Tribological damage of highspeed rotor safety bearings - testing and simulation

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Abstract

The deceleration dynamics and tribology of a high speed rotor system are studied. The rotor is vertical and normally supported by magnetic bearings. In emergencies it is stopped upon retainer bearings. Feasible tribological material pairs are tested by simulation for selecting the best ones for field tests. Dynamic behaviour of the rotor bearing system is studied using analytical modelling and simulation solution and also using a multibody 3D dynamics program. The selected specimens are experimentally tested and the wear damage is characterized with high performance analytical instruments. These observations are used to formulate physical models for explaining the observations and to develop better simulations to explain the surface wear damage at high speeds.

1 Introduction

The general need for saving energy has aroused interest to obtain use of optimally designed high speed machinery for applications like in kinetic compressors, small gas and expansion turbines and milling machines.It has been noted that high thermal efficiencies can be achieved with high speed small rotors designed to rotate on low friction magnetic bearings and to stop in emergencies with minimal damage. At present, dry contact friction bearings are often used to brake the rotors.

The main goal in this study was explore ways to optimise tribologically the safety or retainer bearings to endure decelerations. The first subgoal was to to find feasible material pairs for virtual testings using thermal and dynamical simulations of deceleration and wear. The second subgoal was to manufacture and test the samples in a test rig with closely the same load condition as the actual machine. The third was gain understanding of the physical models using the material characterisation and the fourth was explore models to explain the results for further optimisation.

In this work the scanning electronmicroscopy and the microhardness measurements were performed by R. Westergård and N. Axén at Uppsala University. The tribological tests and the simulations were performed at Lappeenranta University of Technology.

2 Experimental work

2.1 Materials

The goal was find out optimum material pairs for tribological testing at a test machine. In principle virtual optimum design of a multiphase material pairs would be ideal but still too complex. Thus experimental testing is necessary for final selection. The first step was to innovate a large group of feasible tribological material pairs and coating combinations. The second step was to make a preliminary selection using several constraints. One was commercial availability and cost effectiveness. Another was technical constraints on endurance in a design space of pressure, sliding velocity, and temperature variables. Calculations indicated that the flash temperatures would be high and also thermal stresses. For the base ring material the final choices were typical high strength steel and stainless steel. For the retainer bearing the choices were narrowed down to nonmetals and to metals. The selected metals were copperbase and iron base alloys with or without hard or soft coatings, Table 1. Some nonmetal bearings were considered which had performed well at smaller speeds.

Table 1. Materials analyzed at Uppsala University (UU).
Base materials: HT = hardened and tempered steel, SS = stainless steel

Material X of the bearing ring to Base Y	Base: HT	Base :SS
Material A, copper alloy	A to HT	-
Material B, copper alloy	-	B to SS
Material C, hard coating on iron alloy	C to HT	C to SS
Material D, Mo coating on iron alloy	D to HT	D to SS

2.2 Component testing

Several coated and uncoated material pairs of wear ring and base ring, Table 1, were run in a highspeed test rig at LUT, Fig. 1a. Temperatures, deceleration dynamic behaviour and the wear are measured.In the actual vertical machine, Fig.1b, the normal force is gravity force *mg* and in the test machine the same force was generated by a pneumatic cylinder to get the same pressure on the bearing.

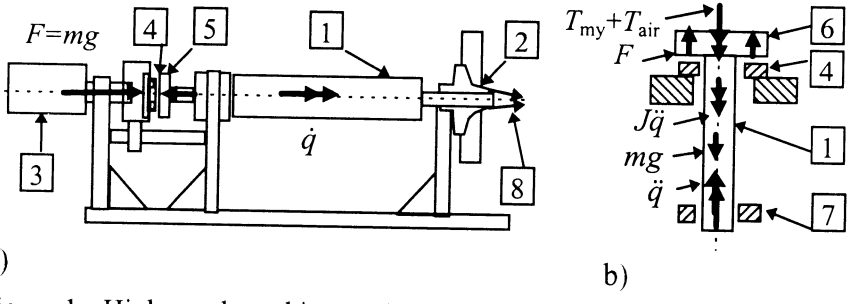


Figure 1: Highspeed machinery. a) Rotor bearing test rig.1. Rotor, 2. air turbine, 3. pneumatic cylinder, 4. safety ring or retainer bearing and 5. base ring. b) Rotor in working position as a FBD,6. radial compressor, 7. radial auxiliary bearing and 8. gas flow.

2.3 Test results with hardened and tempered steel base material

The test results for materials A to D are shown in Fig.2.

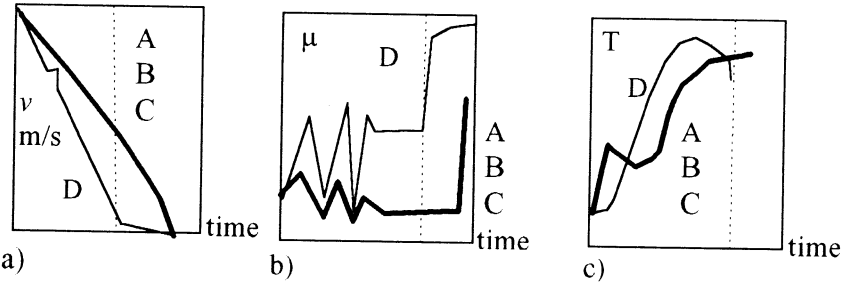


Figure 2: Test results vs. time for the HT base material. a) Sliding velocity, b) coefficient of friction and c) temperature of the midring.

2.4 Test results with stainless steel base material

The test curves with stainless steel base material are shown in Fig.3. Now the lowest friction was with the D material.

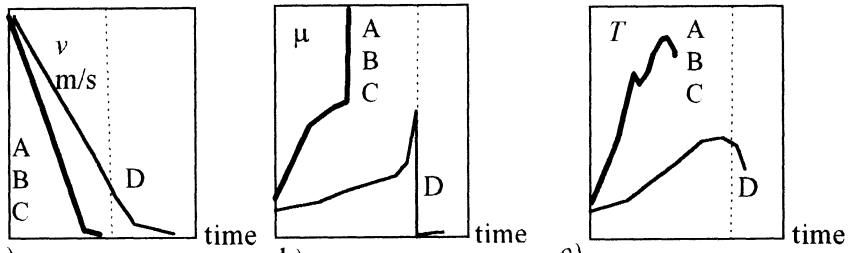


Figure 3: Test results vs time for the SS base material. a) Sliding velocity v(m/s) , b) coefficient of friction and c) temperature of the mid ring.

2.5 Hardness

To achieve hardness profiles of the near surface regions, the microhardness of each specimen was measured on polished cross-sections. The results are shown in Fig.4. Constant hardness distributions were obtained for the brass A 140 ±9, for B 110±5 and 322±18 HV for the countermaterial HT of A. Hardness of coated steels C and D decreased slightly when run against SS but steeply when run against HT, Fig.4. The hardness distribution of soft Mo on iron alloy, D run against HT, was higher and extended deeper than distributions of other samples. The hardness of HT base to D increased in depth while the hardnesses of others decreased in depth.

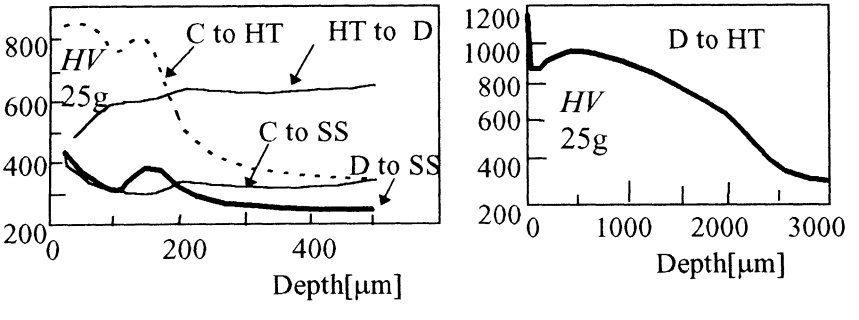


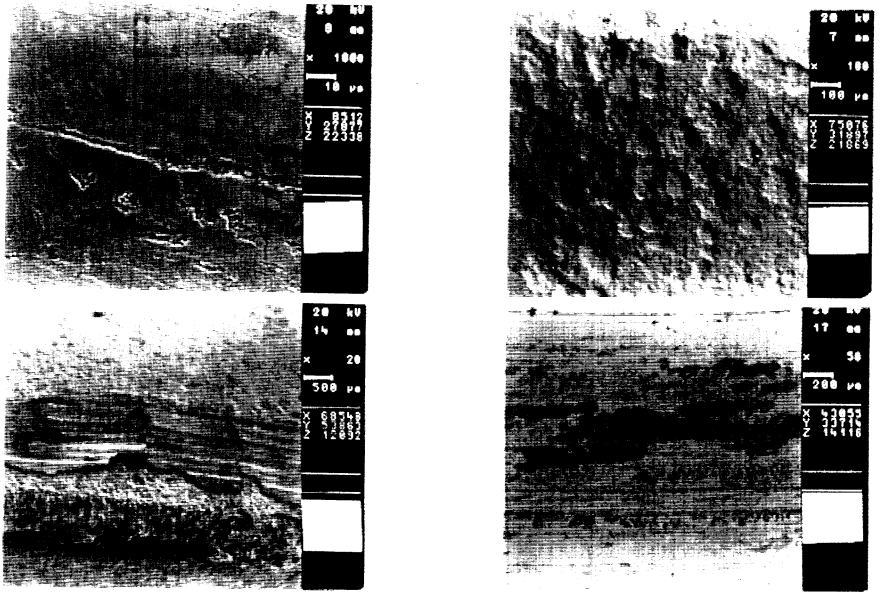
Figure 4: Microhardness measurements of some samples.

2.6 Scanning electron microscopy of the worn surfaces

To characterise the wear mechanisms the worn surfaces were studied by SEM, see Fig.5. A typical feature of rings was material transferred from countersurfaces.

Cracks were found on most surfaces, in particular on the coatings.
The specimen D to HT, Fig. 5a, showed that the Mo coatings on the iron alloy (IA) tested against HT steel remained adhered to the substrate with cracks in coatings and the transferred material.
The specimen D to SS , Fig.5b, showed that the soft Mo coatings on IA were intact when tested against SS stainless steel resulting unusually small material transfer with this pair.
The specimen C to HT showed signs of large material transfer to the rings.The surface was covered with occasionally several layers of material from countersurface cladded onto the rings.
The specimen C to SS, Fig.5c, showed surface pits of unusual crater like appearance. Possibly some kind of a chemical attack has contributed to this damage mechanism.
The brass specimens A to HT, Fig.5d, and B to SS showed signs of abrasive grooves and adhesive contacts. Large areas of plastically deformed material could be observed on both of these materials.

a) c)



b) d)

Figure 5: Scanning electron microscopy images (inverse) with scales showing a) soft coating D to HT, 10μm , b) D to SS, 500μm,c) hard coating C to SS ,100μm and d) brass A to HT, 200μm.

3 Simulations

3.1 Dynamic simulation of the tribological loadings

Dynamic behaviour of the rotor bearing system is studied using analytical modelling and simulation solution and also using a multibody 3D dynamics program Working Model. The simulations are used to design the experimental testings.

3.1.1 Dynamic simulation of the braking deceleration with analytical models

The Jeffcott flexible-rotor model is used as formulated by Childs [1]. The equations of motion for the system shown in Fig.6 are

$$\begin{aligned} m\ddot{R}_X + k_r R_X &= f_X + ma_X\dot{\phi}^2 + ma_Y\ddot{\phi} \\ m\ddot{R}_Y + k_r R_Y &= f_Y + ma_Y\dot{\phi}^2 - ma_X\ddot{\phi} \\ J_z\ddot{\phi} &= T_z + ma_Y\ddot{R}_X - ma_X\ddot{R}_Y \end{aligned} \tag{1}$$

where m is the mass of the rotor, mass of the shaft is not considered, k_r is the shaft stiffness coefficient, J_z is the moment of inertia of the disk about its Z-axis. The components of the external force vector are $\mathbf{f} = \{f_X \; f_Y\}$ and the component of the external moment vector along Z-axis is T_z. The transverse motion vector of the rotor is $\mathbf{R} = \{R_X \; R_Y\}$. The vector **a** is the imbalance vector.Some results of Simnon [2] simulation are shown in Figs.6c and 7.

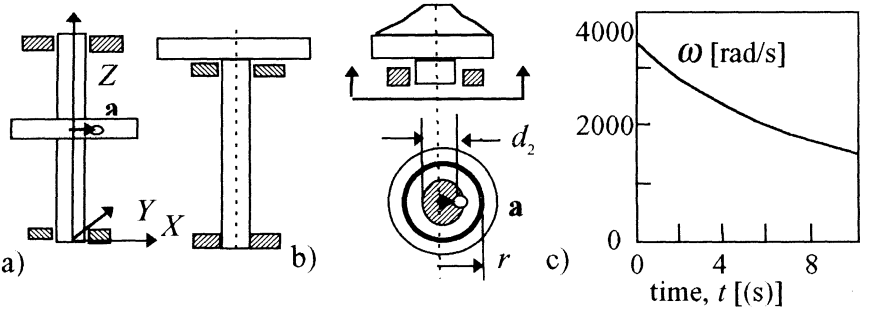


Figure 6: Rotor shaft models. a) The Jeffcott flexible-rotor model, b) present application and c) angular velocity by Simnon simulation [2].

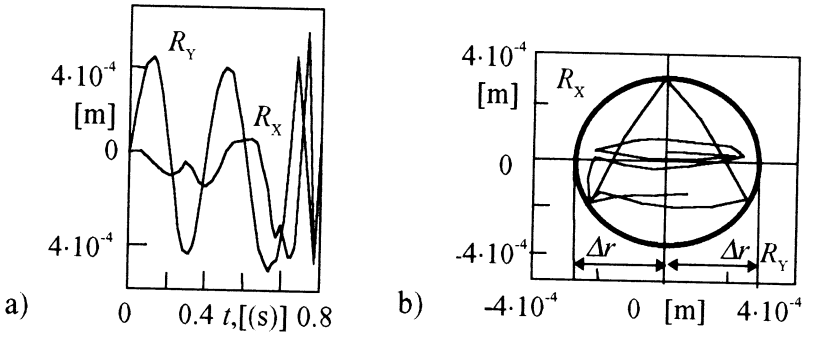


Figure 7: Simnon [2] simulation results. a) Displacements R_X vs R_Y from time 0 to 1s and b) R_X vs. R_Y points as function of time.

3.1.2 Dynamic simulation with multibody dynamics model

The Working Model 3D of version 2 was used [3].A three dimensional AutoCAD model was transferred to WM.The coefficient of friction between the bearing and the shaft was estimated as $\mu = 0.1$. The coefficients of restitution were for the rotor $e = 0.3$ and for the bearings $e = 0.2$. Initial rotational velocity was 20000 r/min. The slowing down time in Fig, 8a is too short and qualitative due to numerical difficulties.

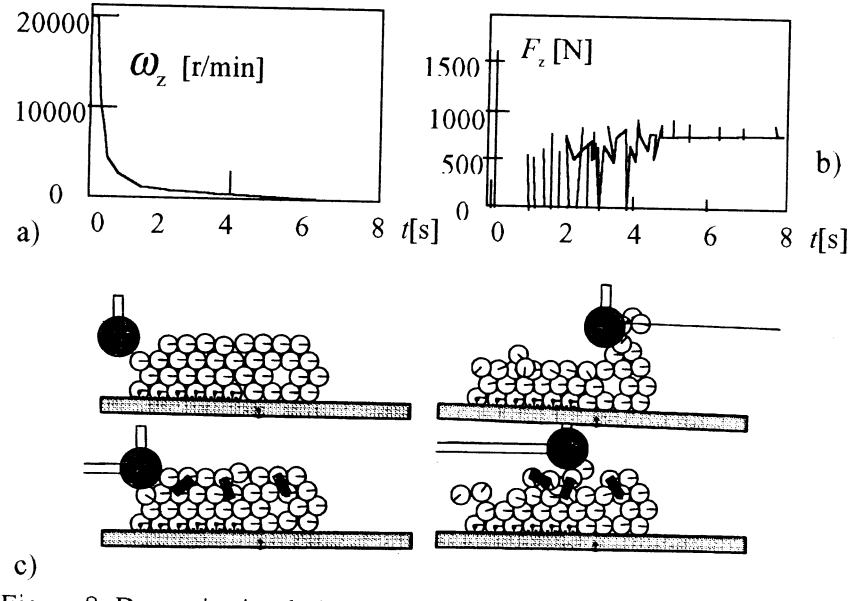


Figure 8: Dynamic simulation with WM. a) Angular velocity vs. time , b) vertical contact between the bearing and the rotor, c) WM simulation of wear with macro elements.

3.2 Models for wear

These observations are used to formulate physical mechanisms and simulation models for explaining them.

3.2.1 Models for simulating the tribological behaviour

The Working Model program is used to study and visualise wear, Fig.8c. The surfaces are modeled with atomistic macro granular elements connected by suitable force law.

3.2.2 Physical models for simulating the tribological behaviour

3.2.2.1 The linear wear model This model gives, Fig. 9,

$$\frac{V}{S} = Z\frac{F}{H} = \frac{F}{\Omega} \quad ,V = A_n h \quad ,h = \dot{h}t \quad ,S = vt, \quad F = pA_n$$
$$\frac{A_n \dot{h} t}{vt} = Z\frac{pA_n}{H} \quad \rightarrow \quad \dot{h} = pv\frac{Z}{H} \tag{2}$$

where V is the worn out volume (m^3), F is normal force (N), A_n is nominal area (m^2), p is nominal pressure (Pa), h is thickness of the worn out layer (m), t is time (s), S is sliding distance (m), v is sliding velocity, Z is the wear coefficient, H is hardness of the wearing surface and Ω is the specific wear resistance. The wear rate of multiphase materials can be developed as by N. Axen [5].The material has two main phases m; matrix and p is the second phase. The volumes worn out ,V_m, V_p ,volume fractions f_m , f_p and normal forces F_m , F_p on each phase are are

$$V = V_m + V_p \quad , \quad 1 = f_m + f_p \quad , \quad F = F_m + F_p \tag{3}$$
$$\frac{dV}{dS} = \frac{dV_m}{dS} + \frac{dV_p}{dS} = A\frac{dh}{dS} = A_m\frac{dh_m}{dS} + A_p\frac{dh_p}{dS}$$

a) EW mode: equal thickness wear rates of the phases

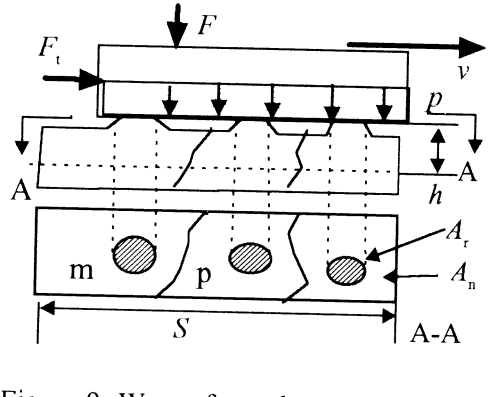
$$\frac{dh}{dS} = \frac{dh_m}{dS} = \frac{dh_p}{dS} \quad , \quad A\frac{dh}{dS} = A_m\frac{dh_m}{dS} + A_p\frac{dh_p}{dS}$$
$$\frac{F_m Z_m}{A_m H_m} = \frac{F_p Z_p}{A_p H_p} \quad , \quad F_m = kF_p = \frac{k}{k+1}F \tag{4}$$

whence

$$\frac{Z}{H} = \frac{A_m}{A}\frac{Z_m}{H_m} + \frac{A_p}{A}\frac{Z_p}{H_p} \quad , \quad \Omega = f_m\Omega_m + f_p\Omega_p \tag{5}$$

b) EP mode: Equal pressure on the phases

$$p = \frac{F}{A} = \frac{F_m}{A_m} = \frac{F_p}{A_p}, \quad F_m = pA_m \quad F_p = pA_p, \quad \frac{1}{\Omega} = \frac{f_m}{\Omega_m} + \frac{f_p}{\Omega_p} \quad (6)$$



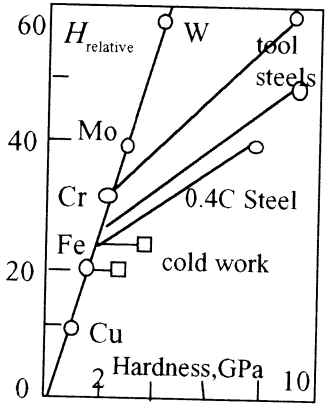


Figure 9: Wear of metals.a) Schemetic of a two body wear of a matrix m and a phase, p and b) abrasion wear resistance S/V=H/(ZF) [6].

3.2.2.2 Hardness and mechanical properties Abrasive wear resistance is proportional to hardness in pure metals but alloys microstructural can increase it also, Fig.9 b).The young's modulus of a polycrystalline metal is not a structure-insensitive property as the yield strength. The average relation for many materials is

$$\sigma_y = 0.003E \quad , \quad \sigma_y = \varepsilon_y E \quad , \quad H = 3\sigma_y \quad (7)$$

3.2.2.3 The adhesive wear rate models for metals and nonmetals According to Rabinowicz [4] the wear coefficient Z depends on the friction coefficient for metals (m) and nonmetals (n),Table 2. Essentially, friction is not a temperature dependent quantity.

$$\dot{h}_k = pv\frac{Z_k}{H_k} \quad , \quad Z_k = A_k \mu^{B_k} \quad (8)$$

Table 2. Coefficients of the wear rate equations [4].

k = m:metals	A_m =3.7·10^{-3}	B_m = 3.7	$H_m = H$
k = n: nonmetals	A_n =26·10^{-6}	B_n= 2.05	H_n = 0.2H

3.2.2.4 Wear under the deceleration stage The model is shown in Fig.6.The rotor is in normal operation supported by magnetic bearings in axial directions. To slow it down an axial retaining bearing is used.It also gives radial guidance to the rotor. The dynamic equation of motion of the rotor in the braking stage is

$$\ddot{q} = \frac{T_{\text{air}} + T_{\text{my}}}{J} \quad , \quad T_{\text{air}} = -sign(\dot{q})k\dot{q}^2 \tag{9}$$

where T_{air} is air torque and T_{my} is frictional torque.
An approximate solution is obtained as follows with numerical estimates

$$\ddot{q} = -K = -\frac{\mu mgr}{J} \quad ,\dot{q}(t) = \dot{q}(0) - Kt, \dot{q}(0) = \omega_0, \quad T = \frac{\omega_0}{K}$$

$$\dot{h} = pv\frac{Z}{H} = (\omega_0 - Kt)rp\frac{Z}{H} \qquad h(t) = \left(\omega_0 t - \tfrac{1}{2}Kt^2\right)rp\frac{Z}{H} \tag{10}$$

$$h(T) = \tfrac{1}{2}\frac{\omega_0^2}{K}rp\frac{Z}{H} = \frac{\tfrac{1}{2}J\omega_0^2}{A}\frac{A_k\mu^{B_k-1}}{H} \quad , \quad Z = A_k\mu^{B_k}$$

Substituting here further the following values: $\mu = 0.35$, $J = 0.18$ kgm^2, ω_o=3351 rad/s m = 85kg, mean ring radius r = 0.04m gives K= 66,stop time T=3551/66=50s. With pressure p =0.61 MPa the wear is $h(T)$ = 20μm.The allowed wear per on deceleration is 100μm.The allowed wear per one deceleration is h = 100μm. The calculated wear at room temperature for steel/steel pair is about 50μm, Fig.10.

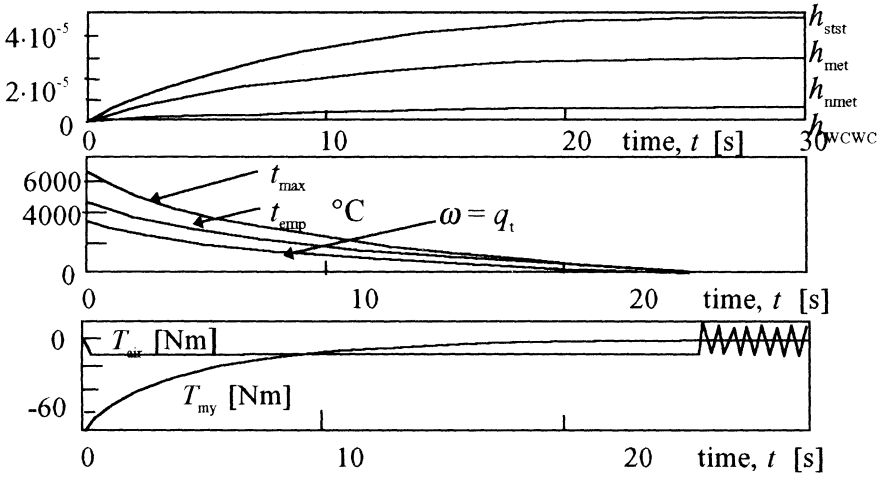


Figure 10: Wear and dynamical models. a) Weared out surface h (m) for h_{met} : for metals, h_{nmet} : for nonmetals, h_{wcwc} for WC/WC pairs, h_{stst} for steel/steel pairs. b) q_t is rotational speed, t_{emp} is mean temperature, t_{max} is flash temperature, c) T_{air} air friction torque, T_{my} frictional torque on the axial bearing surface.

4 Discussion

Successful design of high speed machinery requires interdisciplinary approach combining rotor dynamics simulation, tribology, materials science and heat transfer.

The main goal in this study was to optimise tribologically the retainer bearings to endure sufficiently many decelerations. It was achieved reasonably well. This goal consisted of the following subgoals.

The first subgoal was to find feasible material pairs for testings using simulations of deceleration and wear. Analytical dynamics and wear models gave resonably quantitative results.The deceleration dynamics results with a 3D dynamics simulation were detailed but not fully quantitative. The second subgoal was testing at a test rig. The results showed clearly the best choices.The third subgoal was use of high resolution material characterisation to analyse the tribological wear damage. The fourth was use of predictive tribological models to explain the results for further optimisation. The multiphase wear models by Axen et al [5], [6] showed that the optimal properties of a matrix are high hardness and ductility so that it can support the harder phases without cracking when strains and strain rates are large. The next step would be to develop models which would simulate the observed wear characterisations.

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