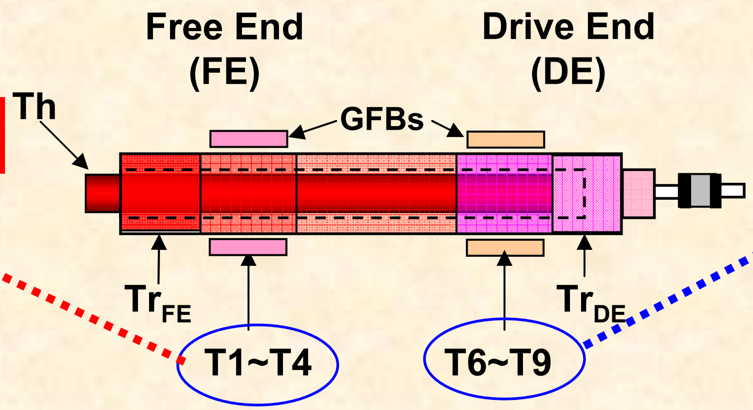
Thermocouples in test GFB

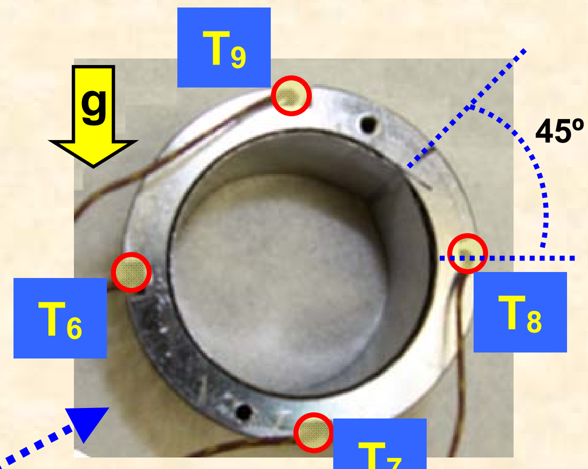






Free end (FE) bearing



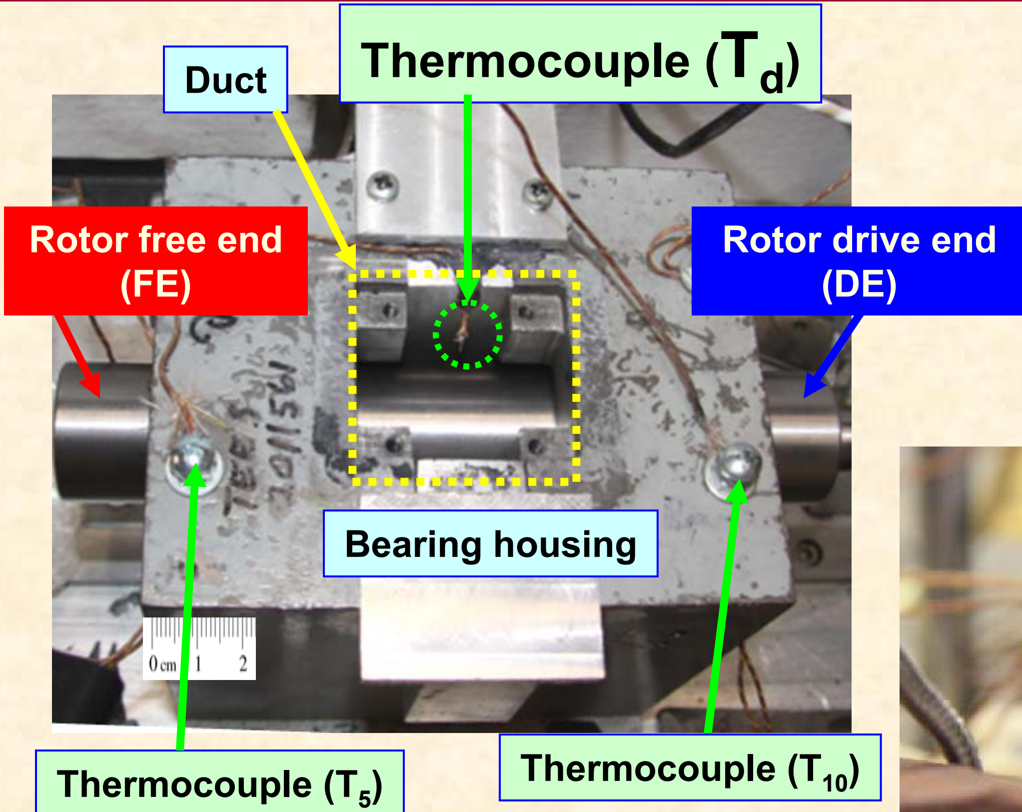


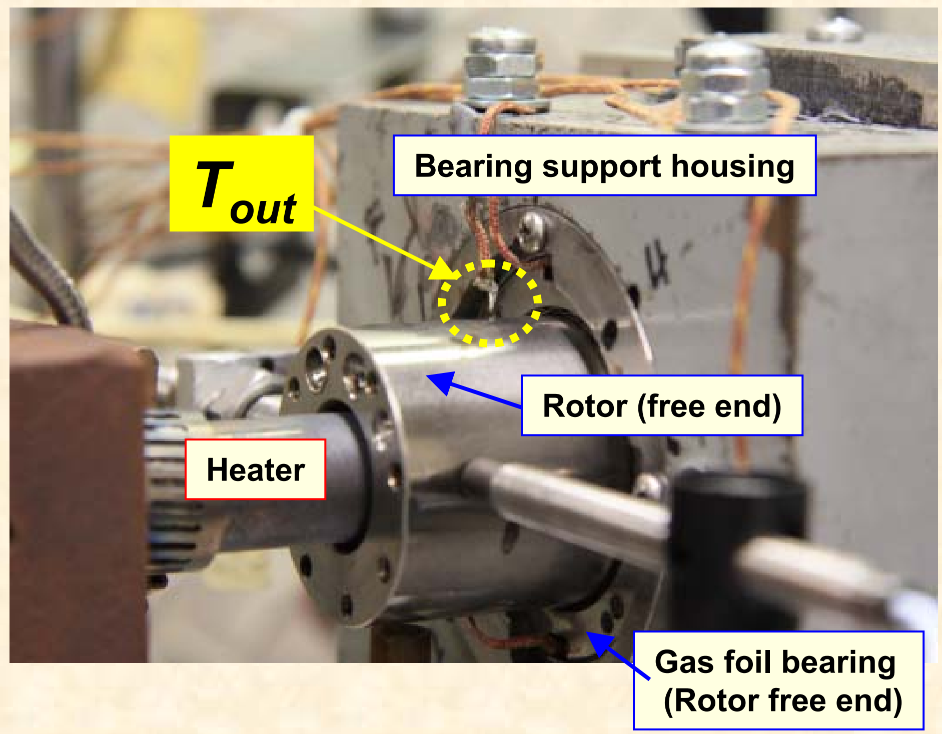
Drive end (DE) bearing

KIST FB uncoated (Generation I)

Four (4) thermocouples placed within machined axial slots.

Thermocouples in bearing housing

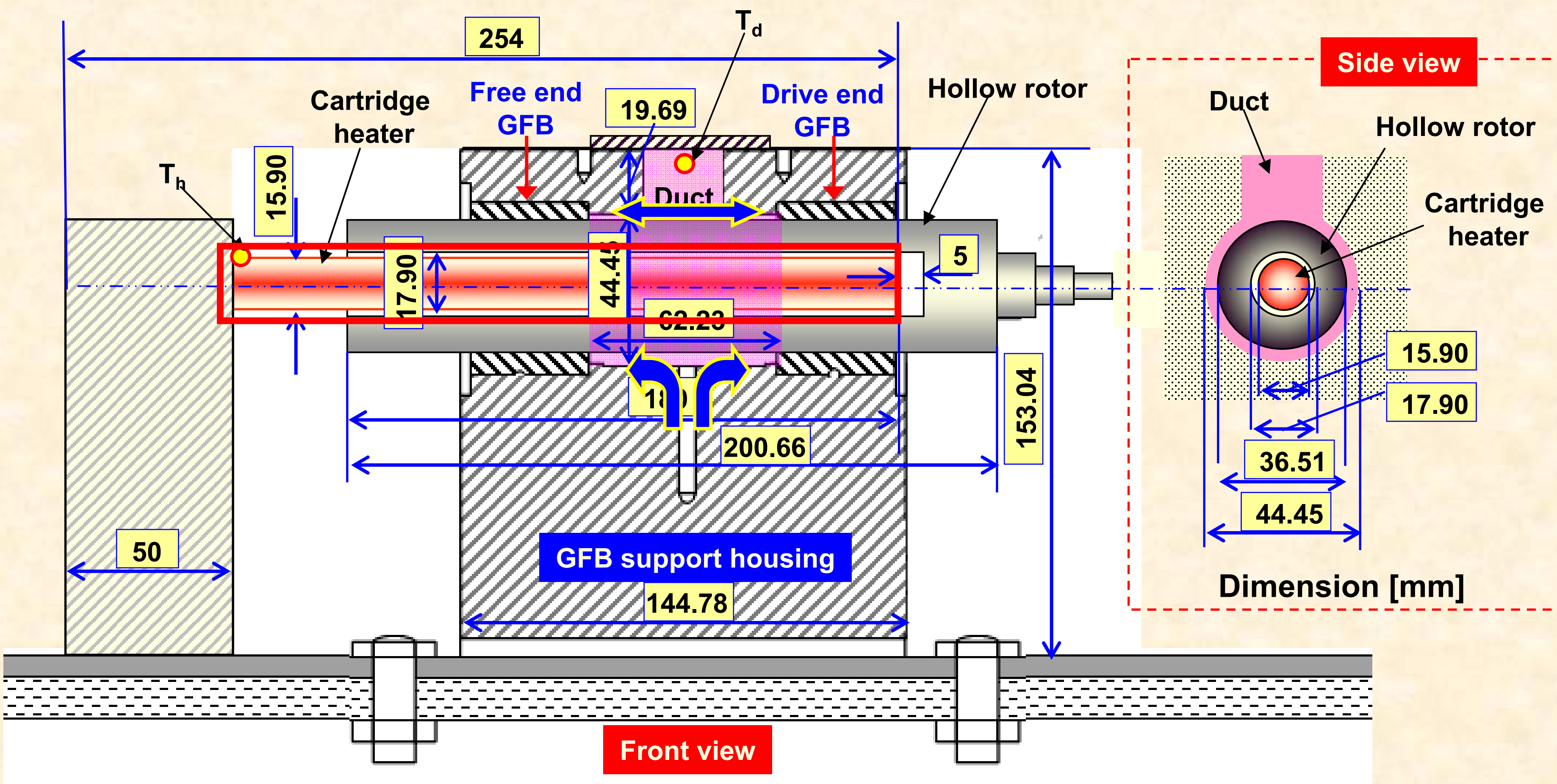




Forced cooling stream temperature

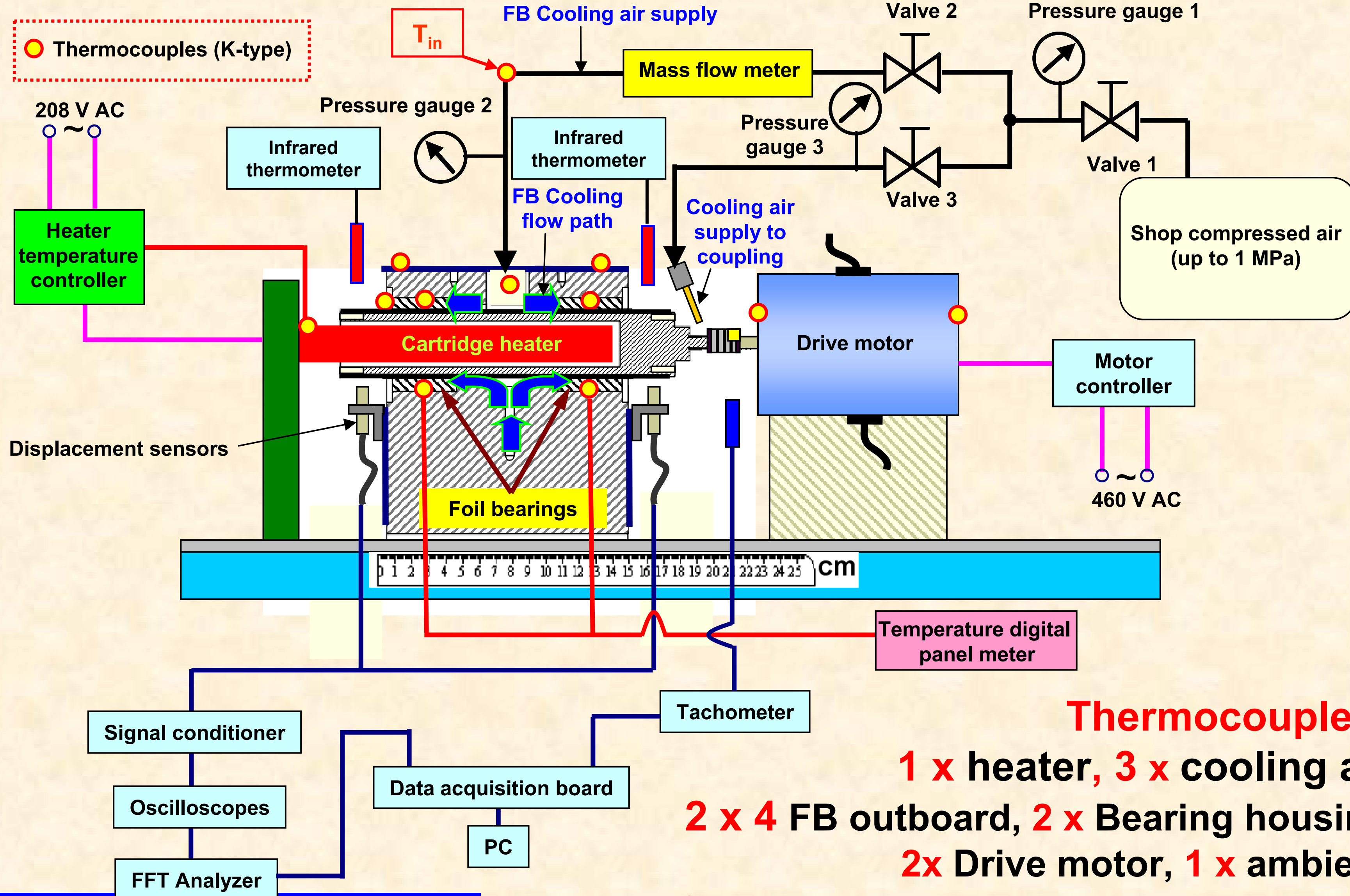
1 in housing duct + 1 at outboard plane of free end bearing

Hot rotor-GFB test rig



Dimensions

Hot rotor-GFB test rig



Instrumentation

Thermocouples:

1 x heater, 3 x cooling air

2 x 4 FB outboard, 2 x Bearing housing

2x Drive motor, 1 x ambient

\+ infrared thermometers 2 x rotor surface

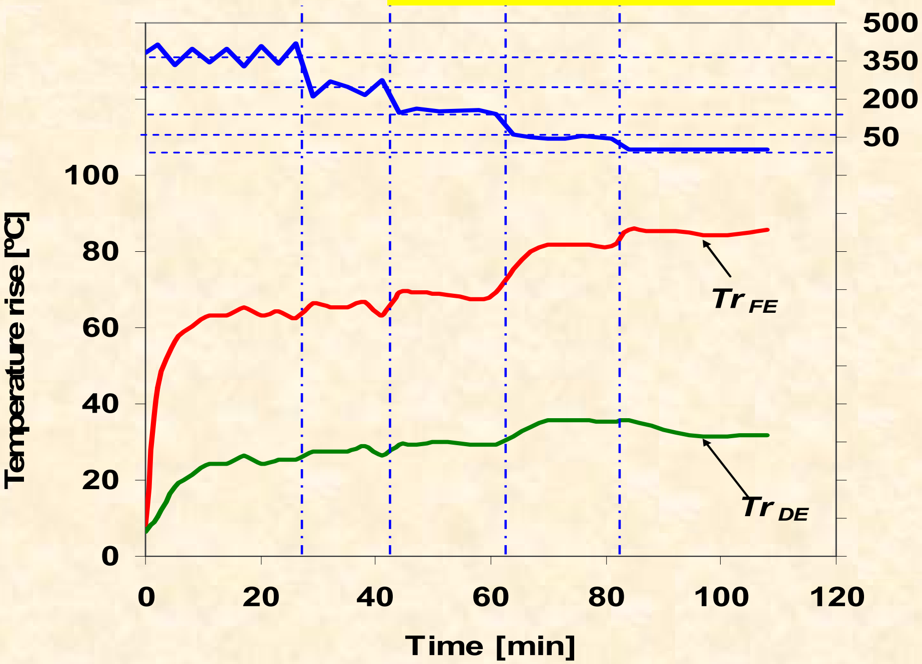
(Total = 19)

Test Cases

Test case #	Heater set temperature [ºC]	Rotor speed [krpm]	Set cooling flow rate (into two bearings) [L/min]	Time [min]
1	65	0	350 → 250 → 150 → 50 → 0	87
2	100	0	350 → 250 → 150 → 50 → 0	84
3	150	0	350 → 250 → 150 → 50 → 0	108
4	65	10→ 20 → 30	350 → 250 → 150 → 50	248
5	100	10→ 20 → 30	350 → 250 → 150 → 50	266
6	150	10	350 → 250 → 150 → 50	136
7	Off	30	350	30
8	65	30	350	30
9	100	30	350	30
10	100	30	50	30
			Overall 1049 min	

Rotor OD Temps. vs Time

No rotor spinning

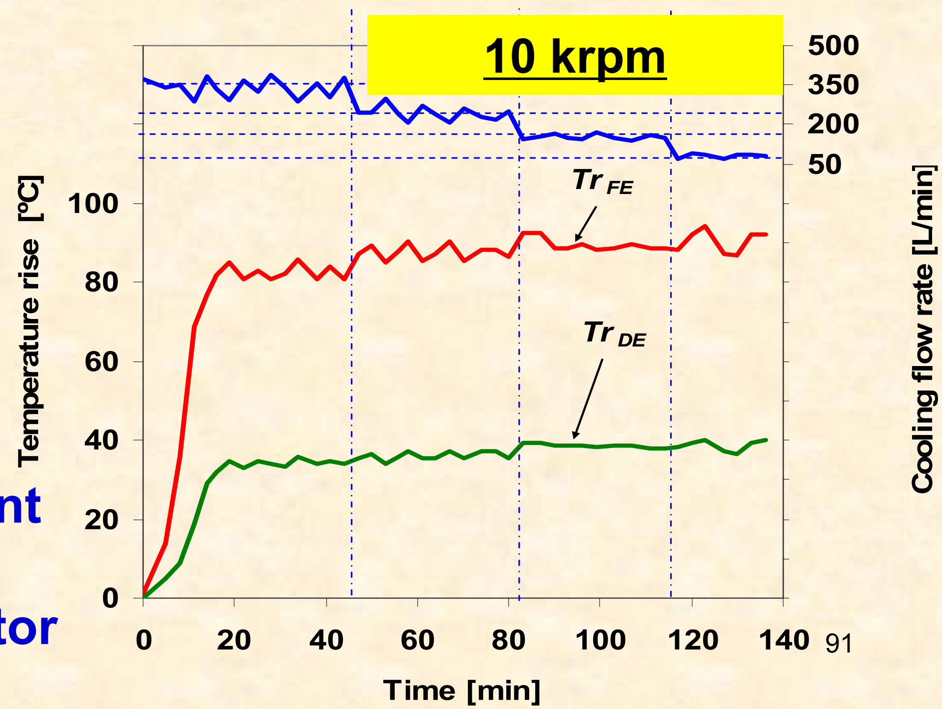


Test cases #1 and #4

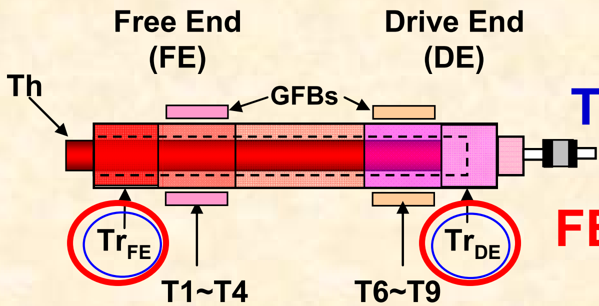
Heater set temperature = 150ºC

Temperatures on shaft OD increase steadily with elapsed test time at each rotor speed and cooling flow rate condition

10 krpm



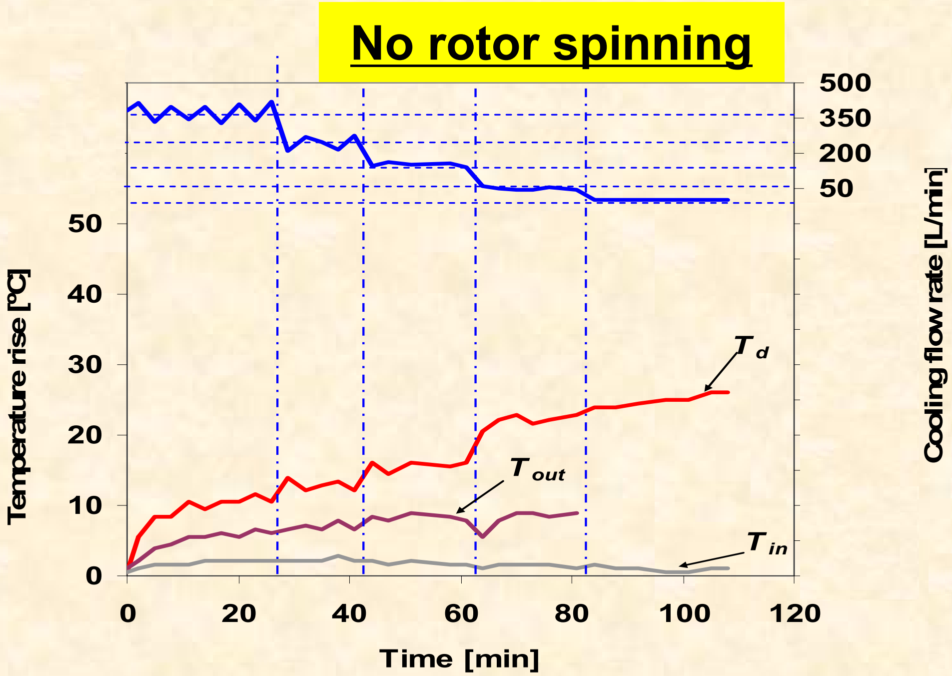
The rotor is a source of HEAT!



Thermal gradient
Hot to cold
FE rotor >DE rotor

Duct & Outboard temperature rises vs time

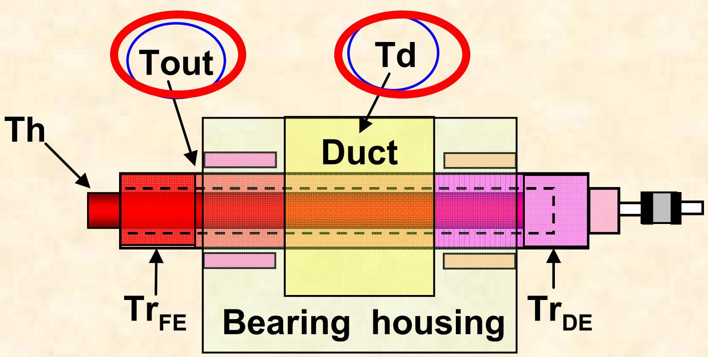
Test cases #1 and #4

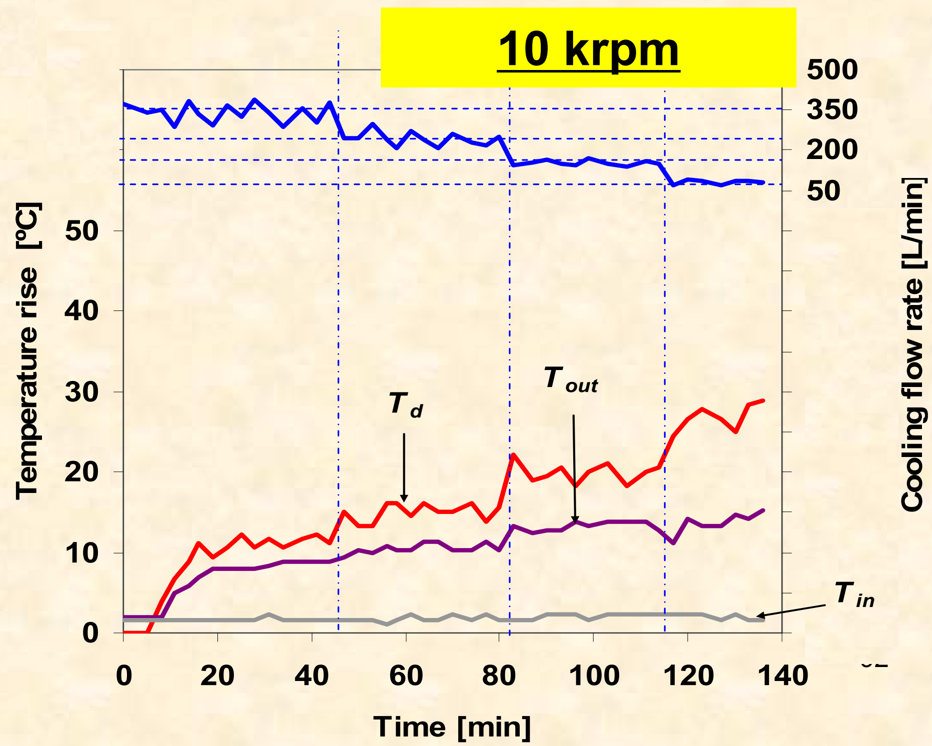


Heater set temperature = 150ºC

Tin ~ Tamb

Td > Tout > Tin



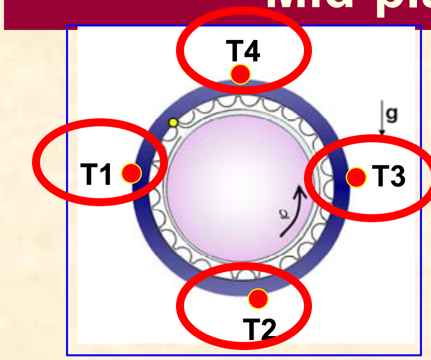


FE bearing temperature rise vs time

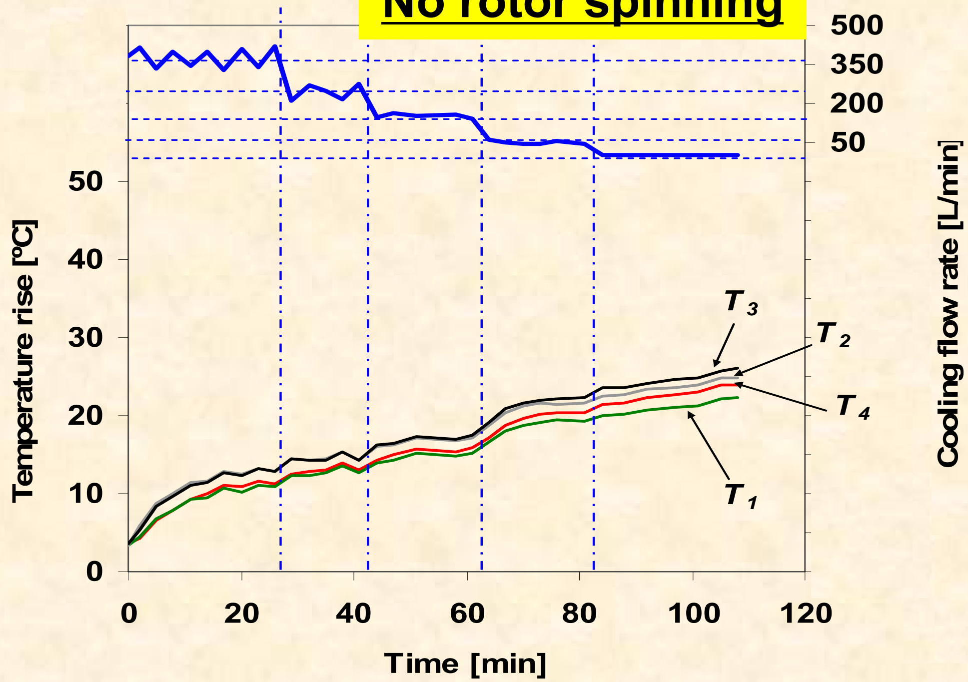
Test cases #1 and #4

Heater set temperature = 150ºC

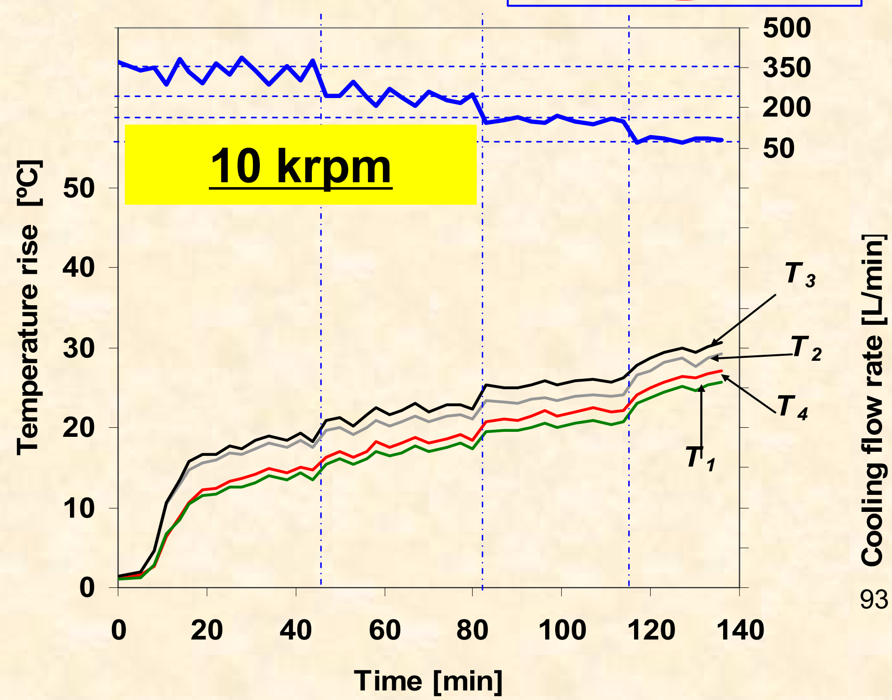
Free End Bearing Mid-plane



No rotor spinning



10 krpm



T3>T2>T4>T1 due to differences in rotor OD temp. along its circumference

Rotor speed makes bearings slightly hotter (a few deg C)

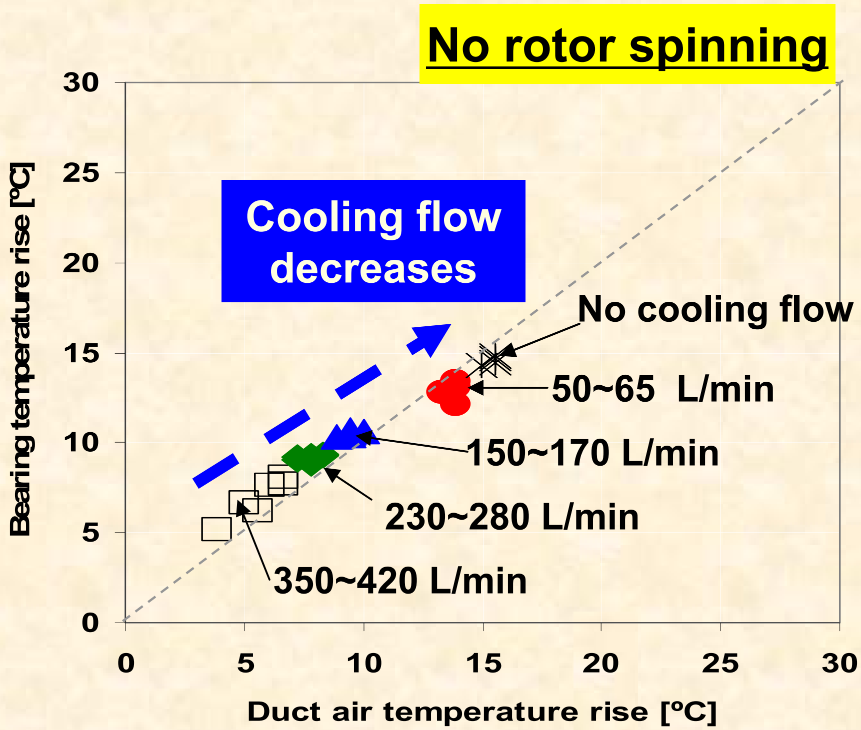
Bearing temperature rise vs duct temp.

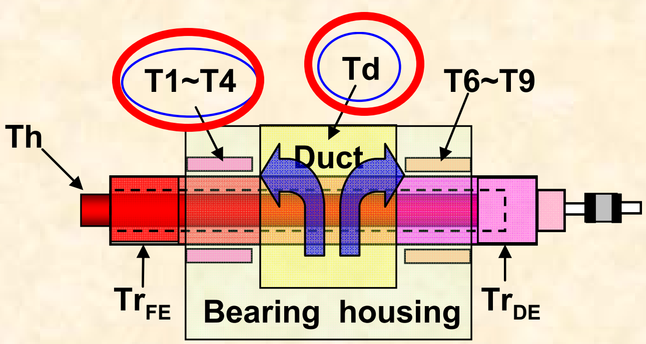
Test cases #2 and #5

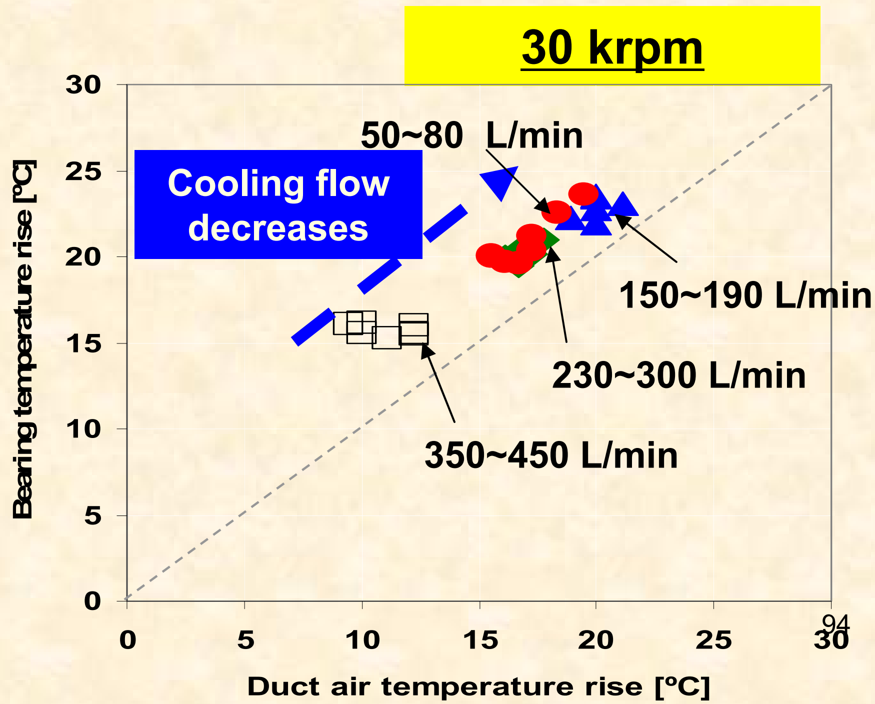
Heater set temperature = 100ºC

Free end Bearing

Temperatures on bearings ODs linearly increase with duct air temperature as the cooling flow rate into the bearings decreases



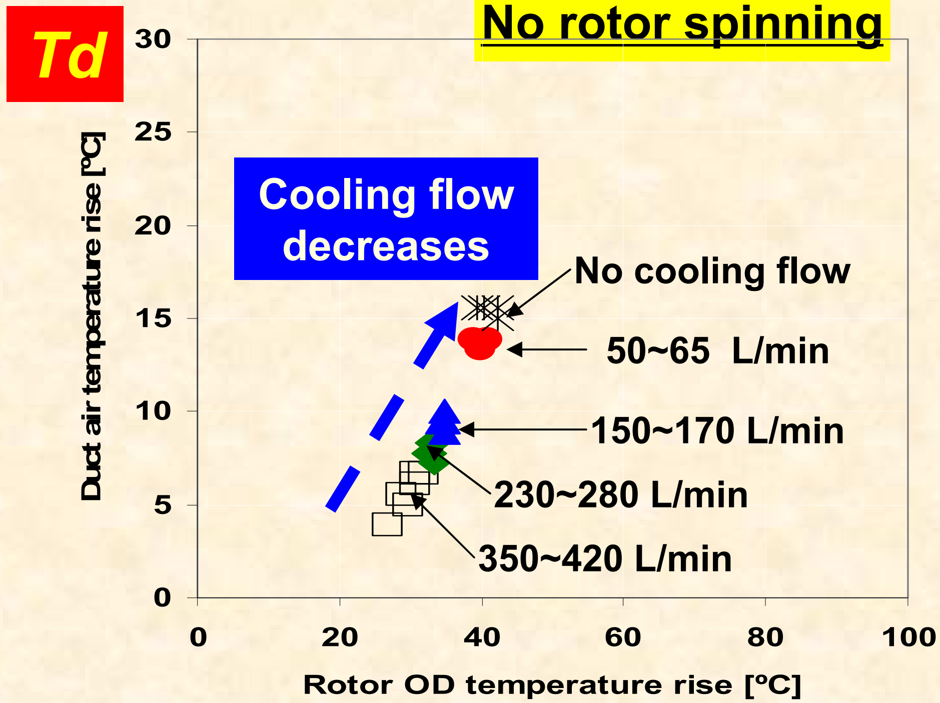




Rise in temperatures: Duct vs Rotor OD

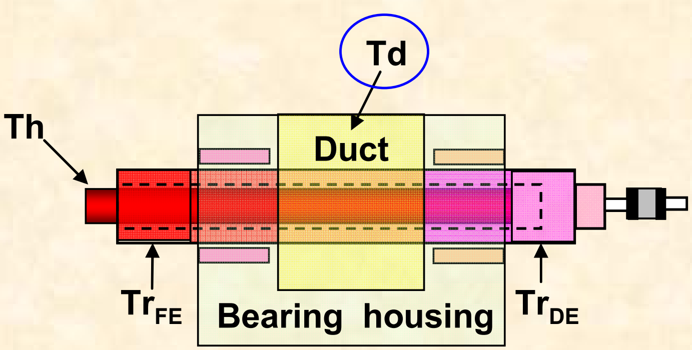
Test cases #2 and #5

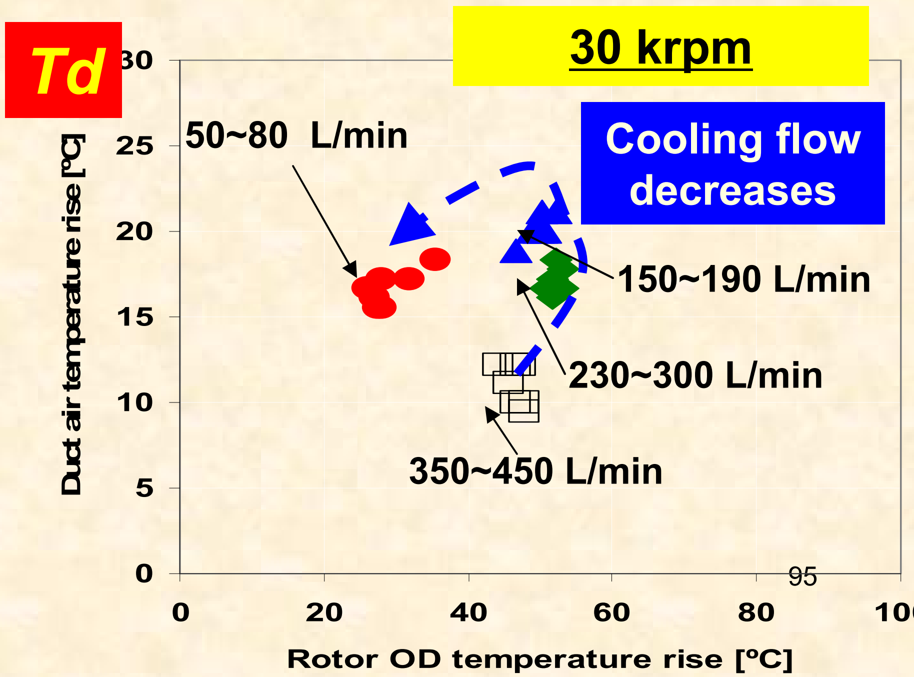
Heater set temperature = 100ºC



Td decreases with cooling flow due to longer residence of air particles in the duct

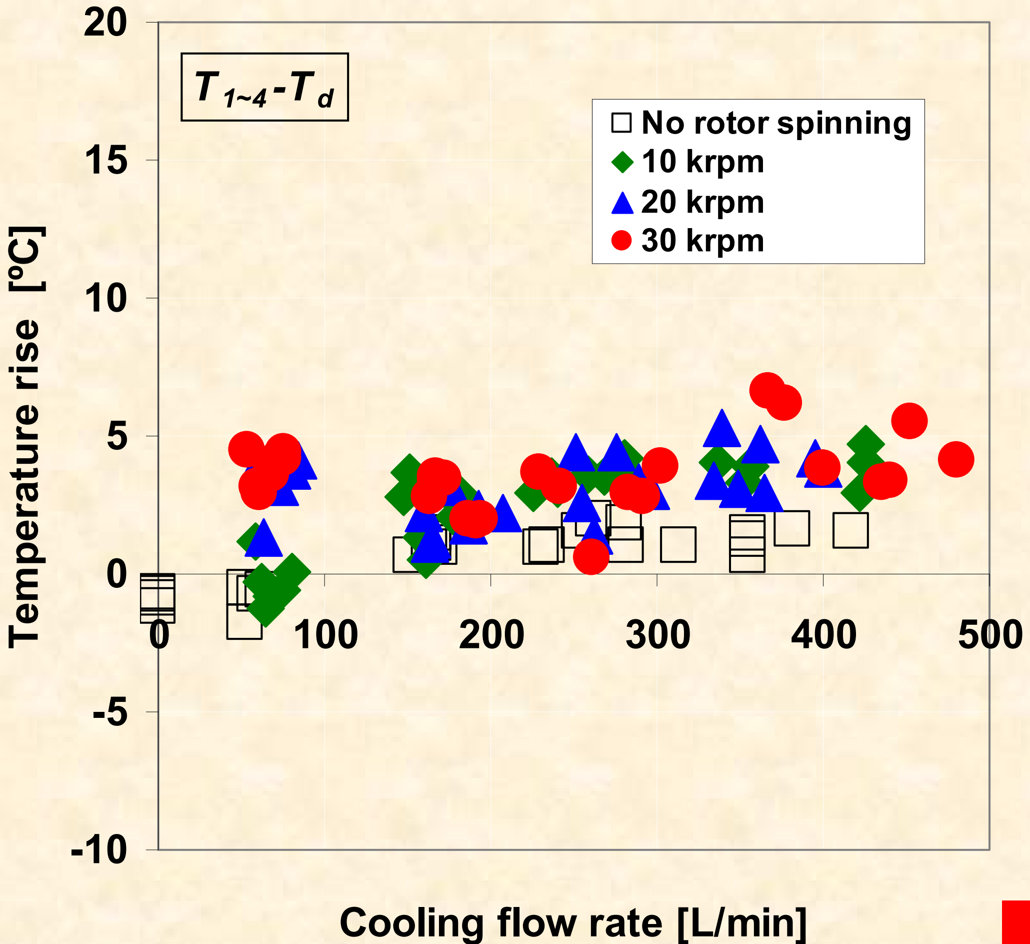
Td increases with rotor speed due to windage effect



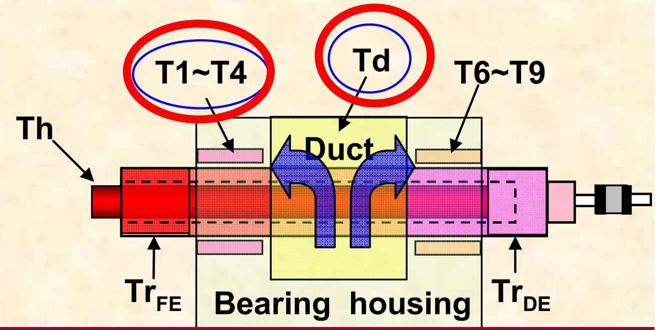


Bearing OD temperature rise vs. cooling flow

Test cases #2 and #5



Heater set temperature = 100ºC



Free end Bearing

Temperature difference (T-Td) is invariant while increasing cooling flow rate!

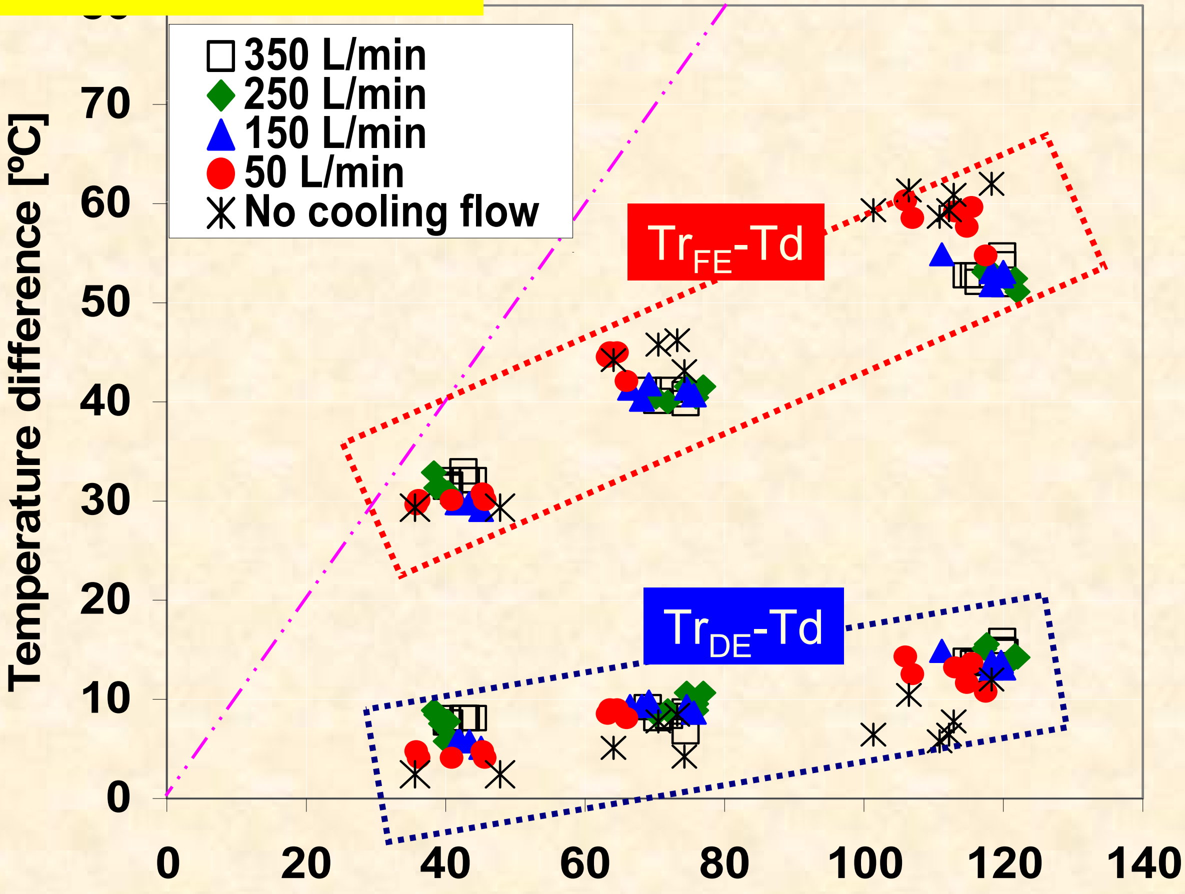
?

Bearing temperature is a small fraction of the heat source temperature (heater and duct air).

Rotor OD temp. vs heater temp. (- duct ???)

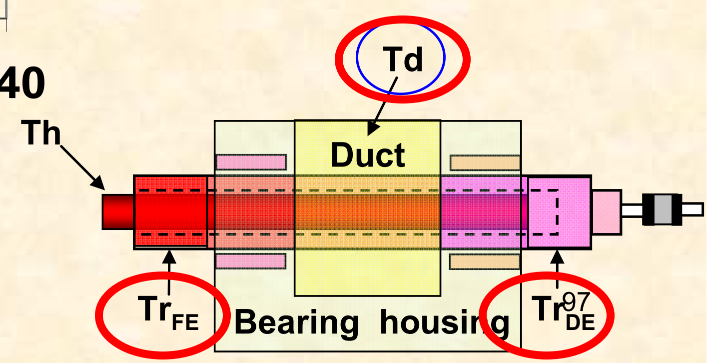
No rotor spinning

Test cases #1~#3



Cartridge temperature (T_{hs}) increases

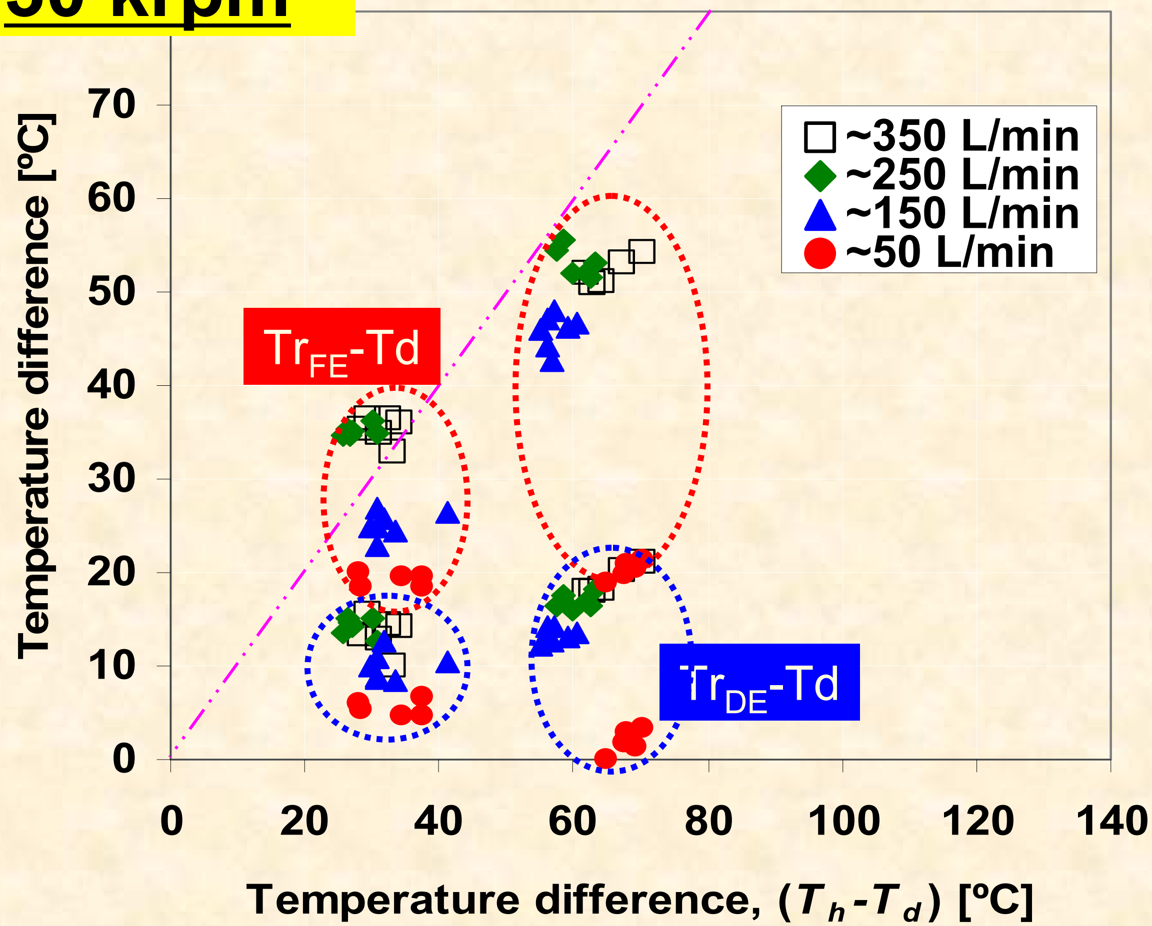
The cooling gas flow removes heat from the top foil back surface, thus *cooling* the rotor OD



Rotor OD temp. vs heater temp.

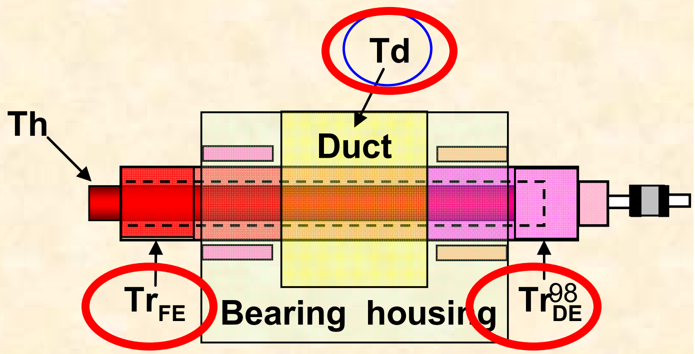
30 krpm

Test cases #4 and #5



Cartridge temperature (T_{hs}) increases

Rotor OD temp (relative to duct temp Td) decrease with cooling flow

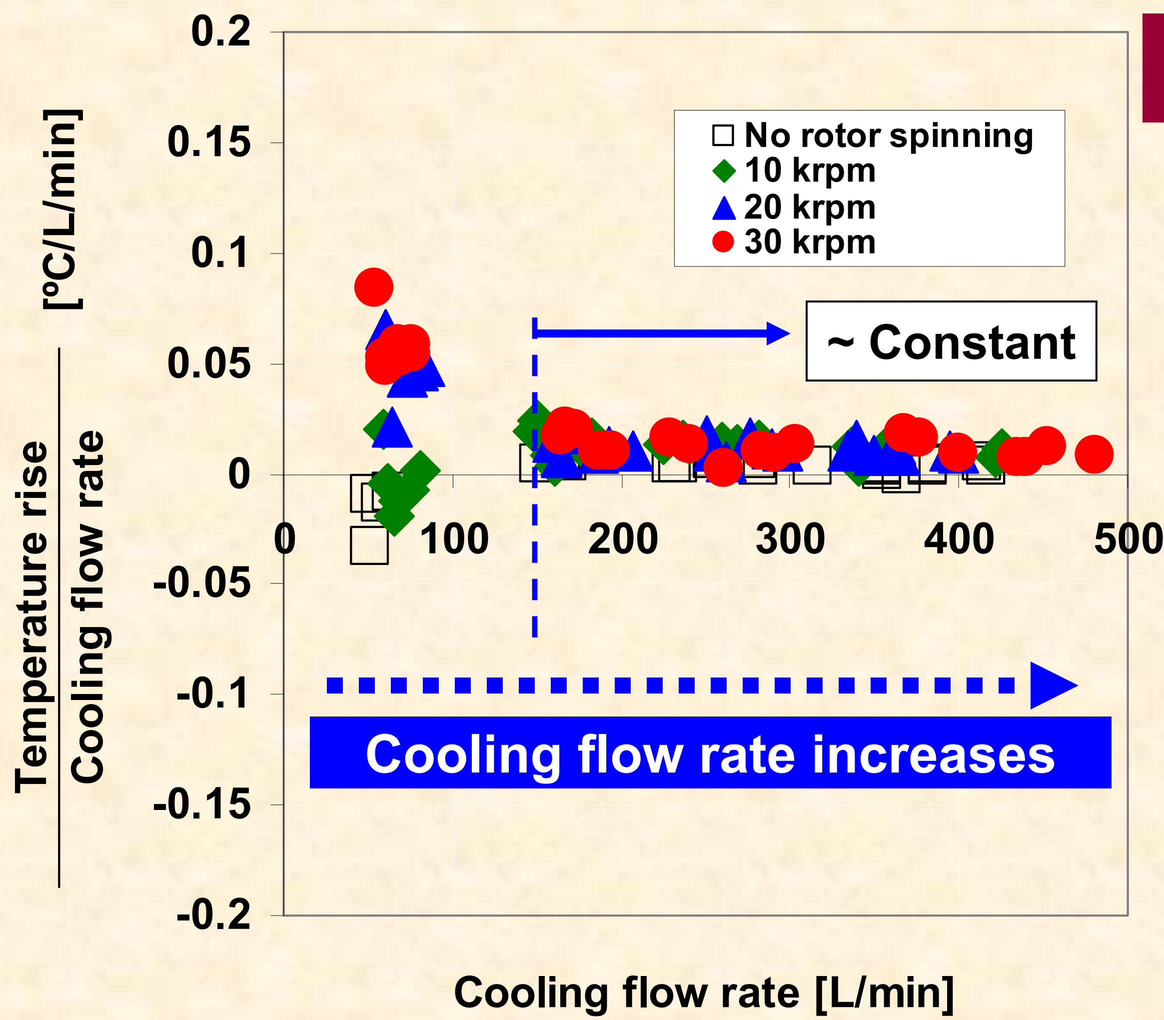


Cooling Capability: Bearing OD temp.

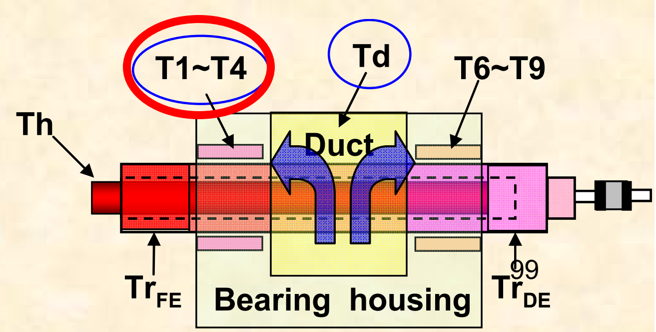
Test cases #2 and #5

Heater set temperature = 100ºC

Free end Bearing

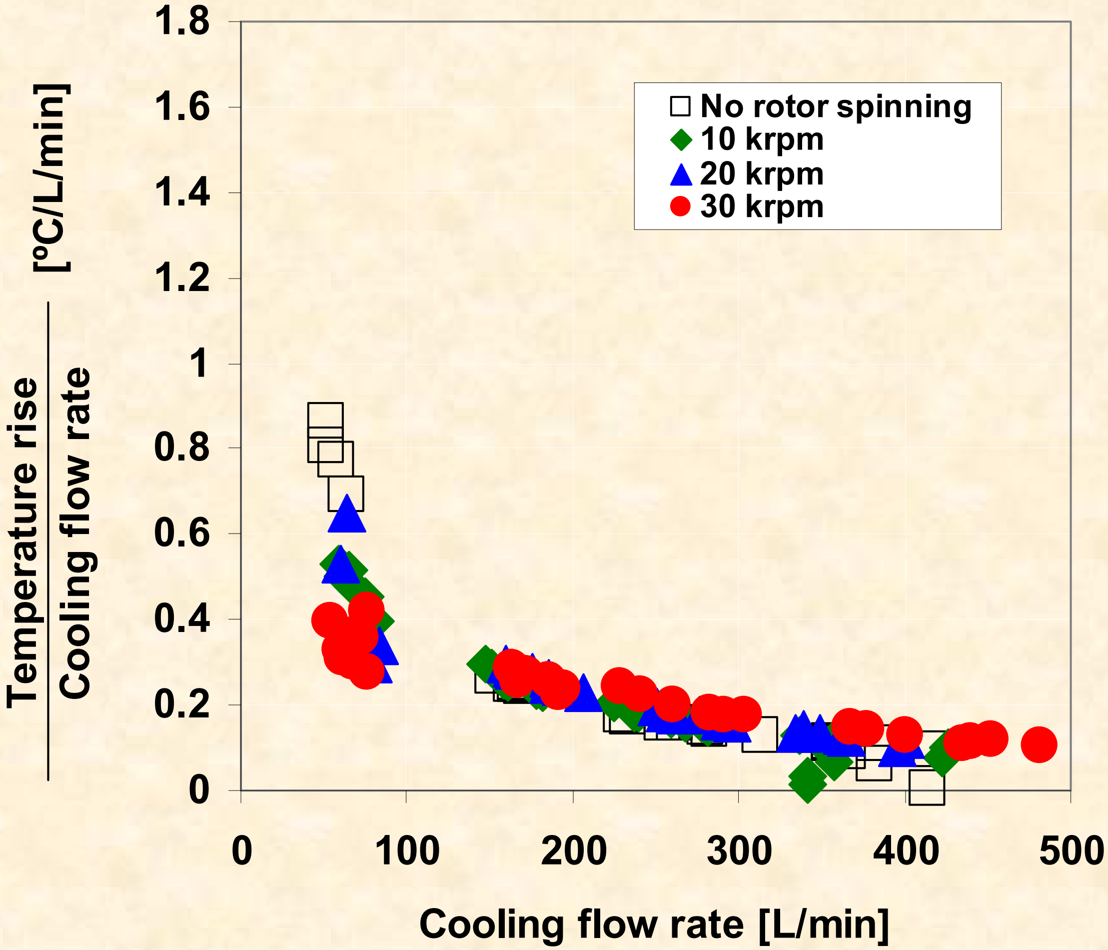


The cooling capability of the forced axial flow on the bearing temperatures **changes little with flow rate**



Cooling Capability: Rotor OD temp.

Test cases #2 and #5



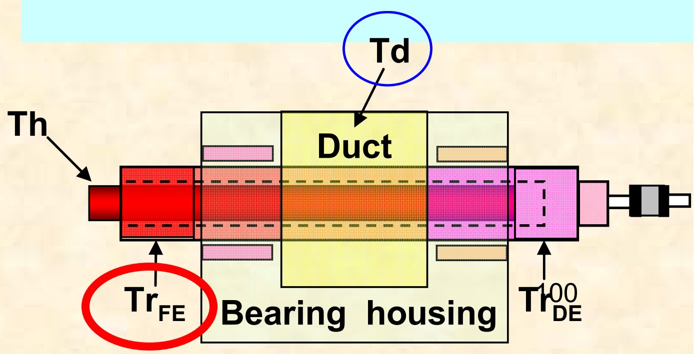
Cooling flow rate increases

Heater set temperature = 100ºC

Free end rotor

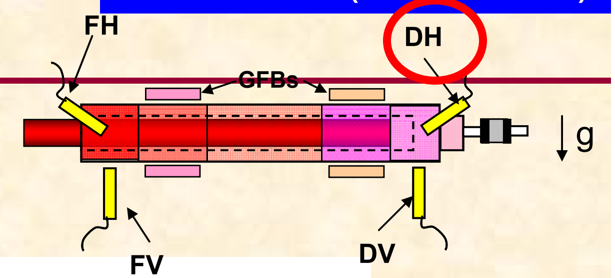
The cooling capability of the forced axial flow on the rotor temperatures appears to have an exponential decay character.

The cooling effectiveness of the forced cooling stream is most distinct at the free end rotor OD.

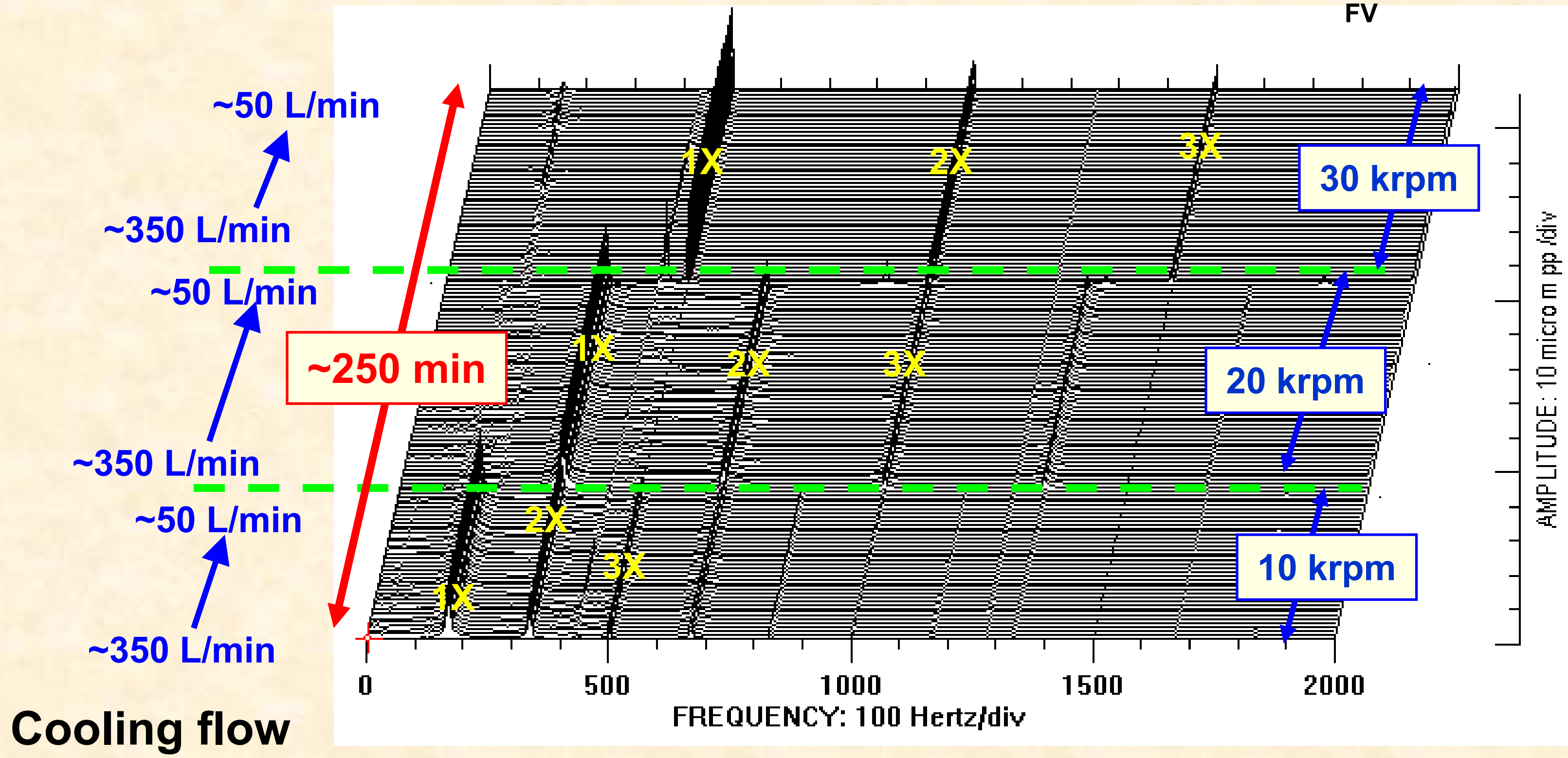


Waterfalls of rotor motion

Drive End (Horizontal)



Baseline, Ths=65C



Test case #4

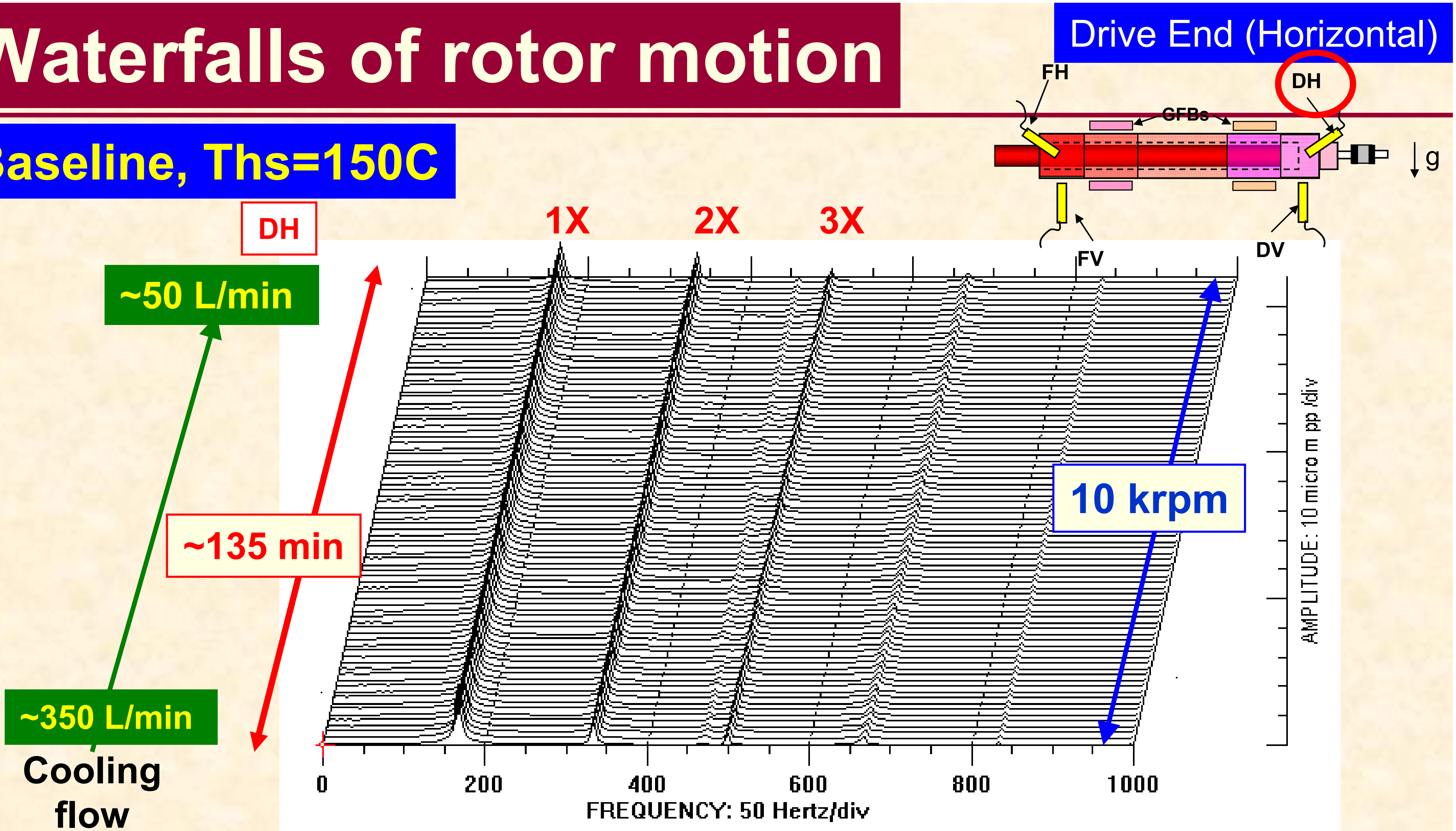
Free of sub synchronous whirl motions

Cooling flow rate does not affect amplitude and frequency contents of rotordynamic displacement

Rotor motion measurements

Waterfalls of rotor motion

Baseline, Ths=150C

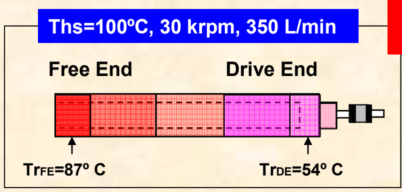


Test case #6

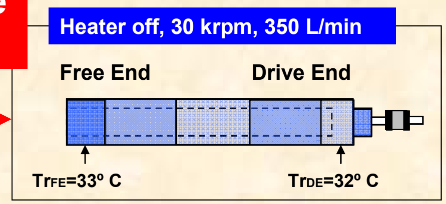
Rotor OD temperature does not affect rotor dynamic displacements!

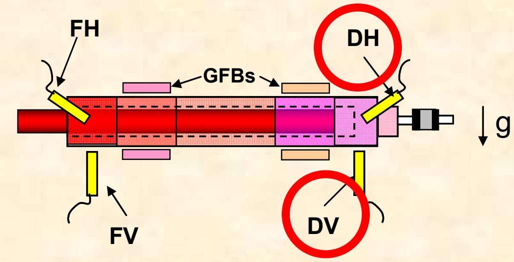
Synchronous rotor response: effect of shaft temp.

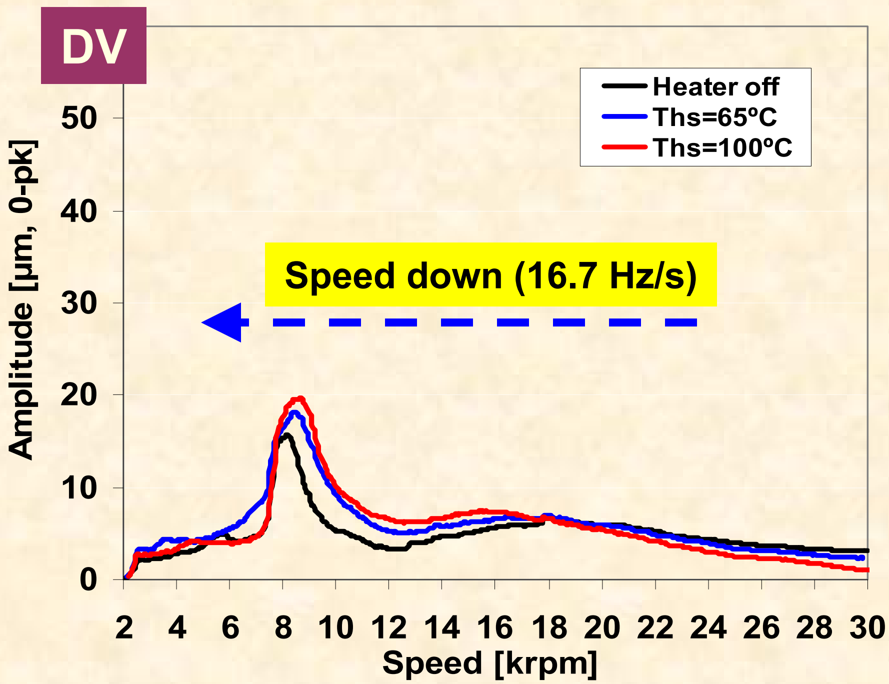
Test cases #7~#9

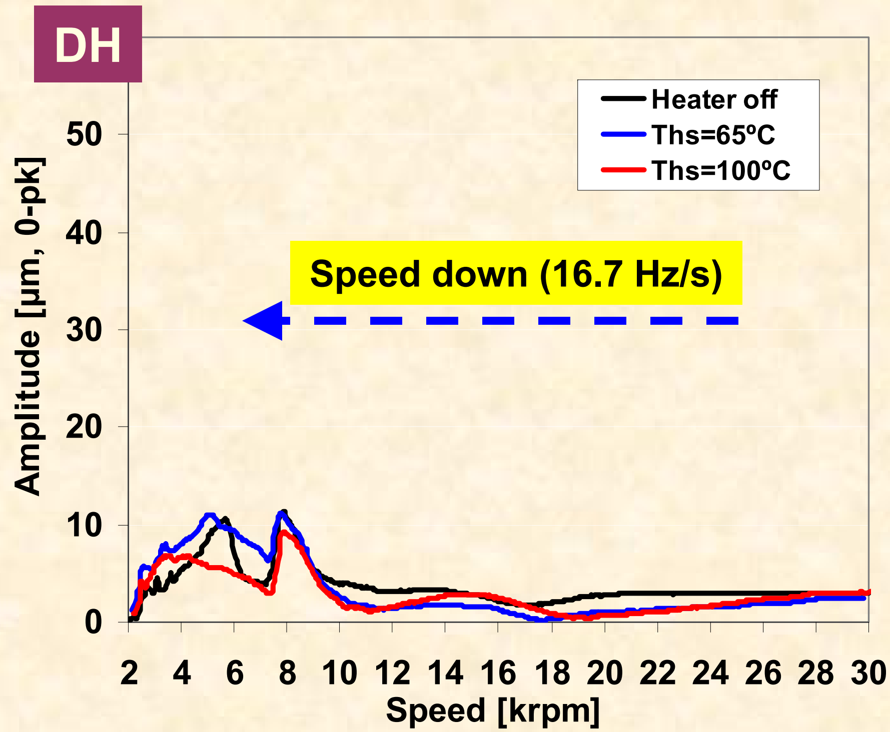


Cartridge temperature (T_{hs}) increases









Test cases #7~#9

Flexible rotor mode at ~90 krpm (rap test)

Critical speed (Rigid body mode) ~ 5 and 8 krpm

No major differences in responses between cold and hot

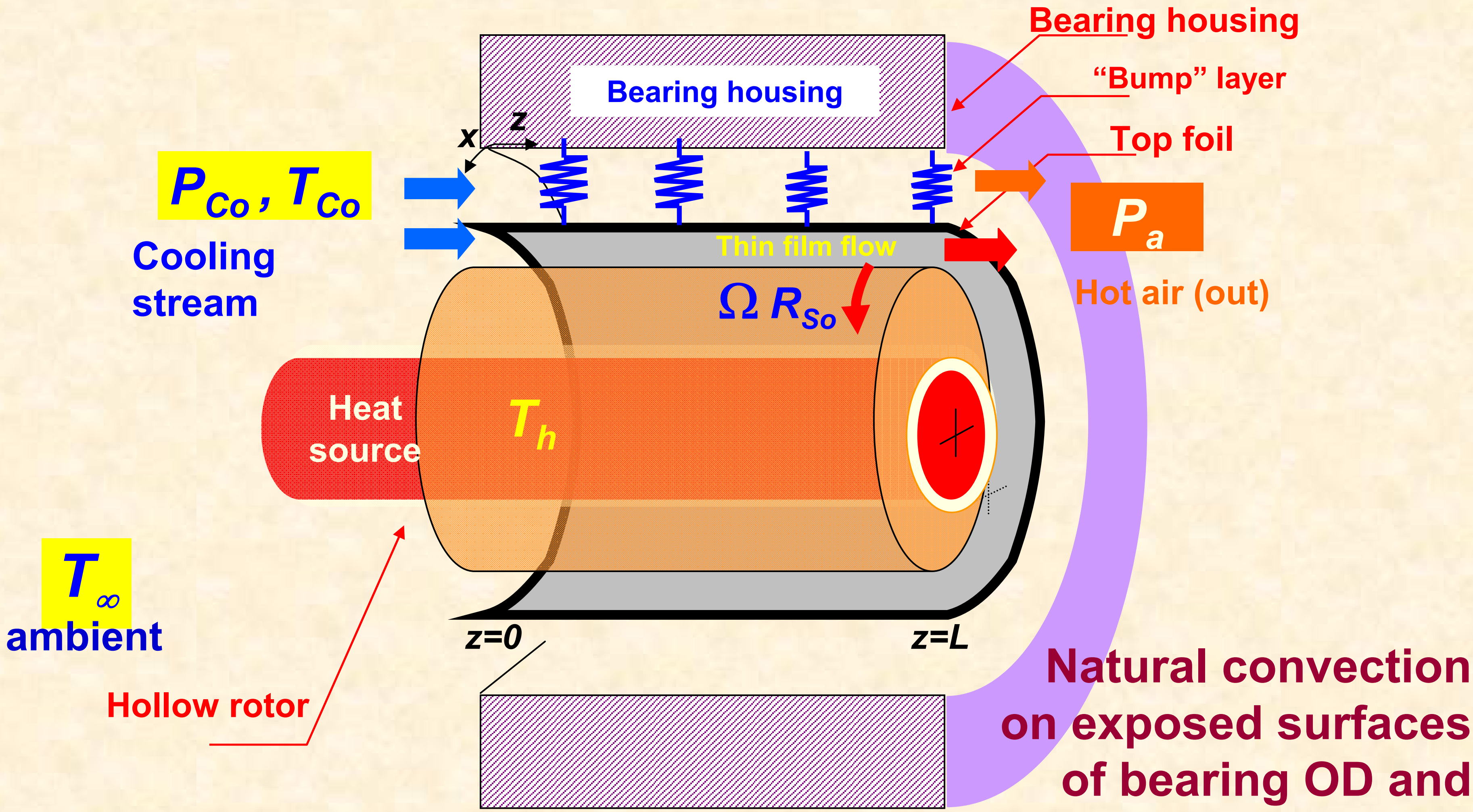
GFB TEHD model

By San Andrés and Kim (2008)

Gas film	**Reynolds eqn.** for hydrodynamic pressure generation **Energy transport eqn.** for mean flow temperature Various surface heat convection models Mixing of temperature at leading edge of top foil
Top foil & underspring	**Thermo-elastic deformation eqns.** Finite Elements and discrete parameter for bump strips. Thermal energy conduction paths to side cooling flow and bearing housing.
Bearing clearance	Material properties (gas & foils) = *f* (Temperature) Shaft thermal and centrifugal growth Bearing thermal growth

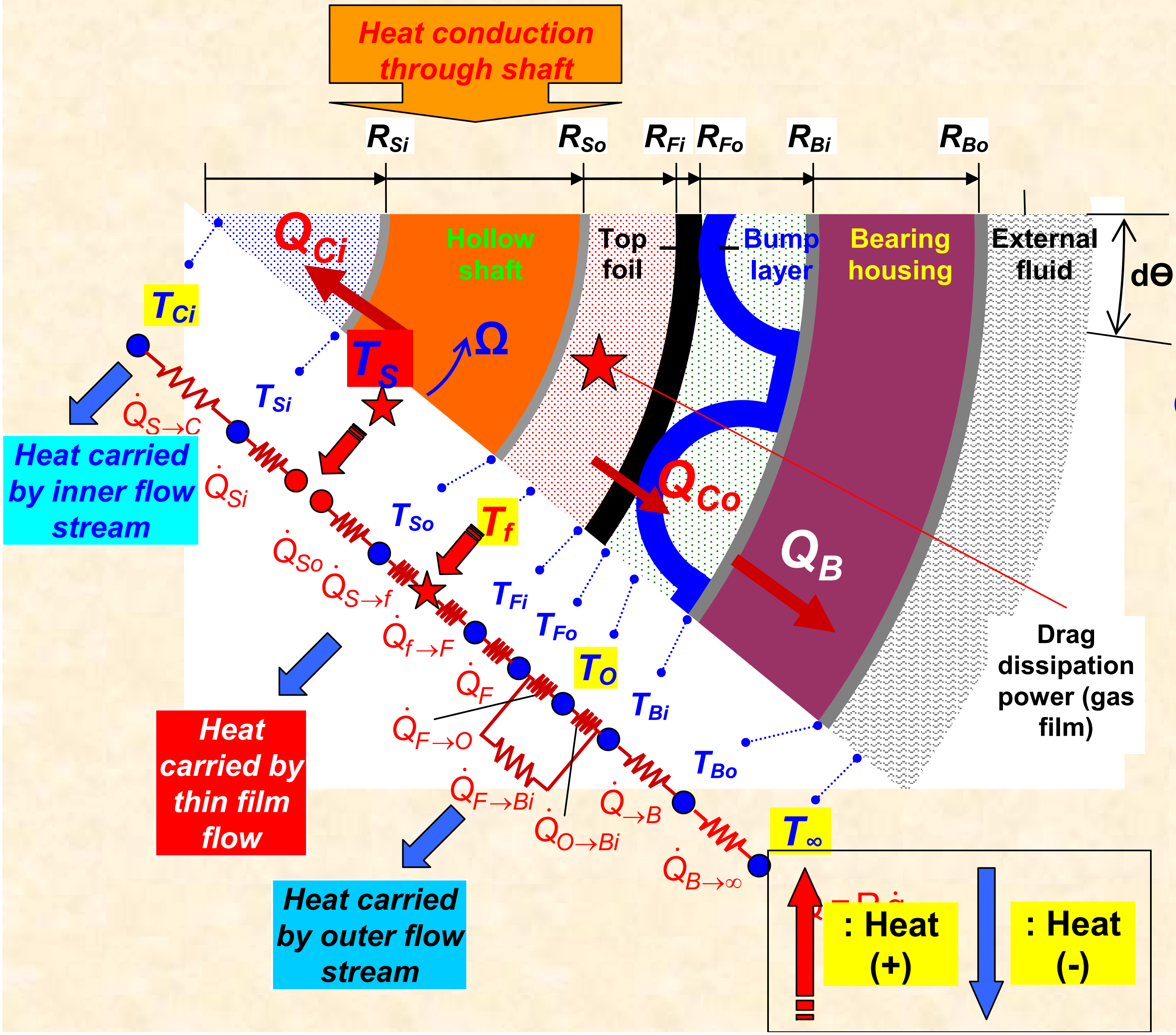
Excel GUI + executable licensed by TAMU

Recap: test rotor and FB



Schematic view of rotor and heater cartridge + side cooling stream

Heat flow paths in rotor - GFB system



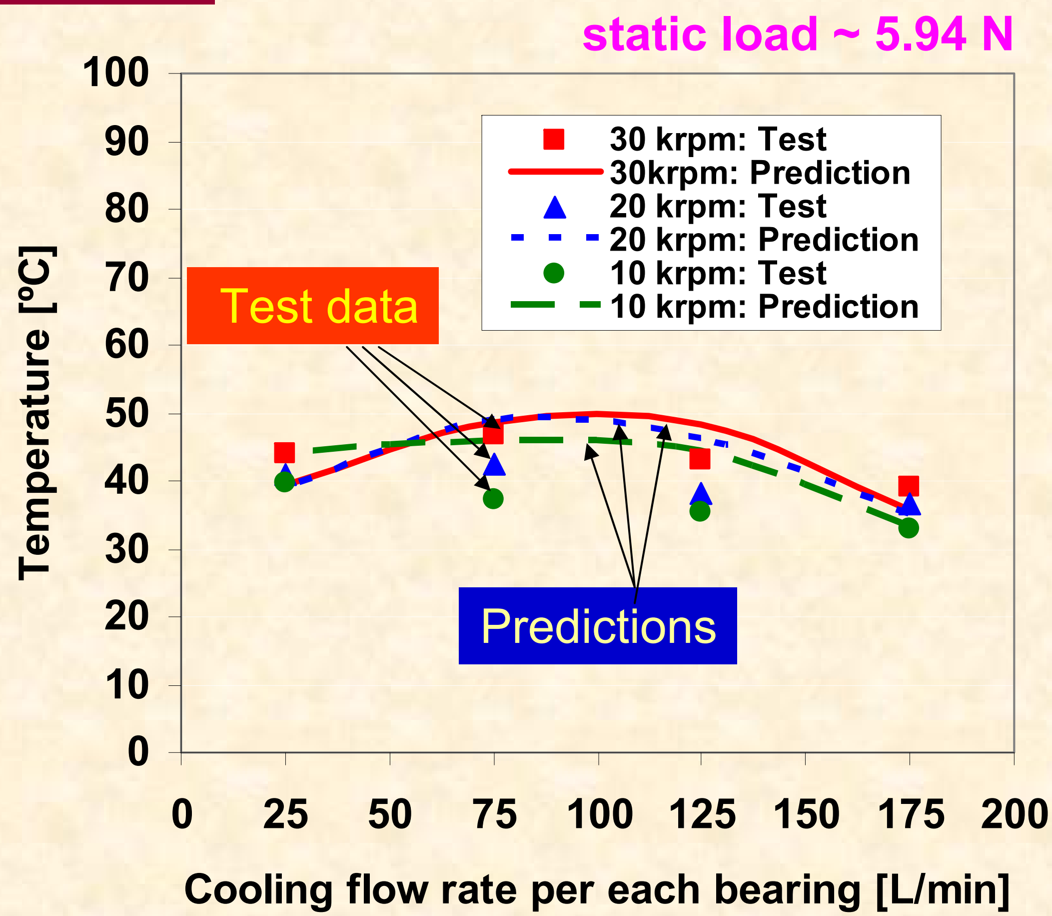
Heat flows & thermal resistances in a GFB & hollow shaft

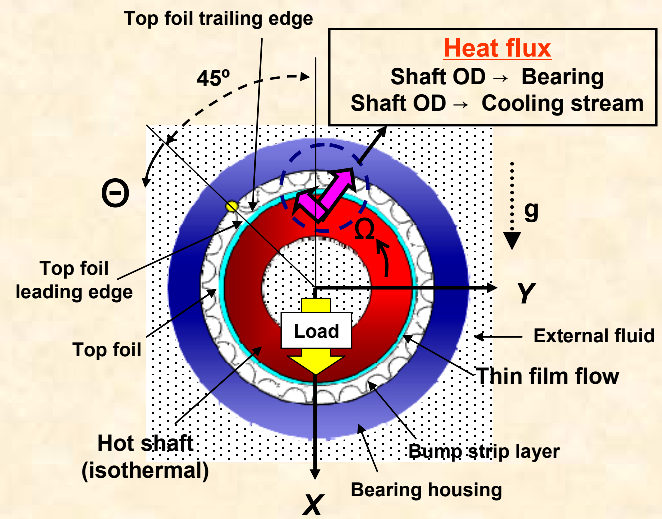
Heat conducted into bearing

Cooling gas streams carry away heat

Bearing temperature predictions & tests

Free end FB



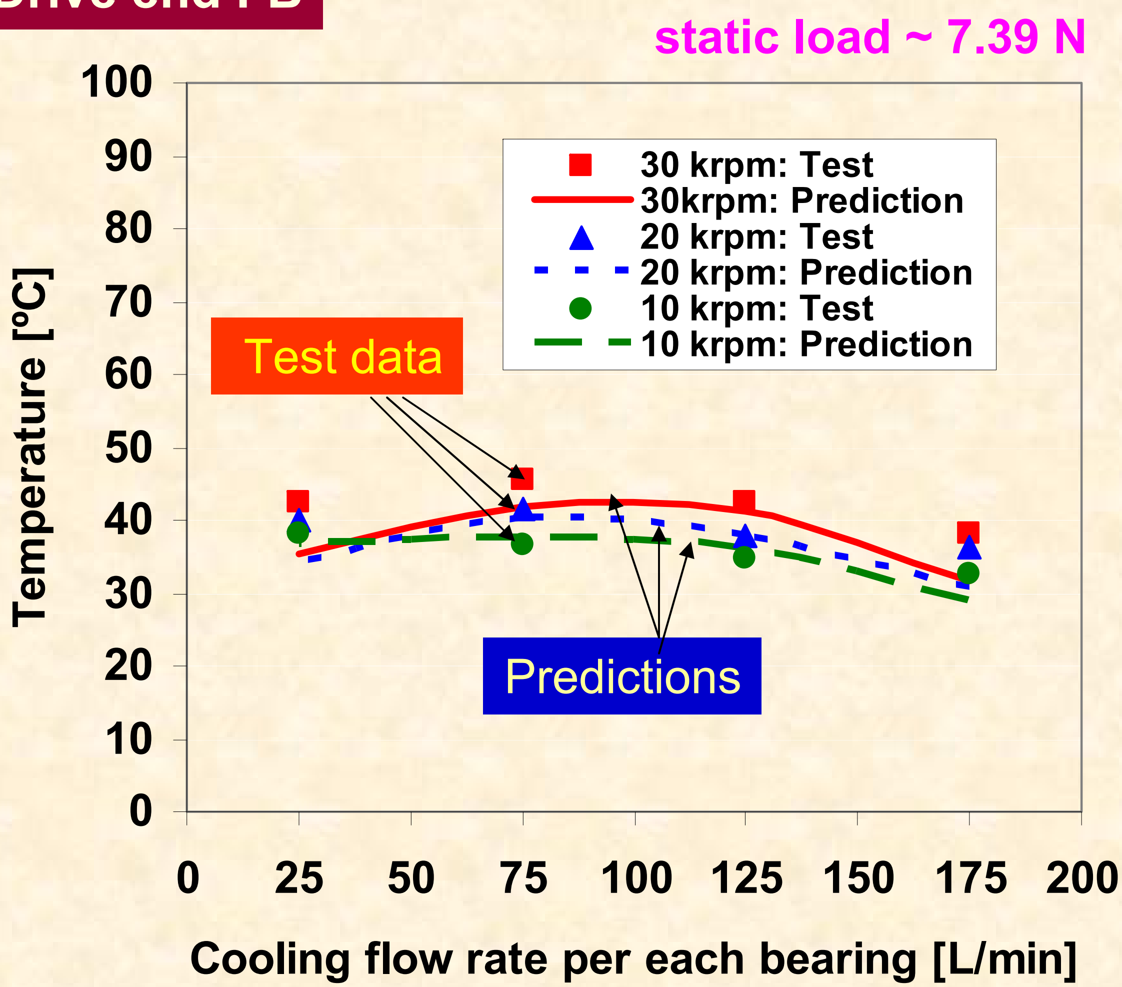


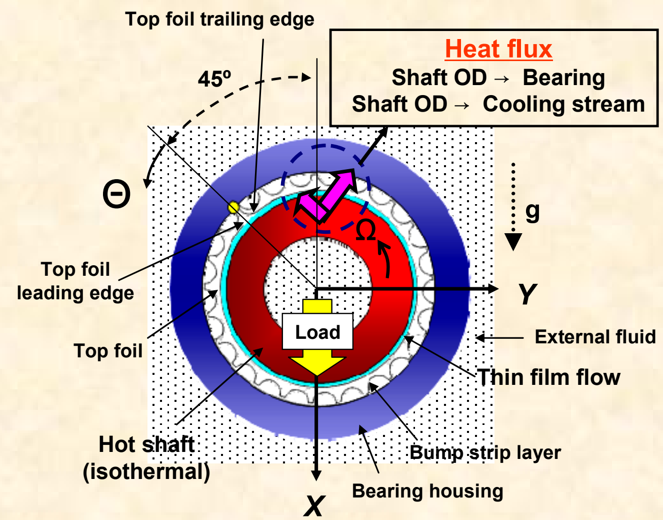
Input:
Measured ambient, rotor OD & inlet cooling flow temps.

Predictions agree with test data!!

Bearing temperature predictions & tests

Drive end FB



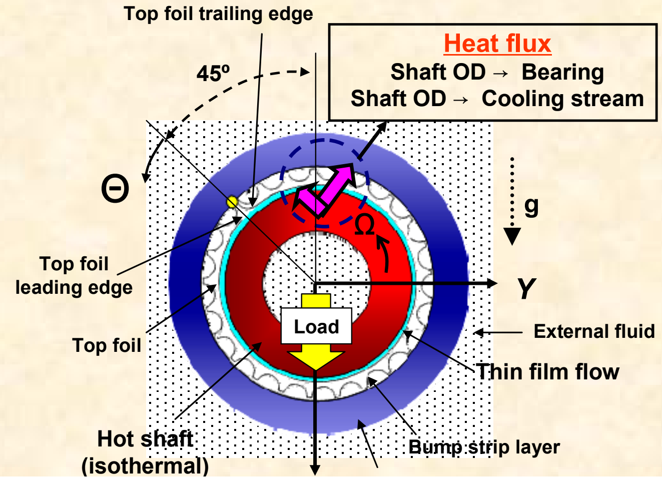


Predictions follow test data: better at DEB since smaller temperature gradient along heater axial length

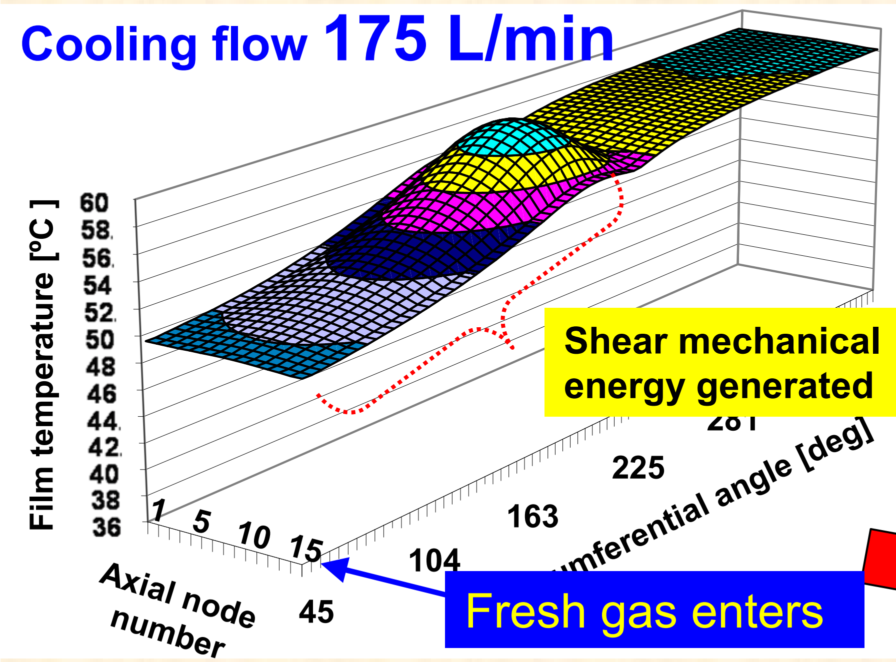
Predictions: *Temperature* fields

Test case #5

***Ths*=100C,30 krpm, Free end bearing**

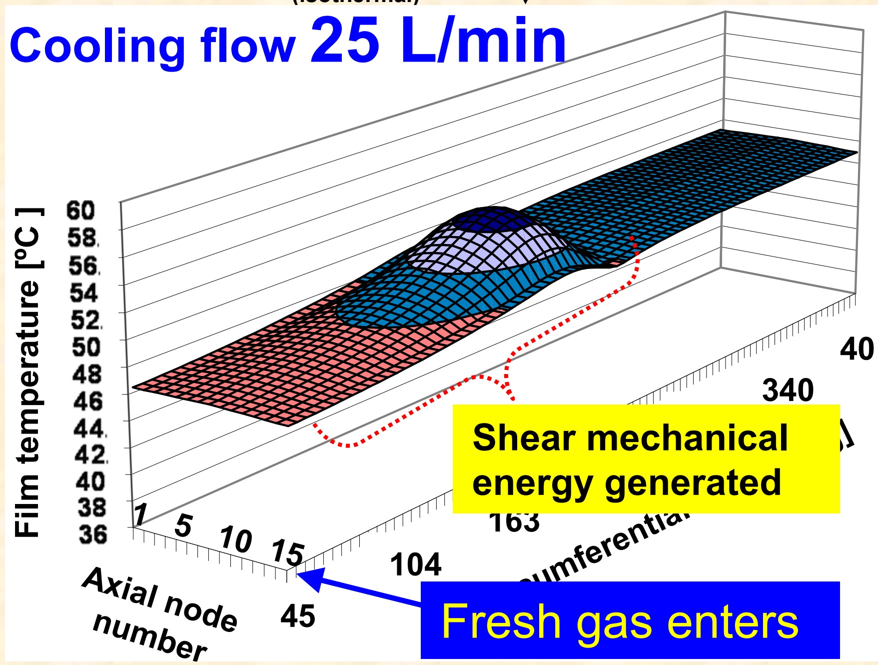


Cooling flow 175 L/min



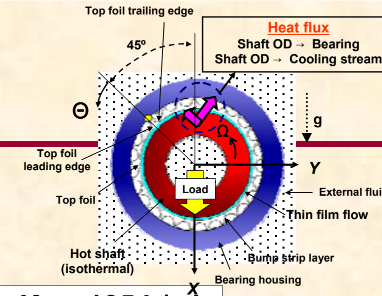
Shaft = 80C
Cooling stream inlet = 33C

Cooling flow 25 L/min



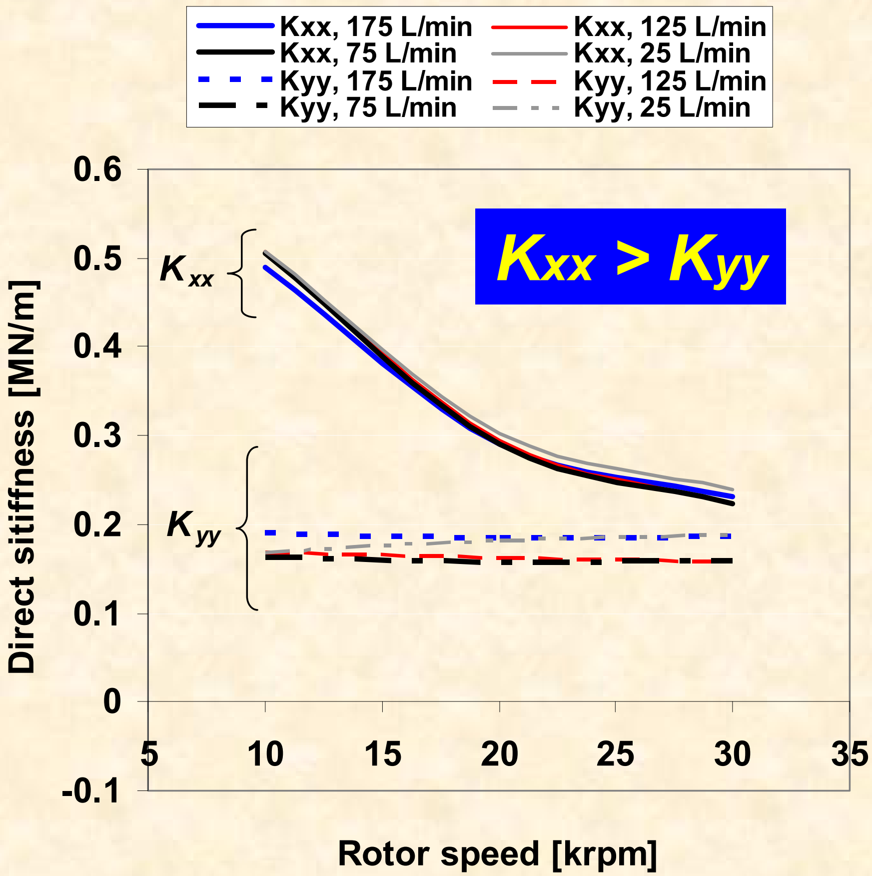
Shaft = 56C
Cooling stream inlet = 41C

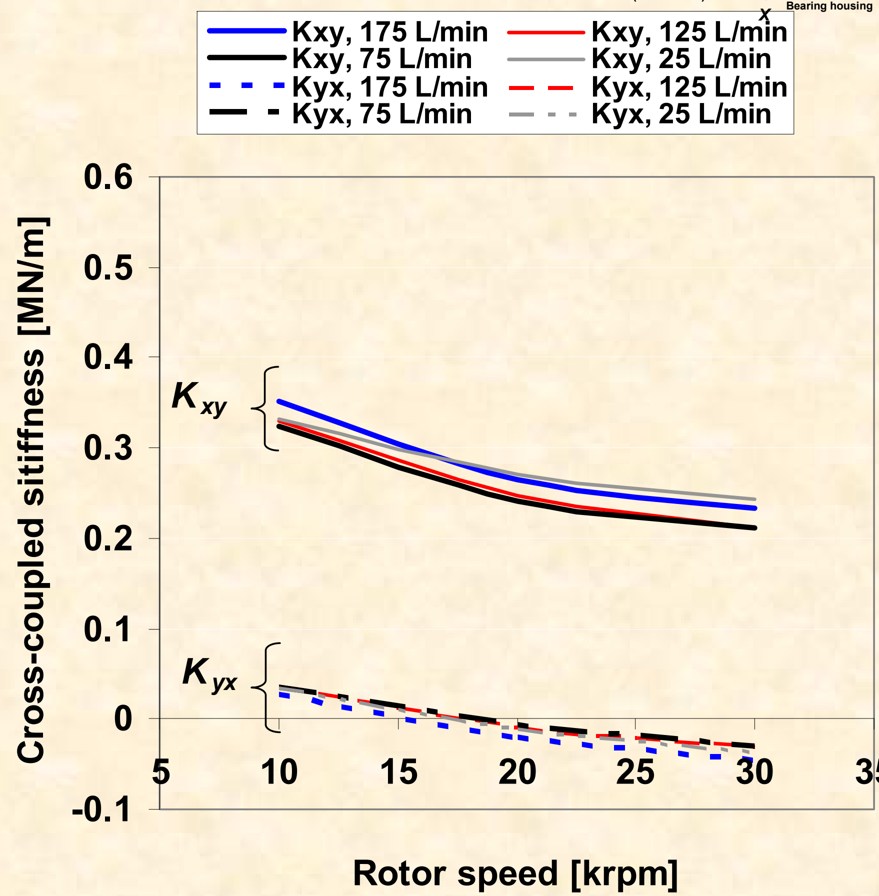
Bearing stiffnesses predictions



Free End FB

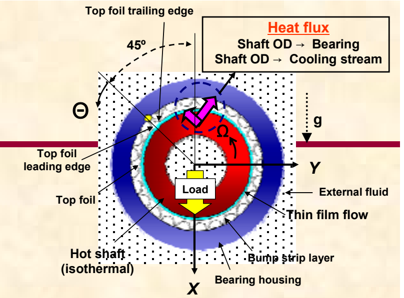
static load 5.94 N





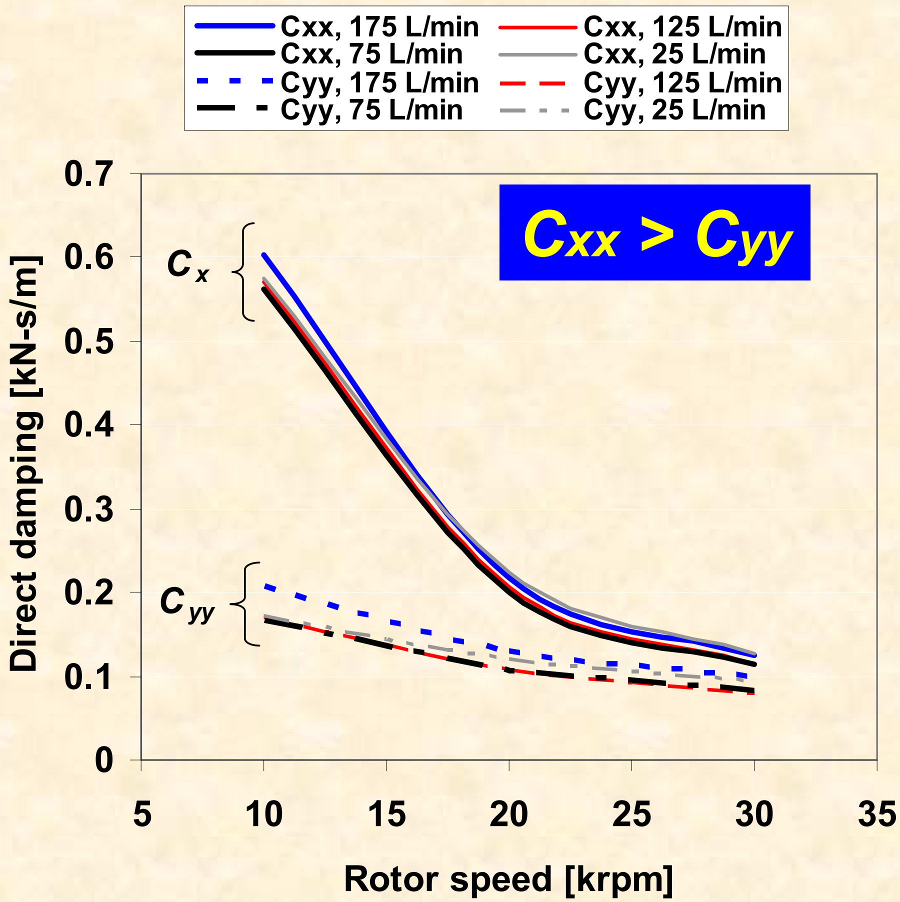
GFB rotorydnamic force coefficients do not change with the strength of cooling flow rate

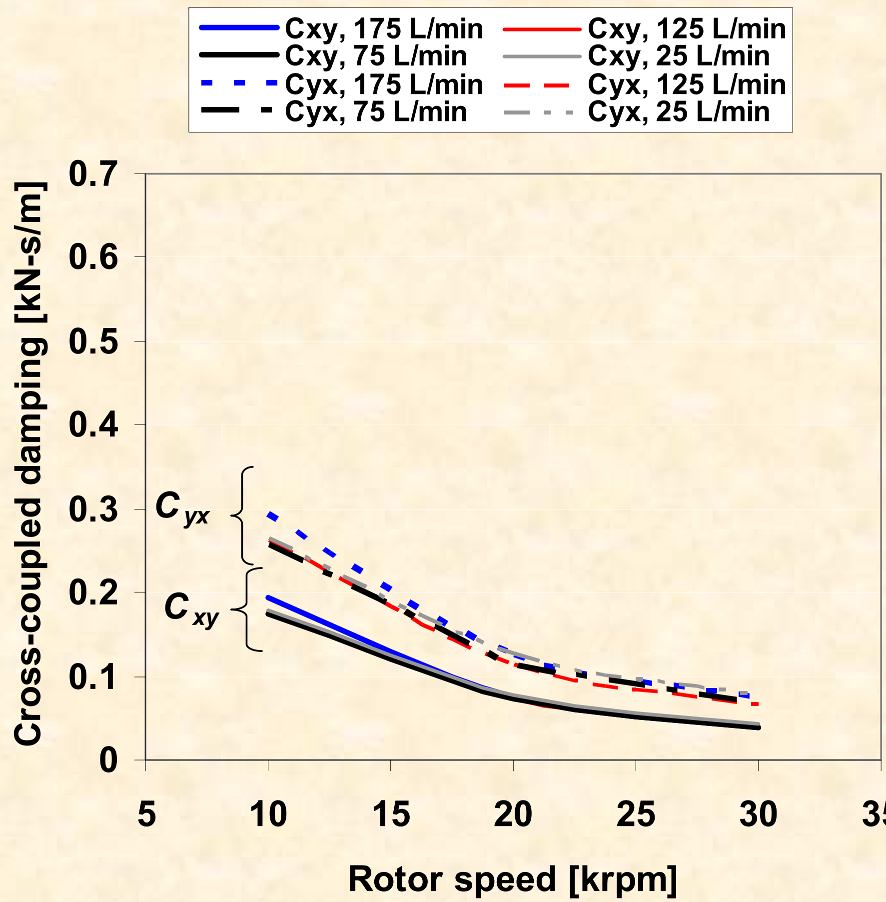
Bearing damping predictions



Free End FB

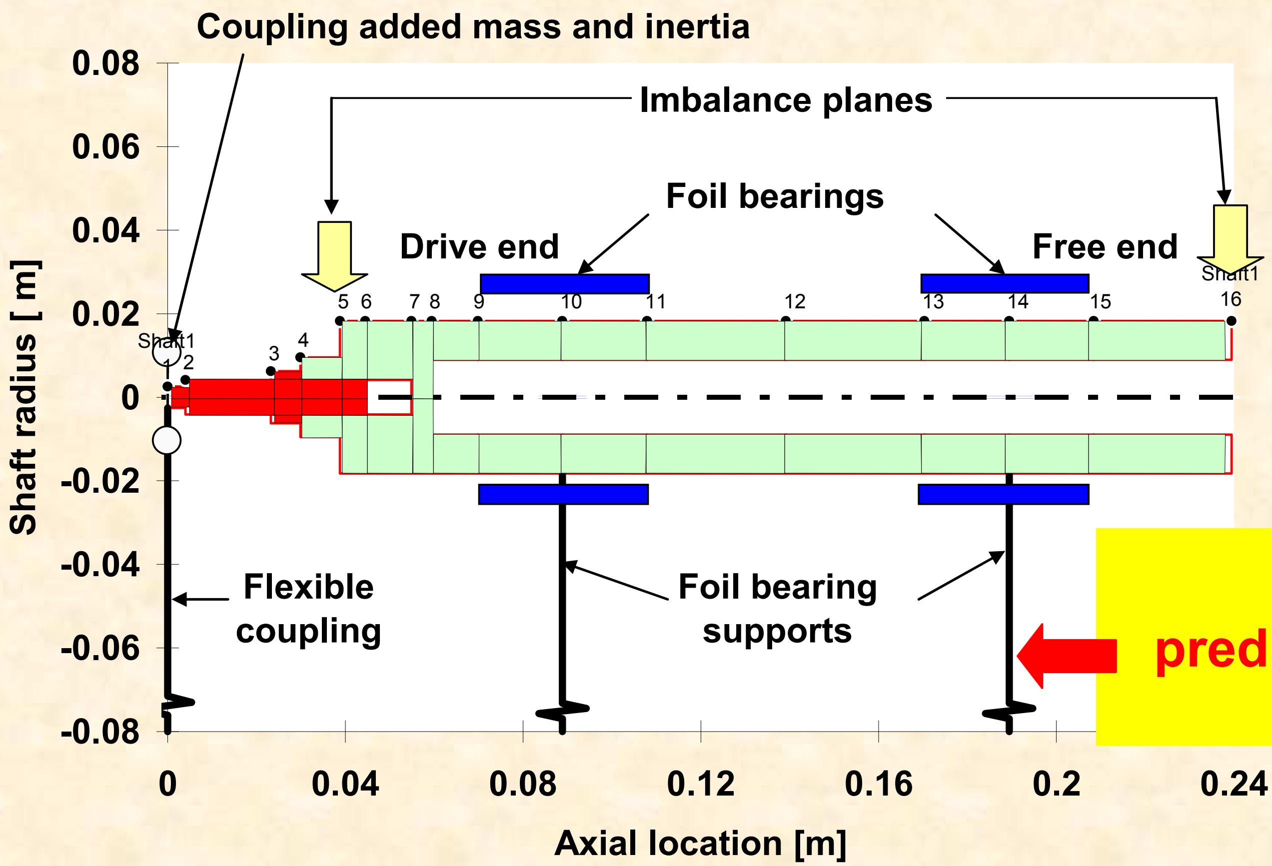
static load 5.94 N





GFB rotorydnamic force coefficients do not change with the strength of cooling flow rate.

FE Model of Test Rotor-Bearing System



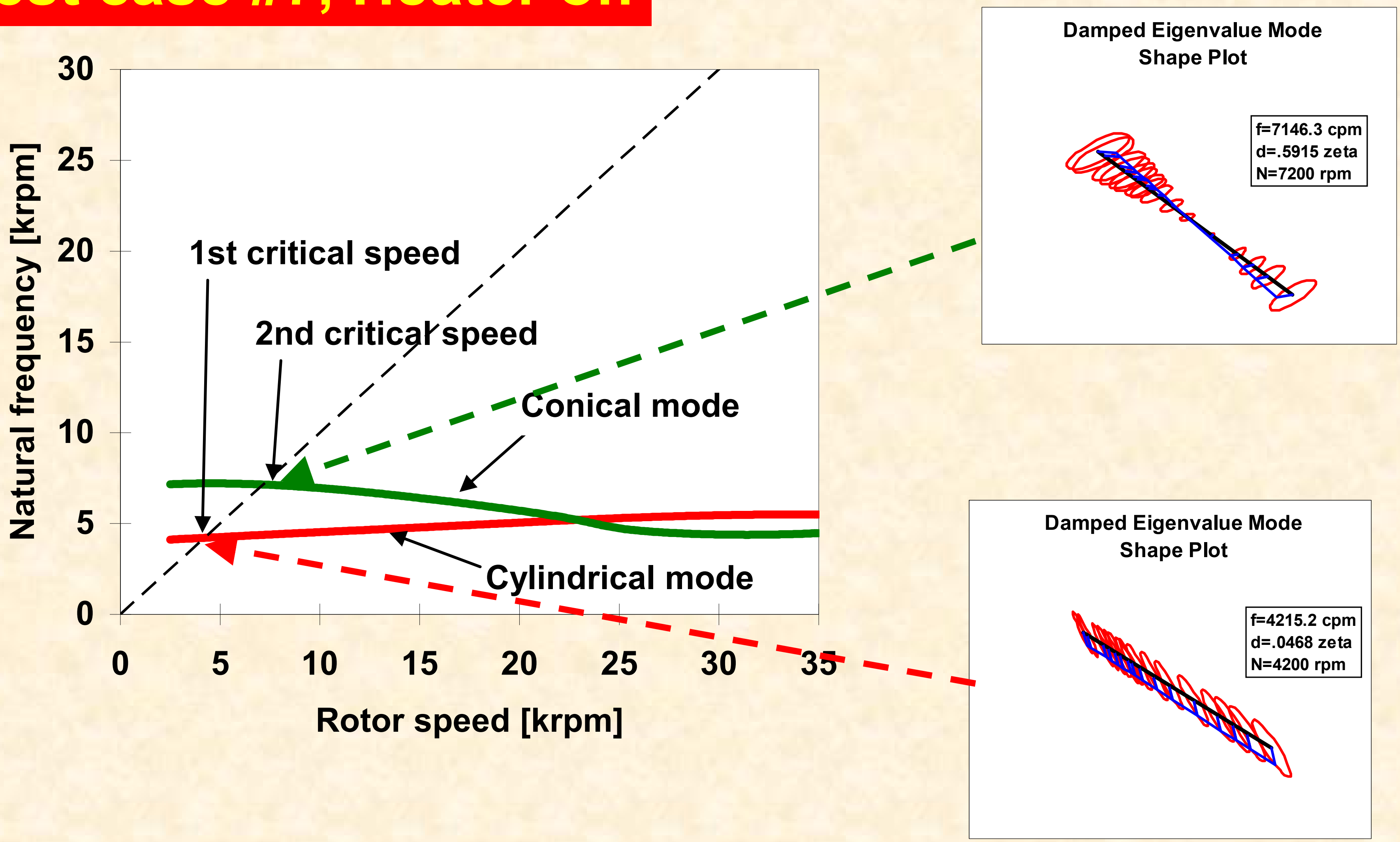
XLGFBTH

predicts synchronous bearing force coefficients

A linear rotordynamics software (XLTRC2®) models test rotor – GFBs system and predicts the rotor synchronous responses

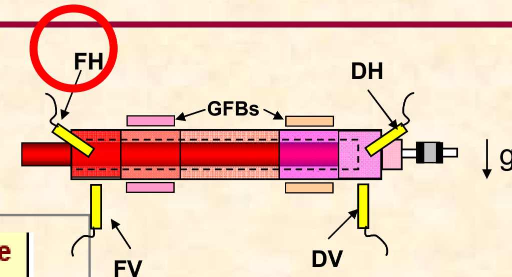
Damped Natural Frequency Map

Test case #7, Heater off

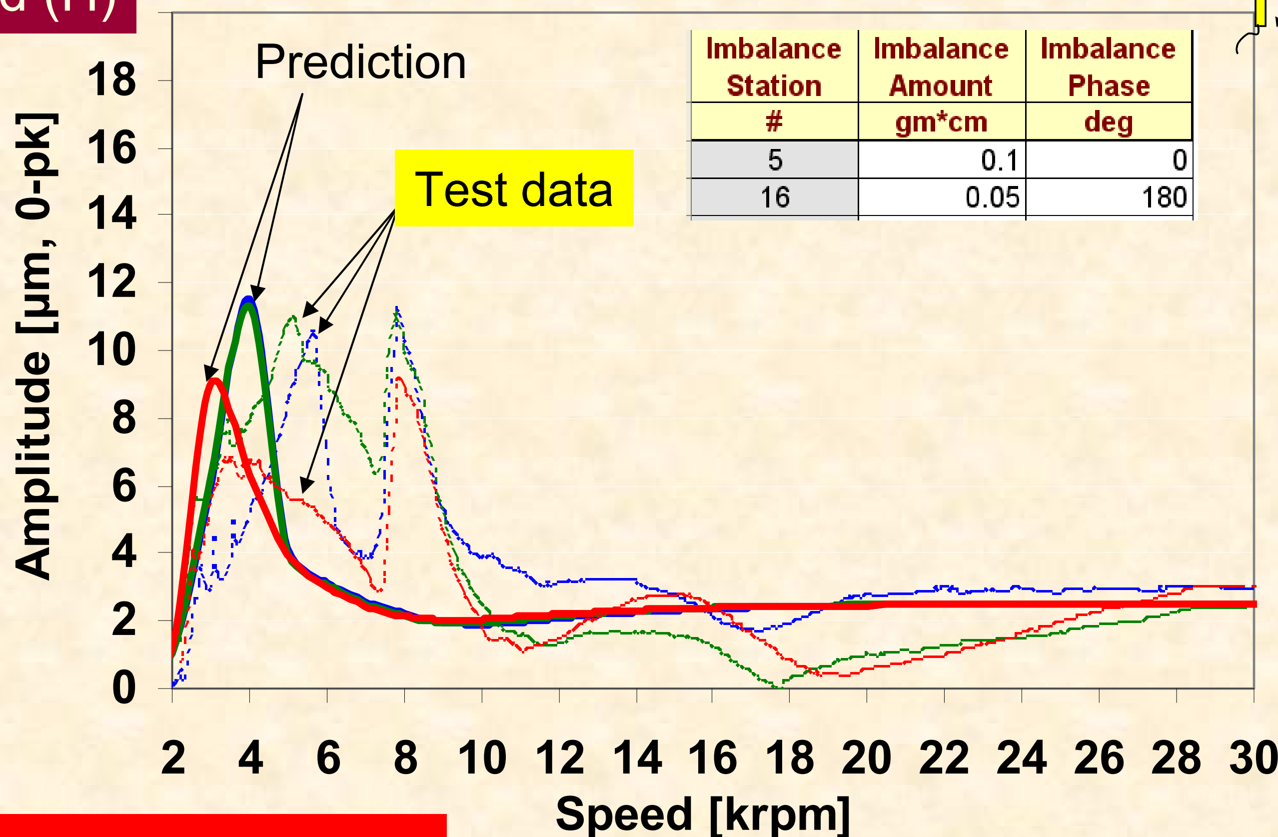


1X rotor response predictions & tests

Forced cooling 175 L/min per bearing



Free End (H)



Imbalance Station #	Imbalance Amount gm*cm	Imbalance Phase deg
5	0.1	0
16	0.05	180

Test cases #7~#9

The predicted rotor responses reasonably correlate with the measurements.

Conclusions

- GFB temperatures linearly increase with the inlet cooling air temperature.

- When the rotor spins, the bearing sleeve temperatures do not change with the cooling flow rate; albeit the rotor OD temperature increases with the strength of the cooling stream,

- The cooling effect of the forced external flows is most distinct when the rotor is hottest and at the highest rotor speed.

- Forced cooling flows do not affect the amplitude and frequency contents of the rotor motions. The test system (rigid-mode) critical speeds and modal damping ratio remain nearly invariant for increasing the rotor temperature and cooling flow strength.

Conclusions

-A physics-based computational THD model predicts accurately measured FB OD temperatures for increasing shaft temperatures with cooling flow

- Rotordynamic analysis integrating predicted FB force coefficients reproduces recorded rotor dynamic responses with increasing cooling flow rate and shaft temperature.

Predictive tool validated & benchmarked to reliable test data base !!!

Major contribution

The present work provides the most complete to date measurements of GFB temperatures and rotordynamic response thereby extending the GFB knowledge database. Comprehensive experiments and benchmarking of predictive tool serve to advance GFB applications for use into high temperature microturbomachinery.

Topic

- Statement of Work & Sources for Presentation
- Objectives and accomplished work in 07-08
  - Computational model. Validation with published data.
  - Rotordynamic measurements at TAMU
- Objectives and accomplished work in 2008-09
  - Description of test rig and foil bearings at TAMU
  - Effect of temperature on bearing temperatures, coastdown speed and rotor motions
  - Effect of cooling flow on bearing and shaft temperatures. Validation of computational model
- The computational code
  - Graphical User Interface. Further predictions
- Current work with MiTi Bearings
- **Added value & closure**

NSF-Research Undergraduate Experience in Microturbomachinery & Manufacturing

To conduct hands-on training and research in mechanical, manufacturing, industrial, or materials engineering topics related to technological advances in microturbomachinery

To develop microturbines to enhance defense, homeland security, transportation, and aerospace applications.

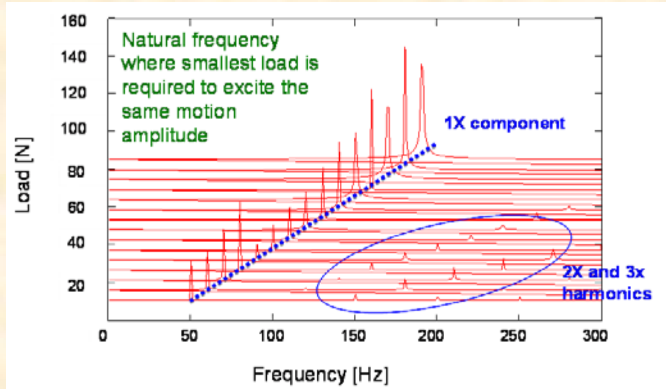
(10 students /year) x 3 y **NSF (06-09) $ 259 k**

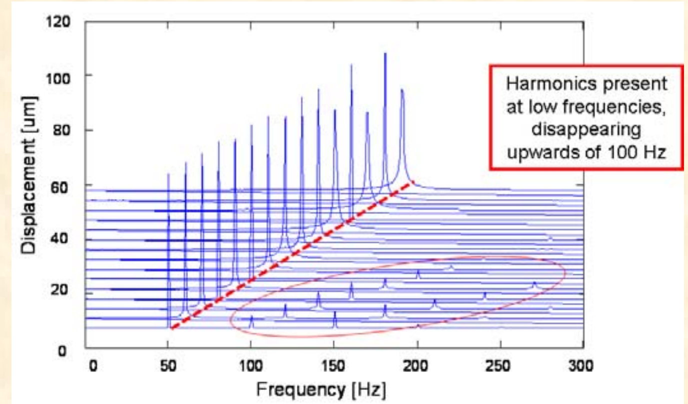
Added value to NASA Project

2009 REU MTM Program

Project #1	Experimental Identification of Structural Stiffness and Damping in a 1st Generation Gas Foil Bearing for Oil-free MTM		
REU student	Shane Muller	Mechanical Engineering	Calvin College

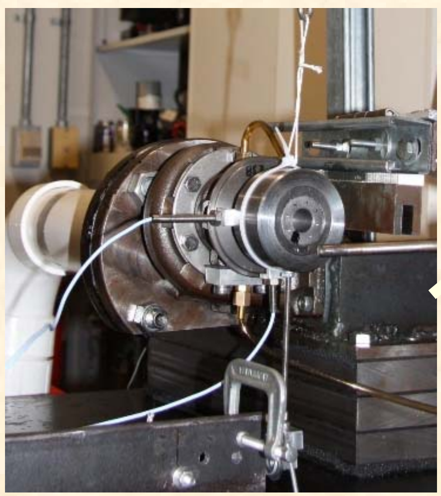
***KIST* FB (1st generation)**

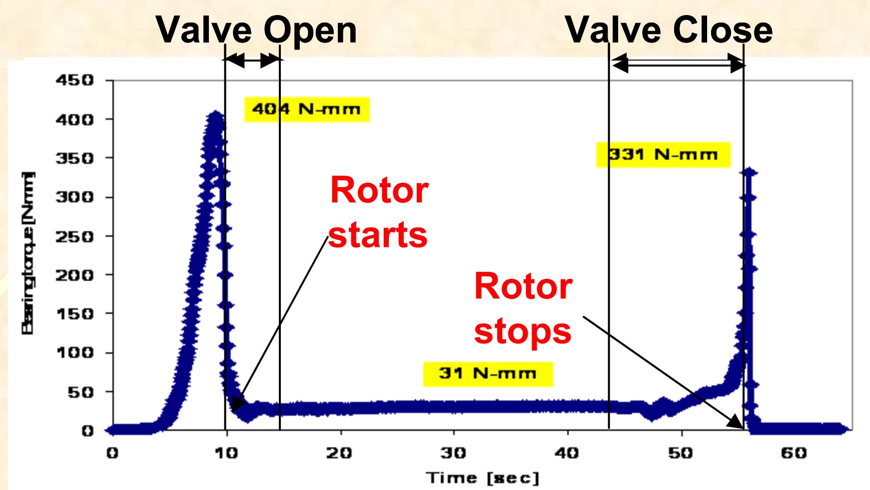




Project #2	Measurement of Drag Torque, Power Loss, Friction Coefficient, Temperature, and Wear in a Foil Bearing & Coated Rotor System		
REU student	Jose Camero	Mechanical Engineering	U. of Texas, San Antonio

***KIST* FB**





Closure: objectives accomplished

- To develop a physics-based computational model of GFB including thermal effects

- To develop a fully tested and experimentally verified design tool for predicting GFB performance

- To measure the rotordynamic performance of a HOT rotor supported on GFBs

- To quantify the effect of feed gas flow on cooling GFBs

Predictive tool validated & benchmarked to reliable test data base !!!

Acknowledgments

- NASA GRC: Drs. S. Howard & Dr. C. DellaCorte
- Turbomachinery Research Consortium
- NSF REUP
- Capstone Turbine, Inc.
- MiTi©, Foster-Miller
- KIST: Korea Inst. Science & Technology
- Honeywell Turbocharging Technologies

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

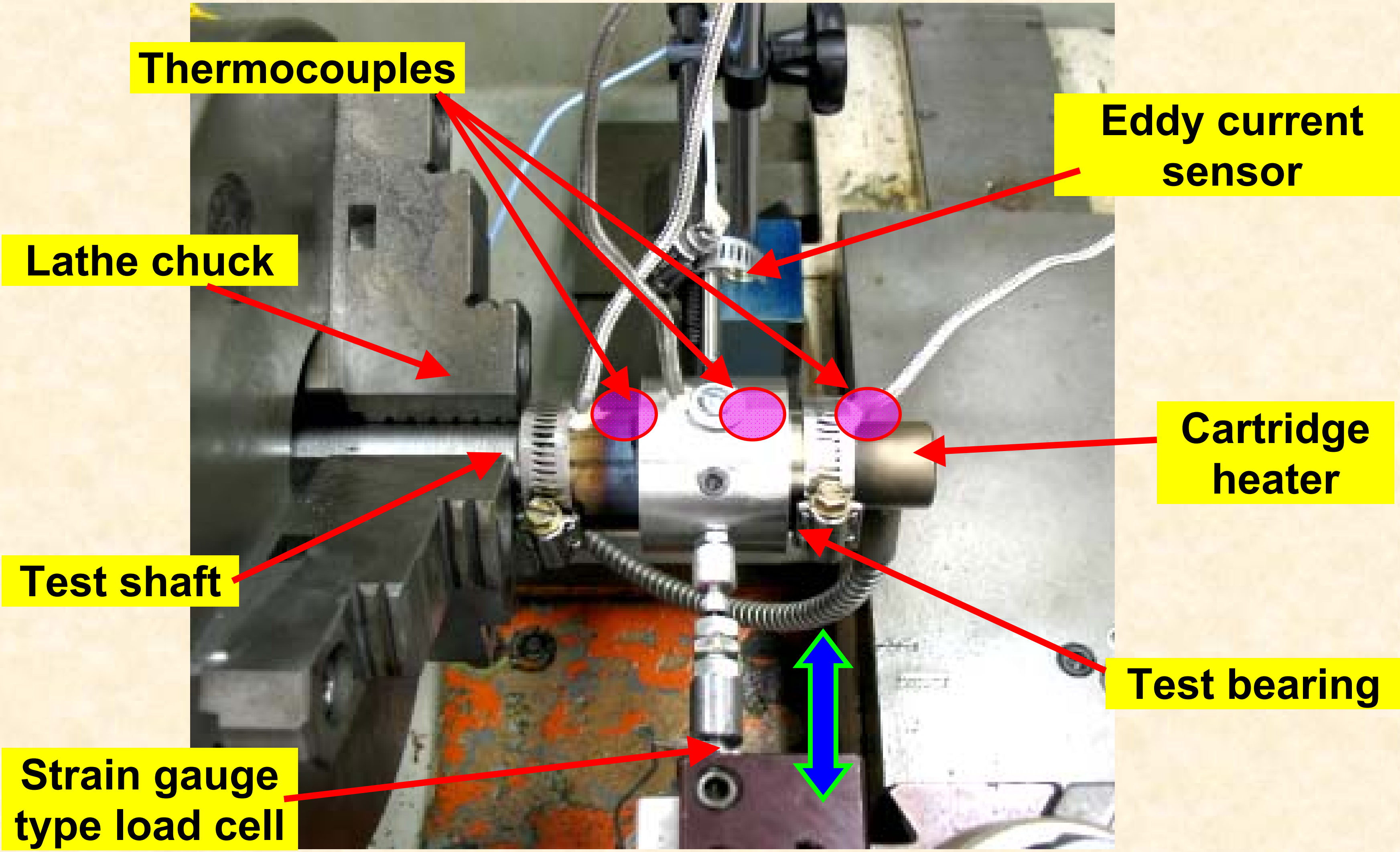
Back up slides

THD Model Validation Bearings at TAMU

Parameter [mm]	Foster-Miller (2nd gen.)	KIST (1st gen.)	MiTi (2nd gen.)
Bearing cartridge			
Outer diameter	50.85	50.80	44.575
Inner diameter	39.36	37.95	37.921
Top foil and bump strip layer			
Top foil axial length	38.2	38.1	25.4
Top foil thickness	0.100	0.120	0.127
Bump foil thickness	0.100	0.120	0.102
Number of Bumps	25 × 5 axial	26 × 1 axial	24 × 3 axial
Bump pitch	4.581	4.300	4.640
Bump length	3.742	2.100	3.950
Bump height	0.468	0.540	0.510
Bump arc radius	5.581	4.161	4.079
Bump arc angle [deg]	68	59	58

Elastic Modulus 214 GPa,
Poisson ratio=0.29

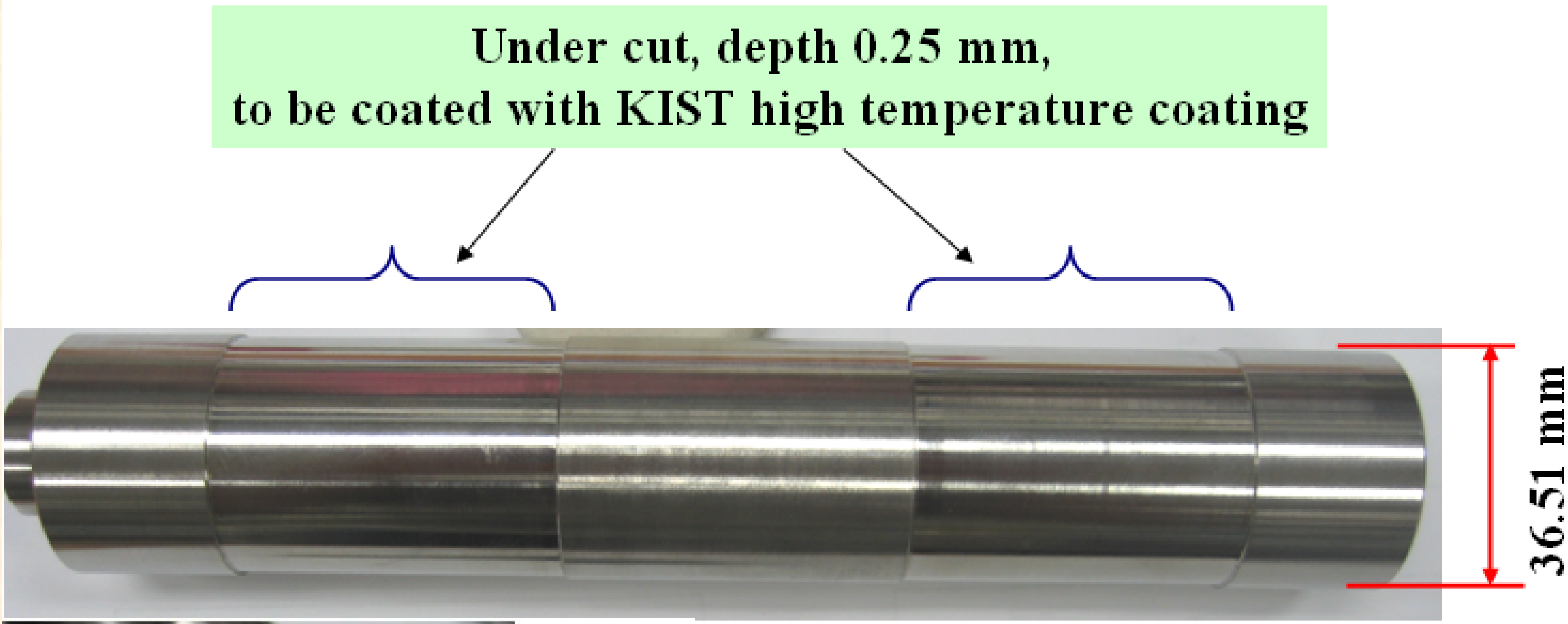
Static load test setup

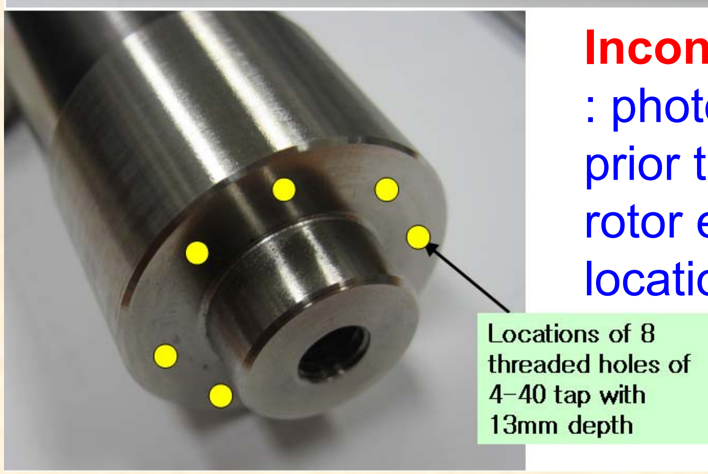


Steady static load (or unload) proportional to linear movement of lathe tool holder

High temperature rotor

NO COST!





Inconel 718 shaft
: photos taken by manufacturer (KIST) prior to machining of threaded holes at rotor ends and coating shaft at bearing locations.

KIST **proprietary solid lubricant (400 ºC)**

Closure Y2

- **Assess effects of temperature (to 160 C) on the structural properties of FB from static load (250 N) tests:**

  Loading and unloading tests show hardening nonlinearity and mechanical hysteresis.

  FB structural stiffness reduces with temperature due to increase in bearing radial clearance (atypical).

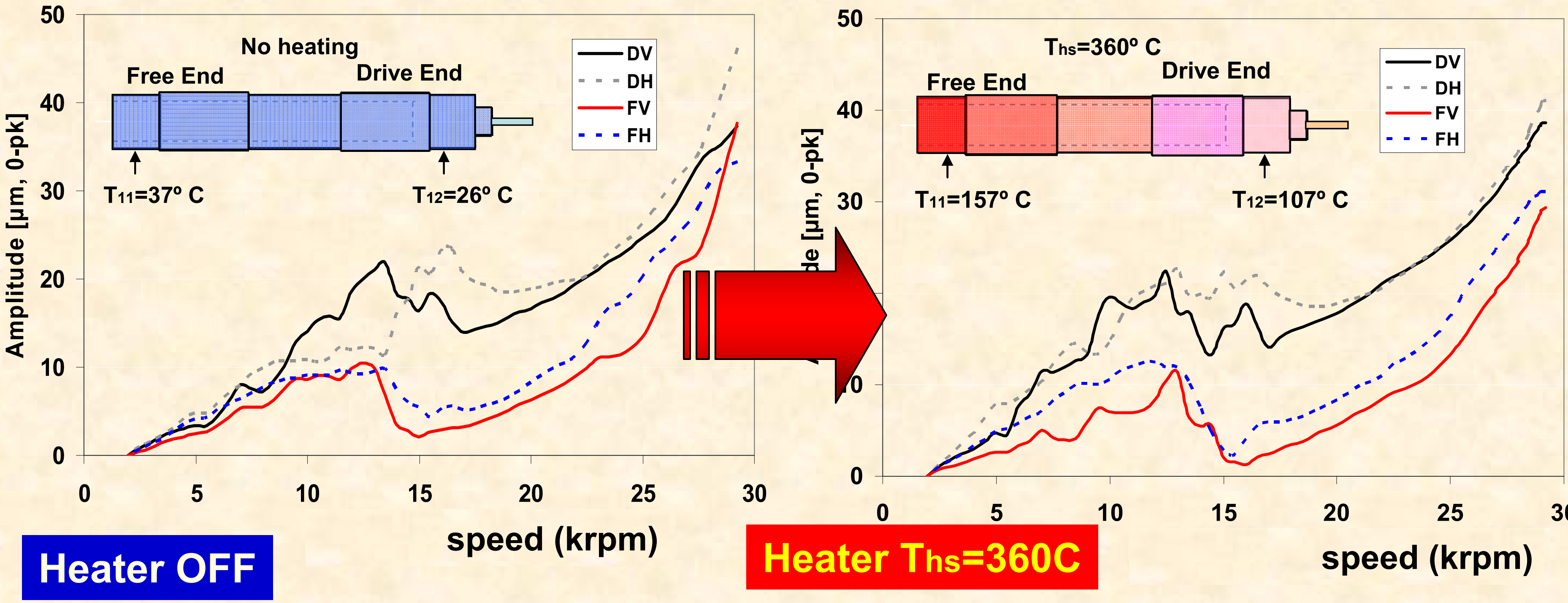
  Model predictions reproduce test data, when accounting for thermal effects in materials properties and components' expansion.

- **ADDED VALUE:**

08 & 09 Summer NSF-REU in microturbomachinery educated six undergraduate students (US citizens).

1X response with cold and hot rotor

Baseline coastdown, No forced cooling



Elastic rotor mode at 29 krpm (480 Hz) : soft coupling and connecting rod

Critical speed (rigid body mode) ~ 13 krpm

Test Data

Similar rotor responses for cold and hot rotor operation