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Wear simulation for the journal bearings operating under aligned shaft and steady load during start-up and coast-down conditions

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ABSTRACT

This paper presents a wear analysis procedure and the wear calculation of journal bearings for a stripped-down single cylinder engine during start-up and coast-down. For this analysis, we assume that a steady load is applied to the journal bearings. We utilize lift-off speed to decide whether a journal bearing is in the mixed elasto-hydrodynamic lubrication (EHL) regime. We formulate an equation for the modified film thickness in a journal bearing considering the additional wear volume. Also, we calculate journal bearing wear by using a modified specific wear rate considering the mixed lubrication regime. We show that the accumulated wear volume after 1 turn-on and off of an ignition switch is normally increasing with increasing surface roughness, with a few exceptions.

1. Introduction

Conventional journal bearing operating under steady load are susceptible to wear when subjected to the following conditions: during start-up and coast-down, operating with a misaligned shaft, or when exposed to contamination with abrasive debris within the bearing gap. Also, overload and high temperature would be relevant for wear under steady load. Under these conditions, the bearing is no longer in purely hydrodynamic lubrication (HL) regime and its operation tend to shift to mixed elasto-hydrodynamic lubrication (MEHL) or even boundary lubrication regime.

If the measured angular velocity is less than the lift-off speed (rpm), we consider that the bearing operates in the mixed lubrication region at the instant. In the ideal case of a bearing operation perfectly aligned without the presence of debris, the mixed lubrication region in a journal bearing is likely to occur at a certain initial stage of the start-up and final stage during the coast-down when the motor is shut down. Under these conditions, the lubricant film thickness is not adequate to separate the surfaces and MEHL regime prevails. This occurs when the operating speed is below the so-called lift-off speed.

One can utilize the lift-off speed to determine the point from the mixed lubrication to the hydrodynamic lubrication (HL) during the start-up and the reverse shifting during the coast-down periods. The onset of mixed lubrication regime can also be related to film thickness parameter Δ defined as the ratio of the film thickness and standard deviation or root mean square (rms) of combined surface roughness, h/σ.

In what follows, we begin by citing some of the pertinent published works available in the open literature. Chun [1] utilized the measurement data of journal angular velocity during the start-up and coast-down conditions measured by Fragoulis [2] to compute bearing friction. He showed that the eccentricities at every cranking angle can be calculated under the corresponding applied loads by employing the mobility method [1,3,4].

He [2] utilizes a stripped single engine to measure these angular velocities. The cylinder head, piston and connecting rod were removed from the cylinder block of a single cylinder diesel engine. Originally, this experiment was designed to measure the instantaneous friction in the related engine components. The operation was involving starting the motor and then letting it to coast down. The duration of this procedure was very short and temperature measurements of the oil indicated that its temperature remained constant in the 30 s or less required for the experiment.

Mokhtar et al. [5] described the behavior of plain, hydrodynamic journal bearings during starting and stopping under a steady load by experimental investigation. At starting, a rapid buildup of hydrodynamic forces occurred. A hydrodynamic film was formed in a very short time, after which the shaft moved in a spiral shaped whirling locus to the steady state operating position. Prior to separation of the shaft and bearing surfaces, the contact was mainly a sliding situation with little or no initial rolling. At

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Nomenclature

a_χ	diameter of an area associated with an absorbed molecular (m)
A_c	cross-section of worn part or wear area of a journal bearing $\left(=\frac{R^2}{2}[2\alpha-\sin 2\alpha]-\frac{(R+c)^2}{2}[2\lambda-\sin 2\lambda]\right)$ (m^2)
b	wear scar width (m)
c	radial clearance of a bearing (m)
$c_w(\theta')$	additional space for a bearing radial clearance ($\doteqdot c'_w(\theta')*\cos(\pi-\theta'-\varphi)$) (m)
$c'_w(\theta')$	y_2-y_1 (m)
$c_M(\theta')$	a modified bearing radial clearance ($=c+c_w(\theta'), \pi-\lambda \le \theta' \le \pi+\lambda$) (m)
$C_M(\theta')$	$1+c_w(\theta')/c$ (m)
$C'_w(\theta')$	$c'_w(\theta')/c=(y_1-y_2)/c$ (m)
d	wear depth (m)
D	Shaft diameter (m)
e	nominal eccentricity of a journal bearing (m)
e_M	modified eccentricity of a journal bearing (m)
E'	half effective modulus of elasticity $\left(=0.5\left(\frac{1-\nu_1}{E_1}+\frac{1-\nu_2}{E_2}\right)\right)$ (Pa)
E''	effective modulus of elasticity $\left(=\left(\frac{1-\nu_1}{E_1}+\frac{1-\nu_2}{E_2}\right)\right)$ (Pa)
E_a	heat of adsorption of lubricant on a surface (kJ/mole)
E_1	Young's modules of a shaft (cast iron) (GPa)
E_2	Young's modules of a bearing (White metal=Babbitt metal) (GPa)
G	material number ($=\alpha E'$)
$h(\theta)$	nominal oil film thickness (compliance) (m)
h_c	central film thickness in line contact configuration (m)
$H(\theta)$	non-dimensional form of oil film thickness $\left(=\frac{h(\theta)}{c}\right)$
H_c	dimensionless central film thickness $\left(=\frac{h_c}{R_e}\right)$
H_s	dimensionless oil film thickness $\left(=\frac{h(\theta)}{\sigma}\right)$
hd	Vickers hardness of the softer material ($=$hd$_2$) $\left(\frac{N}{m^2}\right)$,
hd$_1$	Hardness (Vickers) of a shaft (cast iron, MPa)
hd$_2$	Hardness (Vickers) of a bearing (White metal=Babbitt metal) (MPa)
I_1, I_2, I_3	$\frac{H-\bar{y}_s}{\sigma}, \frac{H-\bar{y}_s+\bar{w}_1}{\sigma}, \frac{H-\bar{y}_s+\bar{w}_2}{\sigma}$ for general film thickness
I_1, I_2, I_3	$\frac{H_c-\bar{y}_s}{\sigma}, \frac{H_c-\bar{y}_s+\bar{w}_1}{\sigma}, \frac{H_c-\bar{y}_s+\bar{w}_2}{\sigma}$ for central film thickness
k_s	specific wear rate or wear modulus ($=K/$hd or $\frac{2\alpha-\sin 2\alpha-(1+\zeta)^2(2\lambda-\sin 2\lambda)}{8\pi p_a n}$) (m^2/N).
k_a	modified specific wear rate for mixed lubrication regime ($=\Psi k/\gamma_2$)
k_b	intermediate modified specific wear rate ($=\Psi k$)
K	dimensionless Archard wear coefficient
l	contact length (m)
l_c	length between bearing #1 and flywheel (m)
L	bearing length (m)
L/D	bearing ratio
L_C	length between bearing #1 and bearing #2 (m)
L_g	central oil groove width (m)
L_r	bearing land (m)
m_C	mass of crankshaft (kg)
m_F	mass of Flywheel (kg)
n	asperity density (m^{-2})
$\bar{n}$	dimensionless asperity radius ($=nR_e{}^2$)
N	total number of revolution or required lifetime revolution
N_T	lift-off speed or transition speed (rpm)
p	total interface pressure between the bearing and the shaft (Pa)
p_h	fluid hydraulic pressure (Pa)
p_a	asperity contact pressure (Pa)
p_b	projected bearing load ($=W_n/DL$) or apparent pressure in a journal bearing (N/m^2)
$p_b kN$	$\frac{2\alpha-\sin 2\alpha-(1+\zeta)^2(2\lambda-\sin 2\lambda)}{8\pi}$
$p_{elastic}$	elastic contact pressure (Pa)
$p_{elasto-plastic}$	elasto-plastic contract pressure (Pa)
$p_{plastic}$	plastic contact pressure (Pa)
P_a	non-dimensional asperity contact pressure $\left(=\frac{4R_e}{E'b}p_a\right)$
P_{in}	oil inlet pressure (Pa)
R	shaft radius (m)
R_e	effective radius of curvature or equivalent contact radius $\left(=\left(\frac{1}{R}\pm\frac{1}{R_b}\right)^{-1}\right)$ (m)
R_b	bearing radius (m)
R_g	gas constant (J/(mole K))
S	sliding distance (m).
S	time differential of sliding distance (m/s)
t	coordinate of time (s)
t_o	fundamental time of vibration of molecule in an absorbed state (s)
T_{in}	oil inlet temperature ($=23.3$ °C)
T_s	absolute temperature of surface film (K)
u_r	rolling velocity of two contacting surfaces ($=(u_1+u_2)/2$) (m/s)
u_s	sliding velocity ($=u_1-u_2$) (m/s)
U	dimensionless velocity $\left(=\frac{\mu_O u_r}{E'R_e}\right)$
V	dimensionless hardness number $\left(=\frac{hd}{E'}\right)$
V_w	wear volume of a journal bearing ($=A_cL$) (m^3)
$\dot{V}_w$	wear volume rate (m^3/s)
V_{w_1}	wear volume at each crank angle in 1 revolution ($=V_w/N$) (m^3)
w	load per contact length (N/m)
$w^*=z^*-h^*+y_s^*$	(starred variables are normalized by σ) ($=z^*-I_1$)
w_1	critical interference at the point of initial yield ($=(0.6\pi V)^2\beta$) (m)
$\bar{w}_1$	w_1/R_e
w_1^*	w_1/σ
w_2	critical interference at the point of fully plastic flow ($=54w_1$) (m)
$\bar{w}_2$	w_2/R_e
w_2^*	w_2/σ
W	dimensionless total normal load (or force) $\left(=\frac{w}{E'R_e}\right)$
W_a	asperity contact load (N)
W_C	weight of crank shaft$=m_Cg$ (N)
W_F	weight of flywheel$=m_Fg$ (N)
W_h	hydrodynamic load (N)
W_n	total normal load or force (N)
W_1, W_2	applied load of Bearing #1 and #2 (N)
$\overline{W}$	dimensionless total normal force $\left(=\frac{W_n}{lE''R_e}\right)$
x	$\frac{b_w}{2}=R\sin(\pi-\theta)$ (m)
y_1	$\left\|\left[\sqrt{R^2-x^2}\right]\right\|$ (m)
y_2	$\left\|\left\{\sqrt{\left(R^2-x^2\right)}-d\right\}\right\|$ (m)
y_s	distance between the mean line of the surface heights and the mean line of the surface summits, ($y_s=\frac{0.0459}{n\beta\sigma}\sigma$ or $y_s=0.92\sigma$) (m)
$\bar{y}_s$	$\frac{y_s}{R_e}$
Z	coordinate of longitudinal direction (L/R)
Z_p	pressure–viscosity index ($=0.48$)
α_p	pressure–viscosity coefficient ($=Z(5.110^{-9}(\ln(\mu_o)+9.67))$)