Turbomachinery Symposium – Case Study

Practical root cause analysis of connecting rod bushing failures in a new reciprocating compressor and the theory behind the failure mechanism

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Case Study Overview

- During start-up of a new reciprocating compressor, multiple connecting rod bushing failures led to a detailed root cause analysis, data gathering, and testing

- This compressor met API 618 requirements for rod load and rod reversal.

- Another compressor of the same design with similar rod load has history of reliable operation

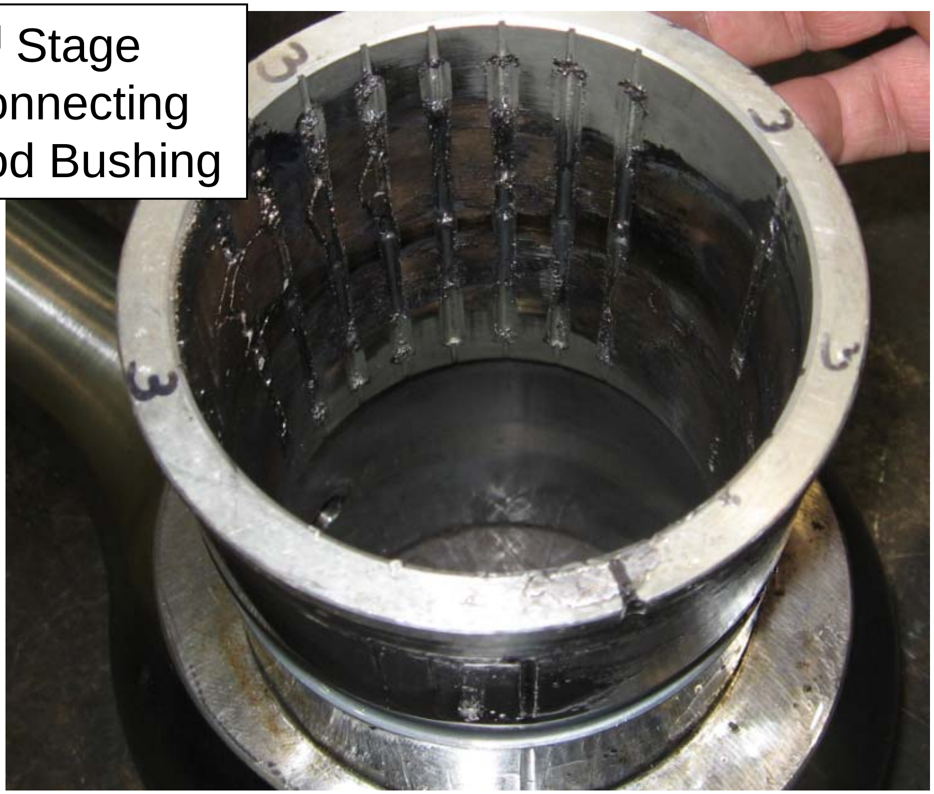
- Applying practical root cause analysis and well known engineering principles, a simple solution was found

- To further the analysis of connecting rod bushing lubrication mechanisms, the OEM has created a new software program to model this complex lubrication application

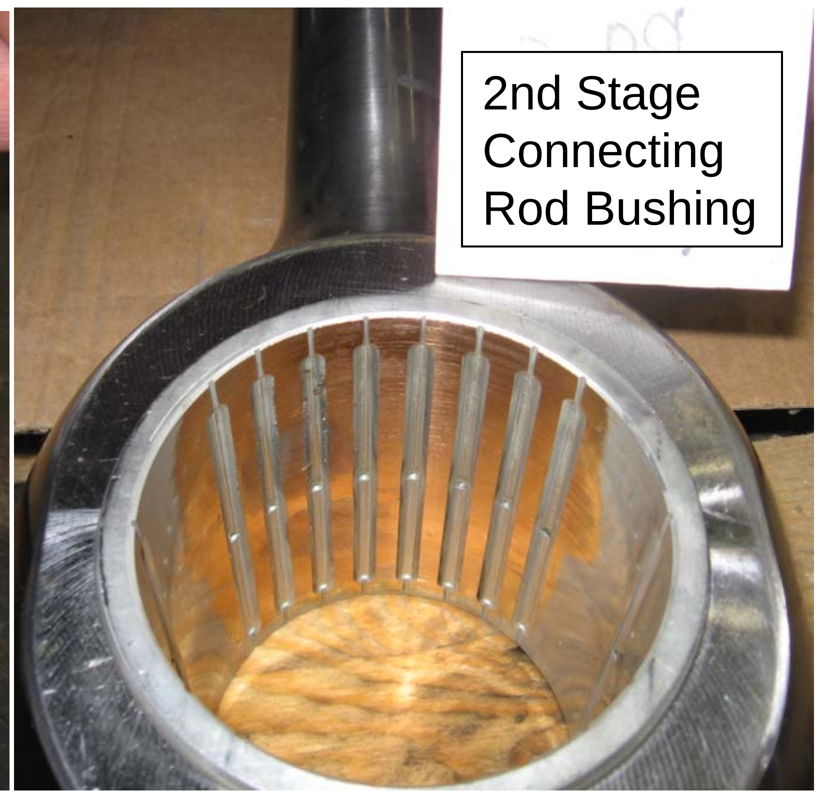
Failed Connecting Rod Bushings

- Compressor Application Data
  - Hydrogen Make-up Compressor
  - 4-Throw ; 3-Stage (1st Stg Suction 220 psig ; 3rd Stg Discharge 2,114 psig)
  - 1,600 Horsepower
  - 8.4 MMSCFD
  - 441 RPM

3rd Stage Connecting Rod Bushing



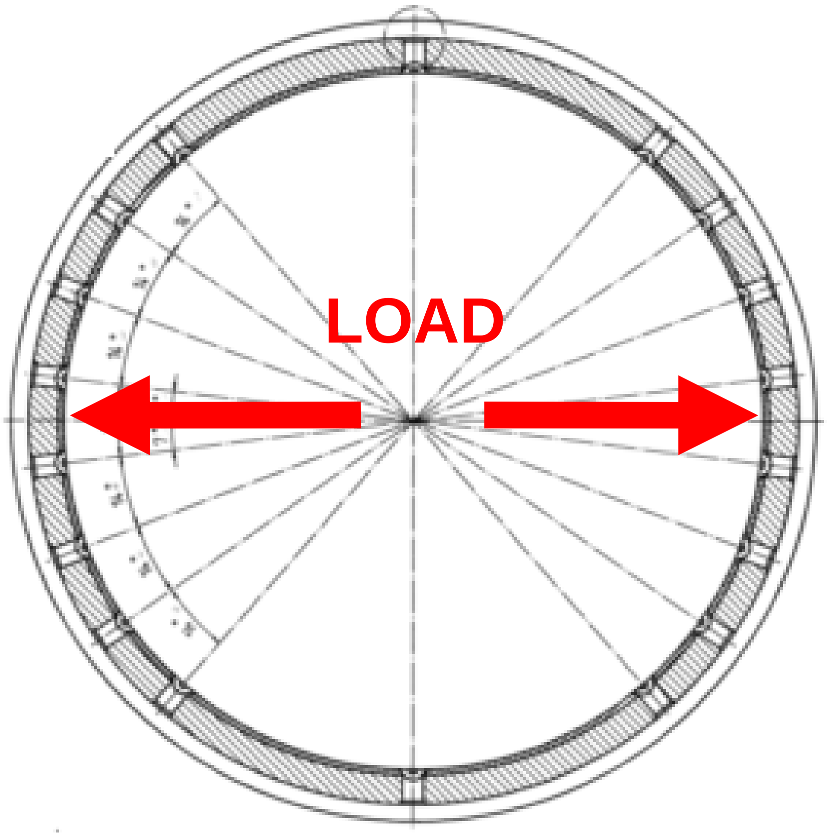
2nd Stage Connecting Rod Bushing



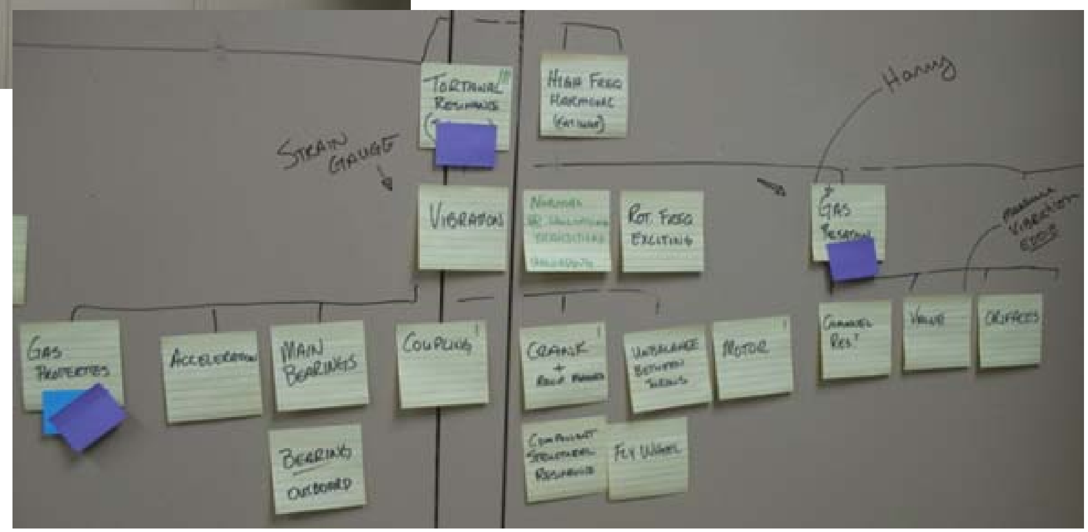
Timeline

October 2006	1st Failure
November 2006	2nd Failure
December 2006	▪ PV Analysis to confirm operating conditions ▪ 3rd Failure occurred during testing ▪ Detailed Root Cause Analysis performed
January 2007	▪ Performed Strain Gage Testing / PV Analysis / Operating Deflection Shape (ODS) Analysis ▪ 4th Failure occurred during testing ▪ Implemented solutions and had successful 10 minute and 4 hour test runs
February 2007	Inspection after 1 month run. Bushings in good condition
June 2008	Inspection after 15 month run. Bushings in good condition

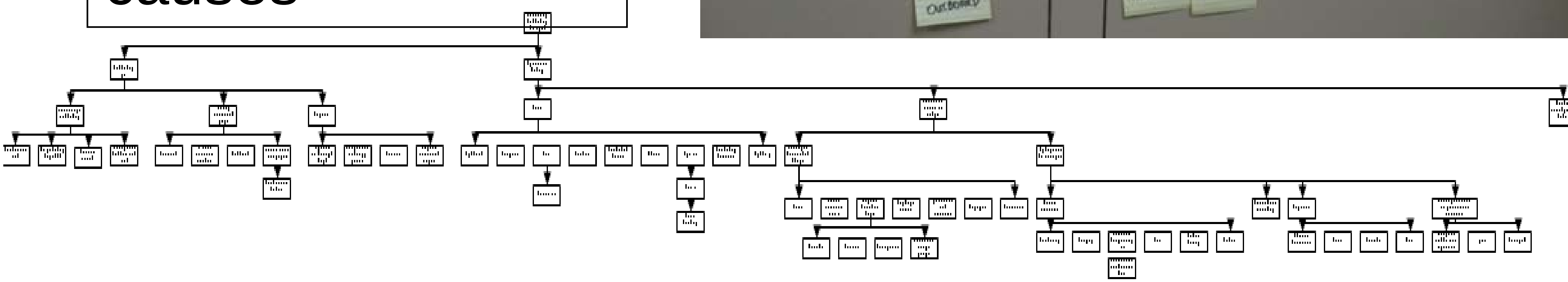
Connecting Rod Bushing



Root Cause Analysis



Identify all potential failure modes and causes

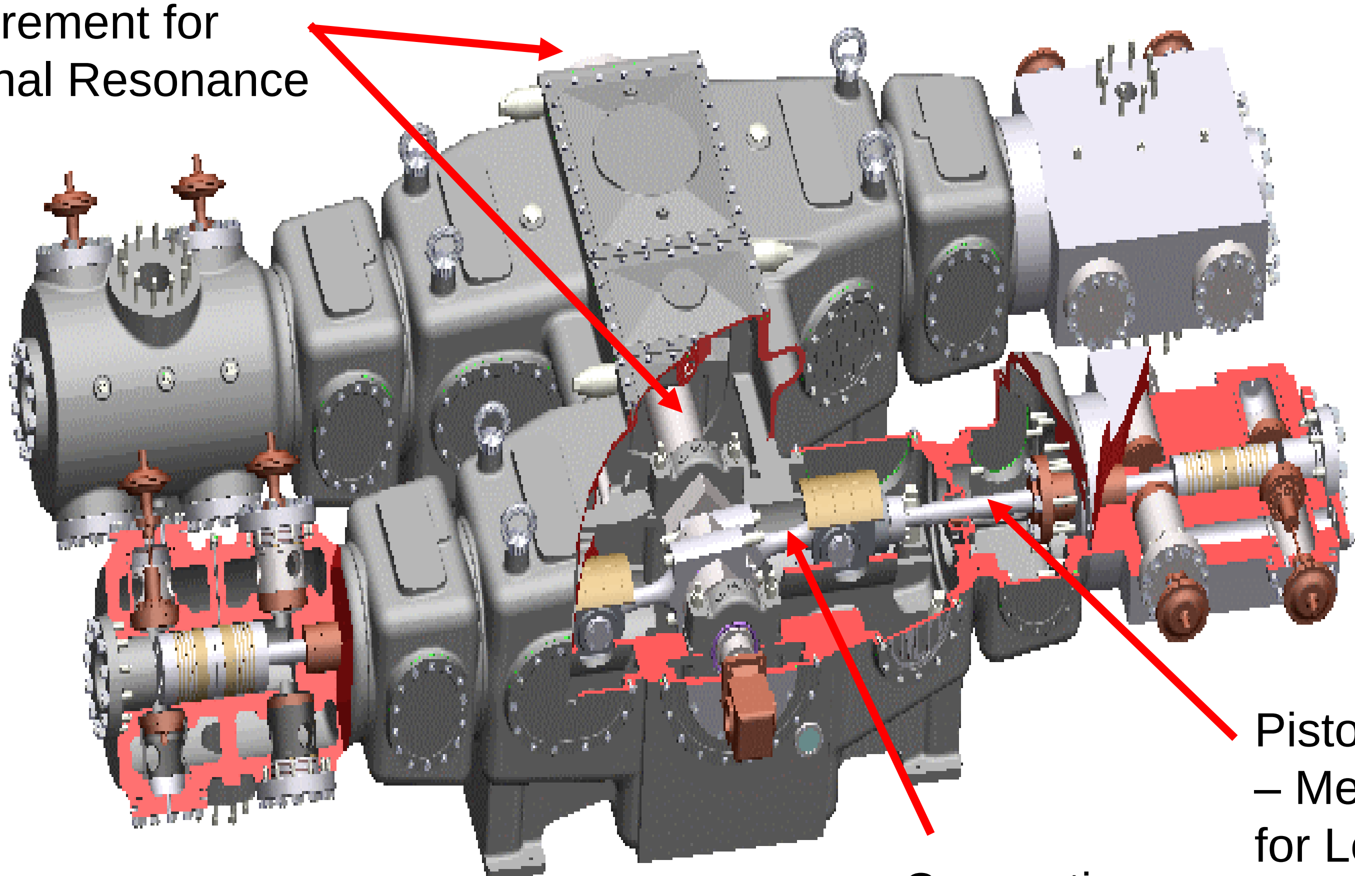


RCA identifies further testing requirements

- PV Analysis and Strain Gage Analysis to measure / confirm rod loads and check for torsional resonance
- Operating Deflection Shape (ODS) to identify any structural resonance
- Confirm actual oil flow

Strain Gage Measurements

Crankshaft (2) – Measurement for Torsional Resonance

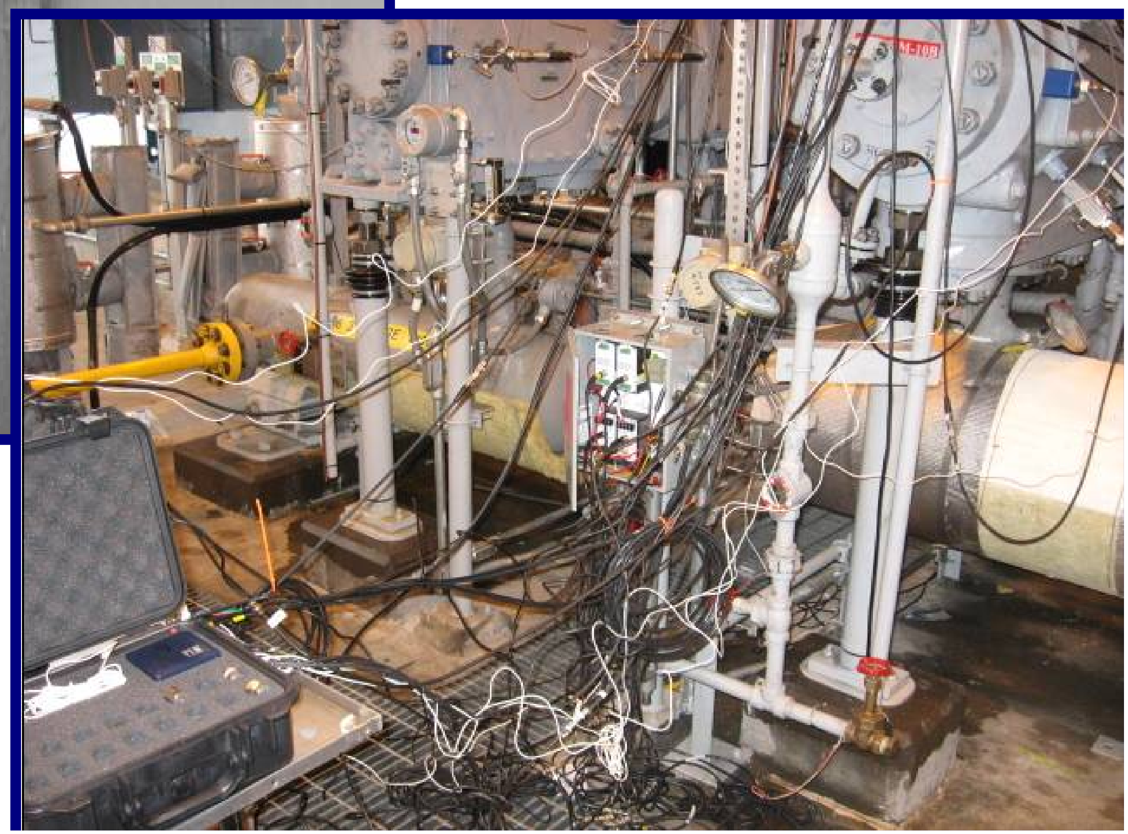


Connecting Rod (3rd Stg) – Measurement for Load

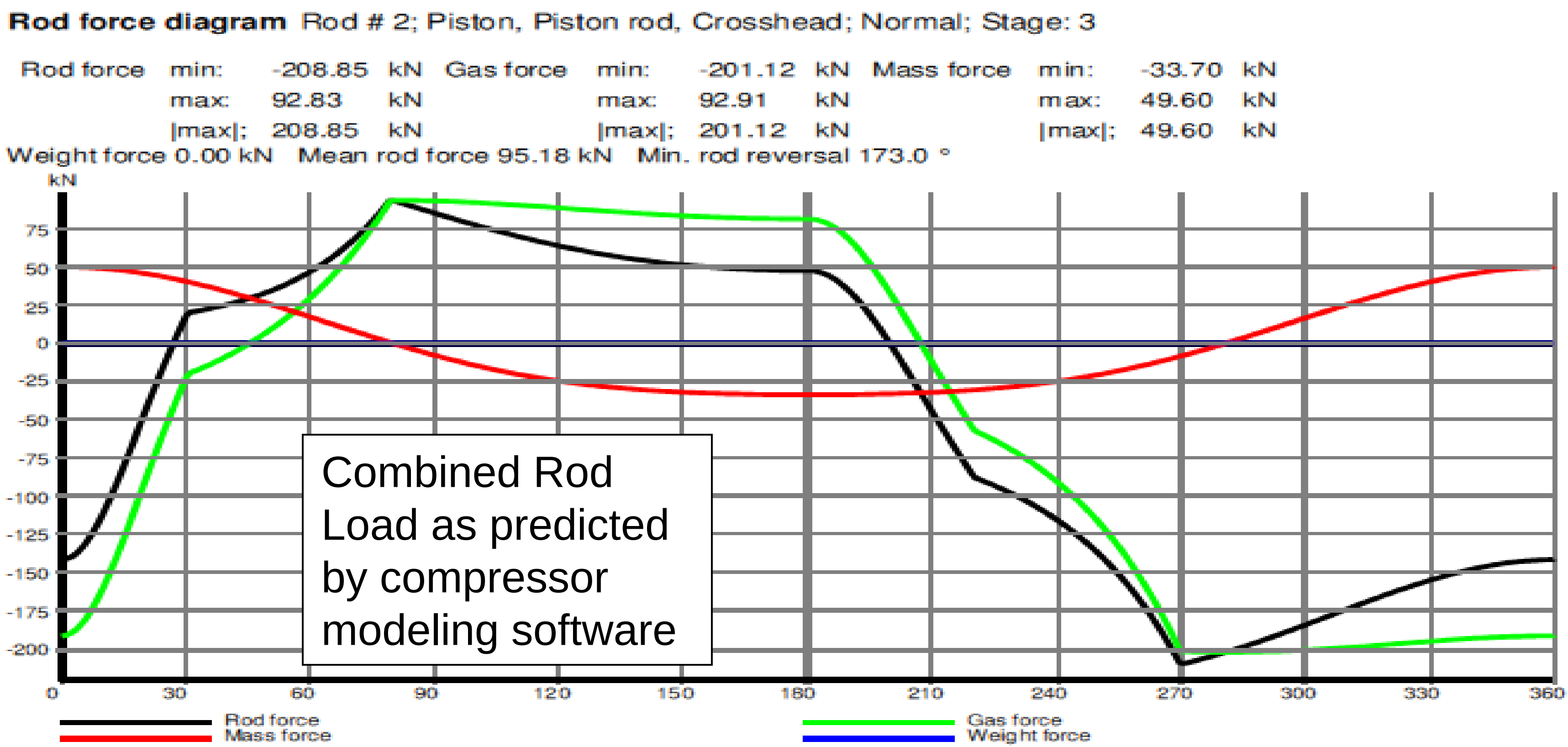
Piston Rods (All) – Measurement for Load and Bending

Data Acquisition

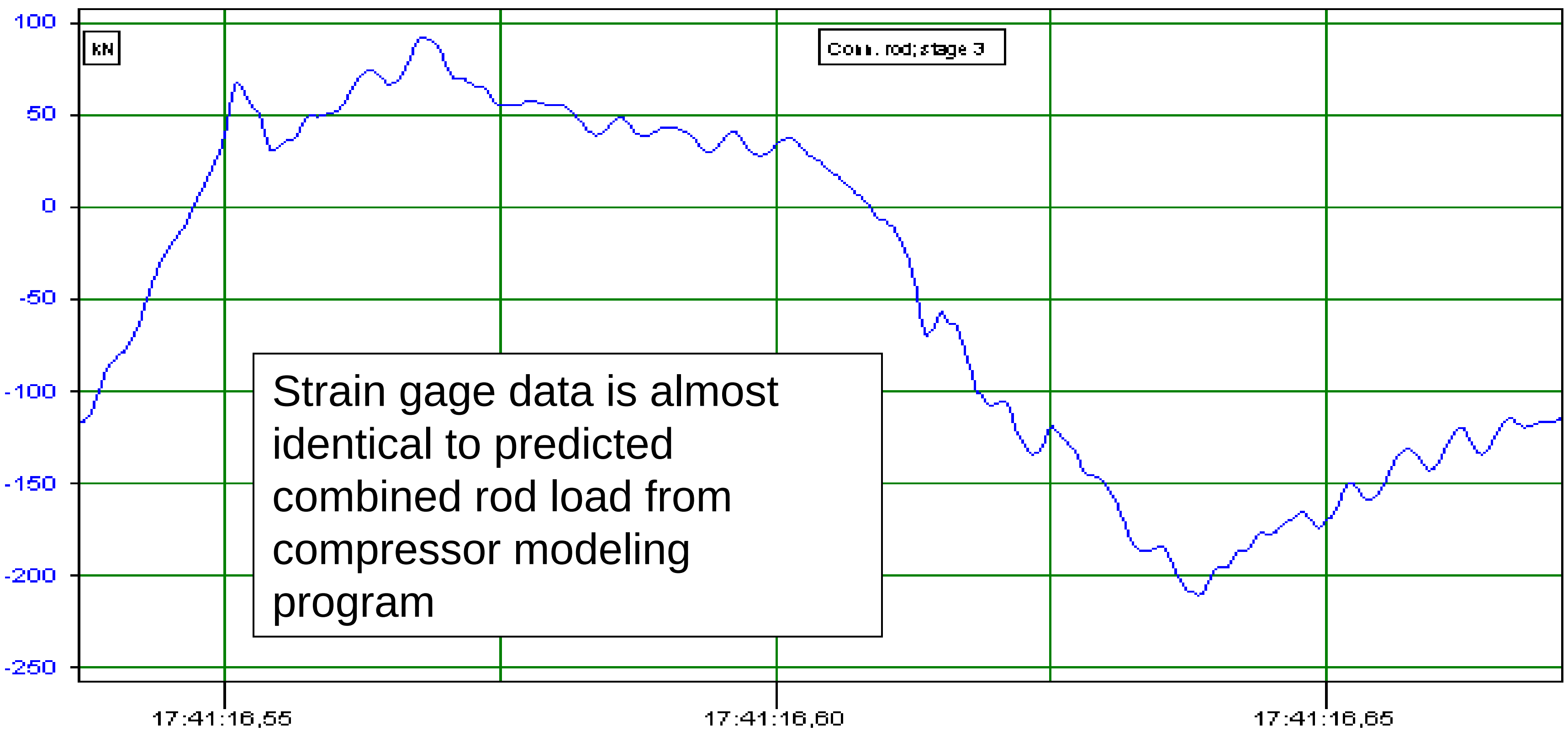
Data Acquisition



Predicted Combined Rod Load Diagram for 3rd Stage



Strain Gage Data – 3rd Stage Connecting Rod



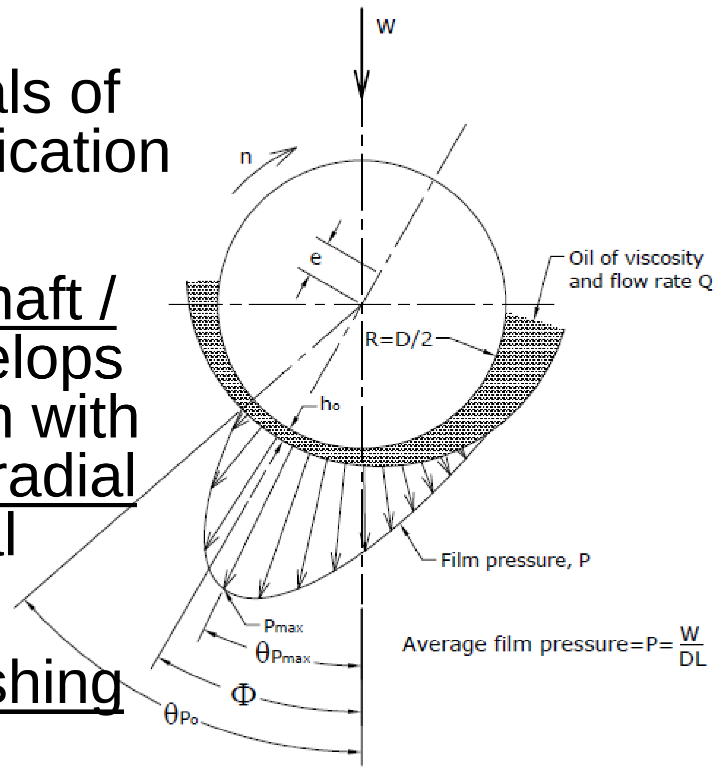
Strain gage data is almost identical to predicted combined rod load from compressor modeling program

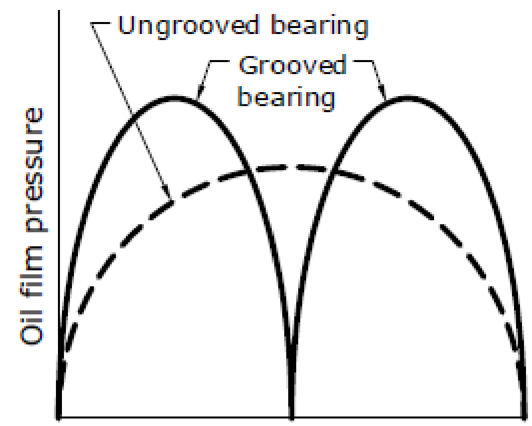
Results of Data Analysis / Conclusions

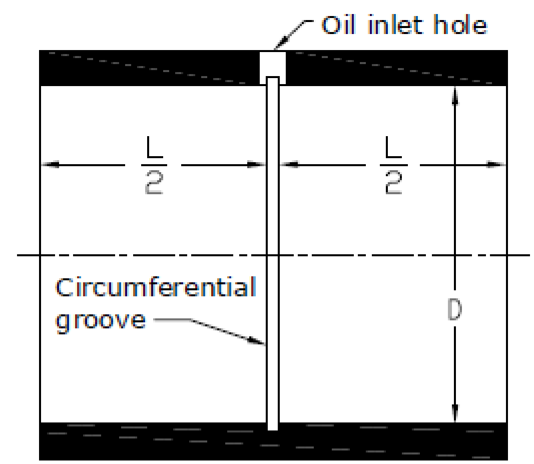
- Measured combined rod load was similar to predicted – Ruled out off-design operation
- Ruled out Torsional Resonance
- Ruled out Structural Resonance
- Problem was with insufficient load capacity in bushing due to lack of oil film thickness
  - Caused by Bushing Geometry
    - Load Surface Area too small
    - Hydrodynamic Pressure created in oil film was excessive and oil film was not maintained
  - Oil Viscosity and Type
    - Need oil with better film strength

Classic hydrodynamic oil film pressure distribution

- Applied fundamentals of Hydrodynamic Lubrication
- Standard rotating shaft / sleeve bearing develops pressure distribution with both rotational and radial movement of journal
- Crosshead pin / bushing develops pressure distribution with only radial movement
- Position of grooves changes pressure profile

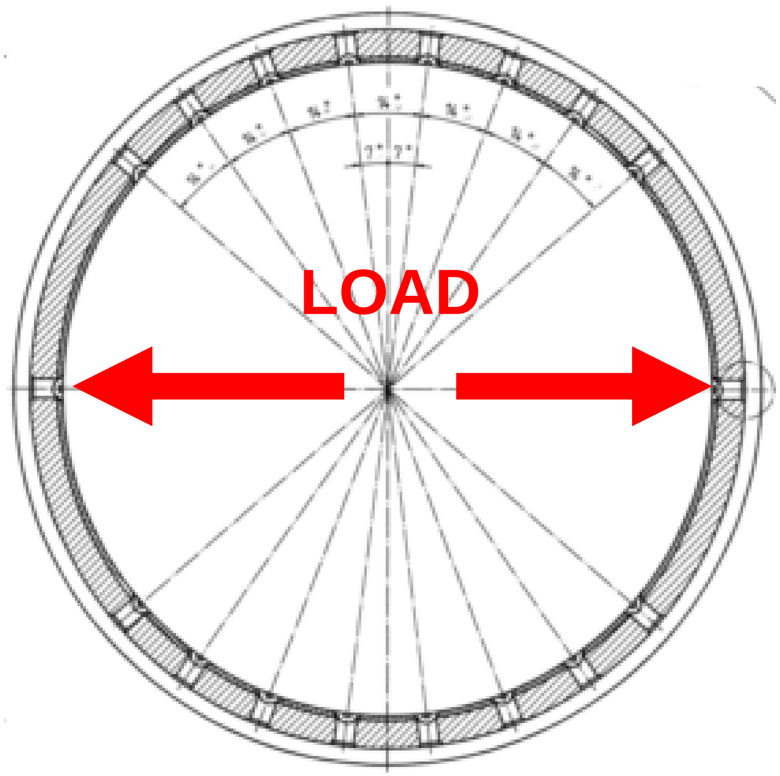






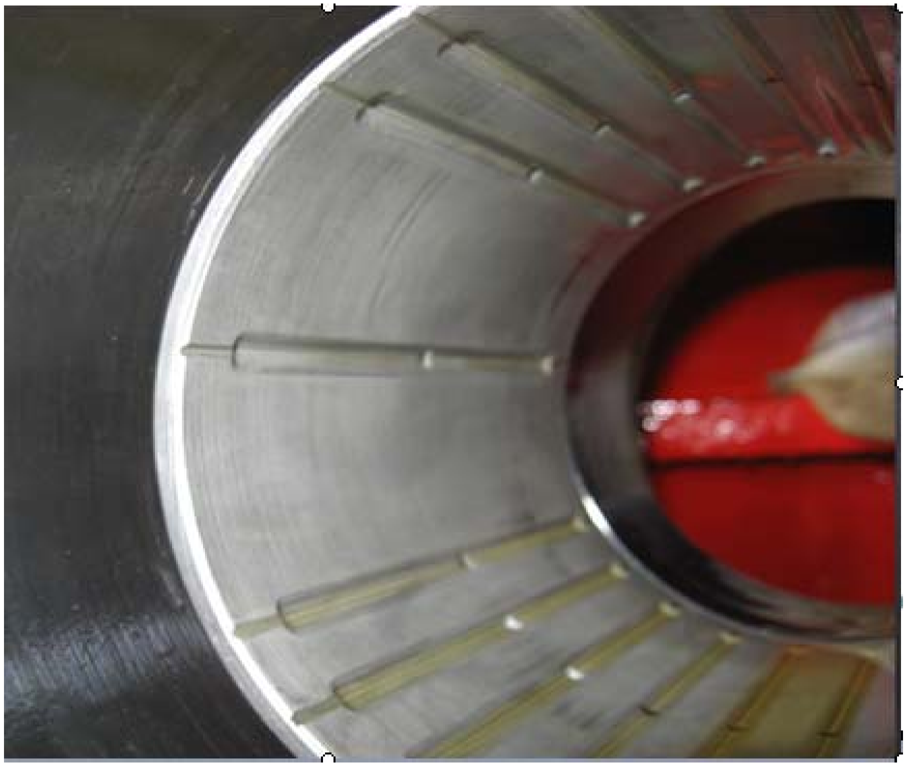
Design of Experiment / Corrective Actions

- Corrective Actions to increase the lube oil film thickness
  - Increased Bushing Load Capacity
    - Rotated Bushing 90 degrees to get more bushing surface area in the load zone i.e. change the hydrodynamic pressure profile in the bushing
  - Changed Lube Oil
    - Changed from Mineral Oil to Synthetic Oil
    - Changed from ISO 100 to ISO 150 Viscosity Grade

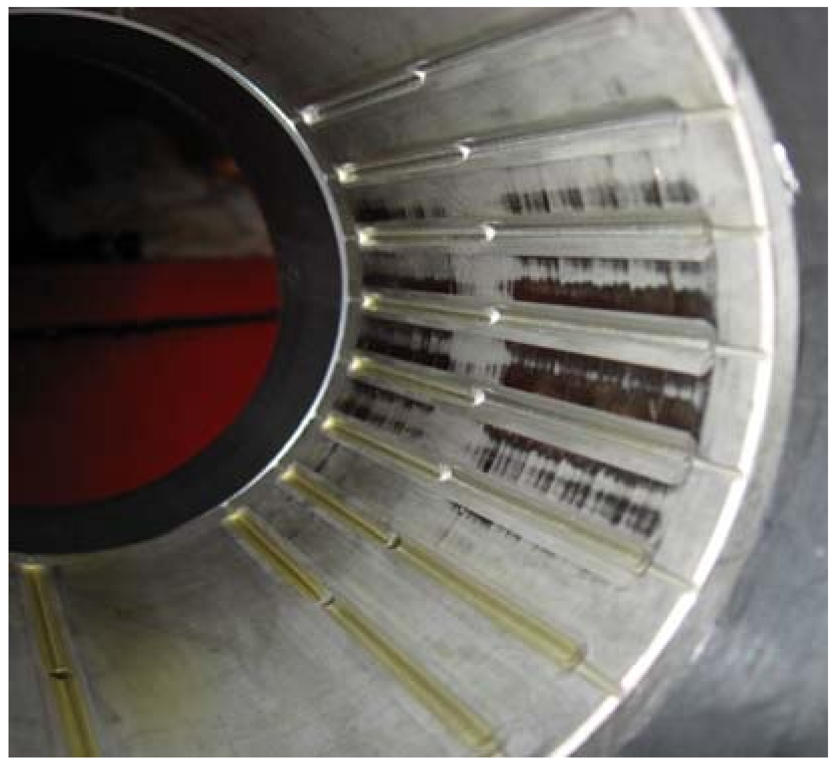


Test Run Results: Rotated Bushing on 3rd Stage and Higher Viscosity Oil

<u>3rd Stage</u>: Rotated bushing and change in oil viscosity - **<u>No damage</u>**



<u>2nd Stage</u>: Change in oil viscosity alone – **<u>Less damage</u>**



Development of Modeling Software

- Classical hydrodynamic analysis methods are not applicable for crosshead pin bushings
  - For this type of bushing, oil supply grooves are often arranged in the highly loaded area to ensure supply to bushing surfaces
  - Also, since there is no real rotary movement of the journal, the oil in the bushing is not continuously replenished
  - This is especially true for load scenarios with less rod load reversal

Development of Modeling Software

Reynolds' Differential Equation for the Crosshead Pin Bearing

- Defining dimensionless Sommerfeld number $\mathrm{So}(\beta, b, t) = \dfrac{\mathrm{p}(\beta, b, t)\,\psi^2}{\eta\,\omega}$

$$3\,(1-\varepsilon(t)\cos(\beta-\delta(t)))^2\left(\frac{\partial}{\partial\beta}\mathrm{So}(\beta, b, t)\right)\varepsilon(t)\sin(\beta-\delta(t))$$
$$+(1-\varepsilon(t)\cos(\beta-\delta(t)))^3\left(\frac{\partial^2}{\partial\beta^2}\mathrm{So}(\beta, b, t)\right)$$
$$+\left[2\,\frac{r}{B}\right]^2(1-\varepsilon(t)\cos(\beta-\delta(t)))^3\left(\frac{\partial^2}{\partial b^2}\mathrm{So}(\beta, b, t)\right)=$$
$$-12\,\frac{\varepsilon(t)\sin(\beta-\delta(t))\left(\frac{\partial}{\partial t}\delta(t)\right)}{\omega}+\frac{6\,\lambda\cos(\omega\,t)\,\varepsilon(t)\sin(\beta-\delta(t))}{\sqrt{1-\lambda^2\sin(\omega\,t)^2}}$$
$$-\frac{12\left(\frac{\partial}{\partial t}\varepsilon(t)\right)\cos(\beta-\delta(t))}{\omega}$$

  - Dimensionless axial coordinate b=2z/B
  - Axial width of bearing B

Development of Modeling Software

Boundary Conditions

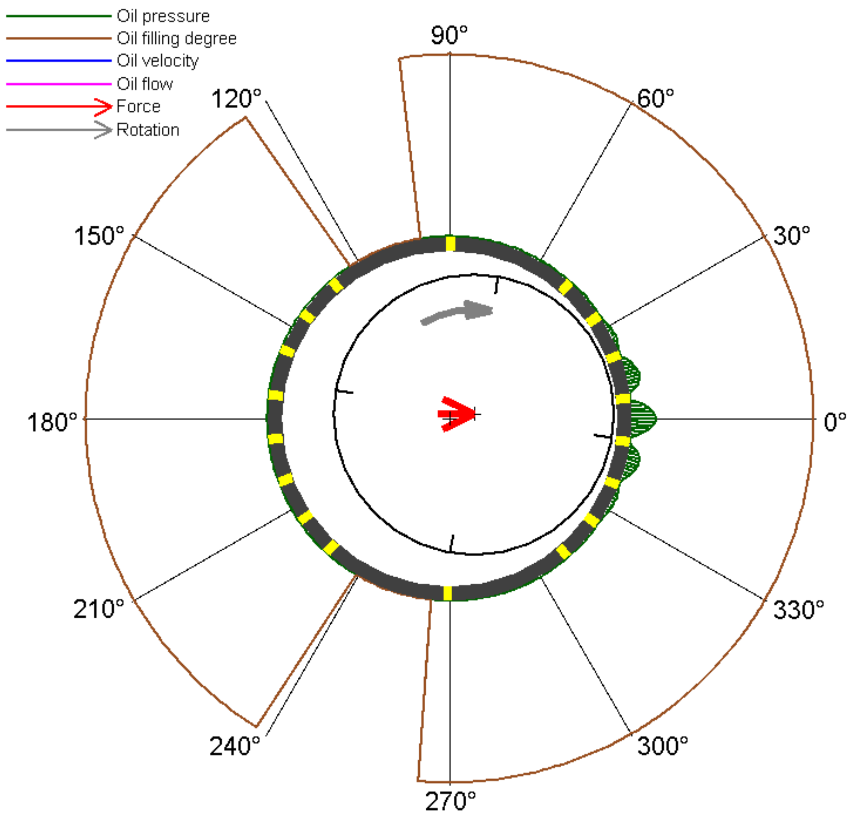
- Differential equation is written in a coordinate system fixed with the bearing shell
  - the external rod load F(t) acts only in connecting rod direction
  - equilibrium between external rod load and hydrodynamic pressure
$$\int_0^{2\pi}\int_{-1}^{1} \cos(\beta)\, So(\beta, b, t)\, db\, d\beta = \frac{2\, F(t)\, \psi^2}{\eta\, \omega\, r\, B}$$
  - perpendicular direction
$$\int_0^{2\pi}\int_{-1}^{1} \sin(\beta)\, So(\beta, b, t)\, db\, d\beta = 0$$
  - Constant oil groove pressure p_{const} of oil unit
$$So(\beta_i, t) = \frac{p_{const}\, \psi^2}{\eta\, \omega}$$

Design approximations

- The relation between load and eccentricity is only valid as long as the bushing is refilled with oil
- To assess the refilling, another relation is needed, $\rho = \dfrac{p_{const} \cdot \psi^2 \cdot J}{\eta \cdot \omega}$
- The calculated value of a given load scenario and bushing design must exceed a <u>critical limit</u> which depends on the bushing geometry
- This “Refilling Characteristic” is much more physically complex than the minimum rod load reversal criterion given in API 618
- Defining only a minimum rod load reversal angle and a corresponding peak load can either be critical or conservative
  - Both of these parameters do not fully describe the refilling mechanism
  - The refilling characteristic contains all variables influencing the refilling

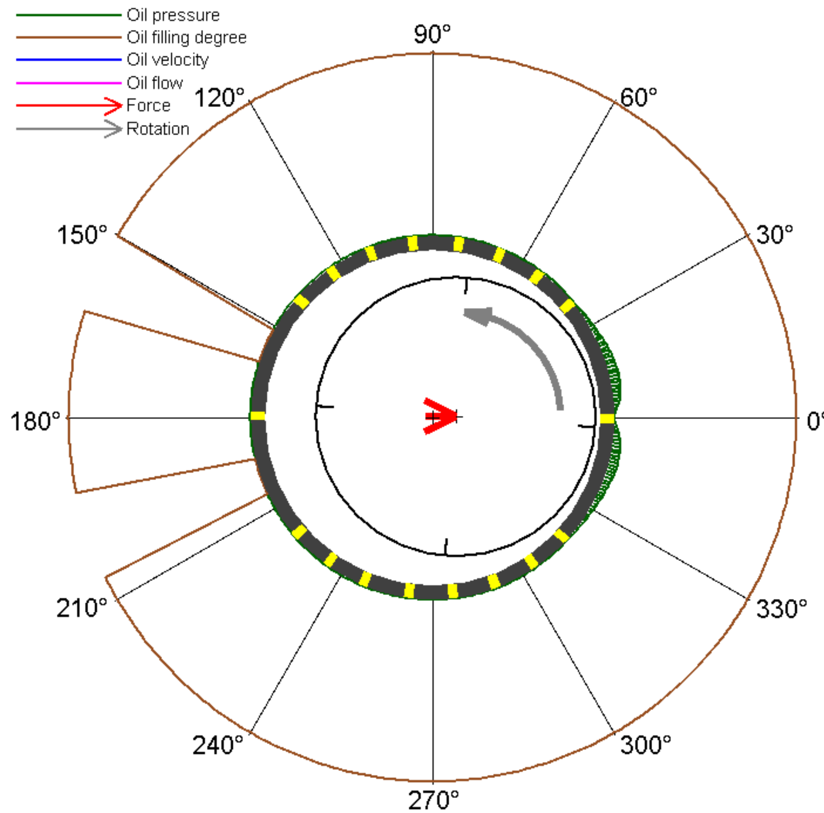
Comparison of Hydrodynamic Oil Film Developed for Different Bushing Geometry and Oil Properties

Bushing with original groove geometry and ISO 100 Mineral Oil	Bushing with modified groove geometry and ISO 150 Synthetic Oil



Peak Pressure = 1.0 (normalized)

Min Oil Film Thickness = 1.0 (normalized)



Peak Pressure = 0.23 (normalized)

Min Oil Film Thickness = 4.7 (normalized)

Conclusions

- Compressor connecting rod bushings failed due to insufficient load capacity / loss of oil film

- This application met API 618 rod load and rod reversal requirements, and a very similar compressor has a history of reliabile operation, yet this compressor was still marginal

- A collaborative effort between End-user and OEM utilizing sound Root Cause Analysis and well known engineering principles resolved the design problem

- The lubrication mechanism of connecting rod bushings has been modeled and has identified Critical Factors:
  1. Maximum oil peak pressure
  2. Minimum oil film thickness
  3. Refilling characteristic of oil to the bushing surfaces

Conclusions / Recommendations

- Compressor OEM's strive to provide reliable compressors utilizing sound engineering principles and practices but are sometimes incentivized to push the envelope

- In reality, the selection of a compressor application depends on the manufacturer's empirical experience with their fleet of compressors

- The end-user, purchaser, and OEM need to confirm the compressor application is "tried and true" in every aspect (rod load, rod reversal, speed, stroke length, materials, etc.)
  - OEM needs to provide references
  - If no suitable references are available, then all parties should at least understand any potential risks and mitigate risks accordingly

Conclusions / Recommendations

- OEM should be able to explain how they model the connecting rod bushing / crosshead pin system with respect to:
  1. Oil film peak pressure
  2. Minimum hydrodynamic oil film thickness
  3. Oil “refilling characteristic” during rod load reversal