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Tribological failure analysis of gear contacts of Exciter Sieve gear boxes

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ABSTRACT

In the present paper the premature failure of gear contact encountered in Exciter Sieve gear boxes has been analyzed. The cause of gear contact failure is identified by simulating the load bearing capacity of lubricants and conducting controlled experiments on an Amsler disk-on-disk tribo tester. The results of performance behavior (i.e., load carrying capacity of lubricants, contact friction and weight loss of test specimens) of the simulated gear contacts have been reported. The theoretical and experimental results indicate presence of mixed to partial elastohydrodynamic lubrication conditions in the gear contact. To mitigate the problem of scuffing and scoring in the gear contacts, lubricating oils with extreme pressure additives and Base oil without additive have been tested and performance results are reported.

1. Introduction

Exciter sieves are excessively used in the coal crushing mills to distinctly separate out the coal particles of varying dimensions using sieves of different grit sizes. A typical Exciter Sieve consists of 3–4 sieves stacked vertically in a frame with coarse sieve at the top and finest at the bottom. The gear box with hanging eccentric counter weights is housed at the top of the exciter frame and is inclined 30–45° from the horizontal. The drive shaft of the gearbox is coupled with an electric motor and the gears used in the gearbox typically are the spur gears. The driver and the driven shafts are supported on spherical roller bearings at both the ends. Being inclined, the gear box is partially filled with the lubricant providing sufficient space for churning and aeration of lubricant while in operation. The entire assembly of Exciter Sieve is supported on 6–8 heavy duty helical springs providing sufficient displacement in all the three dimensions while in operation. The inclined gear box and hanging eccentric weights, while in operation, provides the thrust and vertical movement to the assembly thus, providing movement of the sieves and therefore leading to sieving of the feed i.e., the coal particles. Fig. 1 shows the photographic image of an inclined gear box with hanging eccentric weights while in operation.

The gears in this application are lubricated with mono and multigrade lubricating oils and are exposed to impact loads due to the hanging eccentric weights. The load acting on the gear contact is calculated by assuming that the two

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Nomenclature

b	half Hertzian contact width, $4R\sqrt{W/2\pi}$ (m)
E	equivalent Young's modulus (Pa), $2/E = [(1-\upsilon_1^2)/E_1 + (1-\upsilon_2^2)/E_2]$
$E_{1,2}$	elastic modulus of rollers 1,2 (Pa)
G	non-dimensional material parameter, $G = \alpha E$
h	film thickness (m)
h_0	film constant (m)
H	non-dimensional film thickness
H_0	non-dimensional offset film thickness, $H_0 = h_0/b$
K_{ij}	discretized kernel
l	contact length (m)
N	number of grid points
N3, N4	surface finish N grades as per ISO 1302
p	pressure (Pa)
P	non-dimensional pressure, $P = p/P_H$
P_H	max. Hertzian pressure (Pa), $P_H = Eb/4R$
R	equivalent radius of cylinder pair (m), $1/R = (1/R'_1) + (1/R'_2)$
$R'_{1,2}$	radii of cylinders 1,2 (m)
S	distance between point of gear contact and pitch point
T_0	inlet lubricant temperature (K)
T_m	mean lubricant temperature (K)
T	dimensionless mean temperature, $(T_m/T_0 - 1)$
u_s	average rolling speed, $u_s = \frac{u_1+u_2}{2}$ (m/s)
$u_{1,2}$	velocity of the disks 1,2 (m/s)
$U_{1,2}$	non-dimensional speed parameter, $U = \frac{\eta_0 u_s}{ER}$
w	load per unit length (N/m)
W	non-dimensional load parameter, $W = \frac{w}{ER}$
x, y, z	co-ordinates
$x_{l,c}$	left-inlet and cavitation co-ordinate of the contact zone
$X_{l,c}$	non-dimensional left-inlet and cavitation co-ordinate of the contact zone
X	non-dimensional co-ordinate, $X = \frac{x}{b}$
$X_{in,out}$	non-dimensional in and out co-ordinate of contact zone
z'	Roelands pressure–viscosity parameter, $z' = \frac{\alpha}{(\ln \eta_0 + 9.67)(5.1\times10^{-9})}$
α	pressure–viscosity coefficient (Pa^{-1})
β	thermal expansion coefficient of lubricant
γ	thermal viscosity coefficient of lubricant
$\bar{\eta}$	viscosity of lubricant (Pa-s)
η	non-dimensional viscosity of lubricant, $\eta = \frac{\bar{\eta}}{\eta_0}$
η_0	inlet viscosity of lubricant (Pa-s)
$\nu_{1,2}$	Poisson's ratio of material of surfaces 1,2
$\Omega_{1,2}$	angular velocities of gear surfaces
ψ	pressure angle
$\bar{\rho}$	density of lubricant (kg/m^3)
ρ	non-dimensional density of lubricant, $\rho = \frac{\bar{\rho}}{\rho_0}$
ρ_0	inlet density of lubricant (kg/m^3)

counterweights at the shaft end to be clubbed together. On the basis of this weight, the centrifugal force is calculated to be ≈159 kN. In order to compensate the dynamic conditions, it is assumed that there is a 5% increase in the force. A total load of 168 kN is arrived by adding the weights of bearings, gear, splashing ring, labyrinth ring and the bearing cover. The tangential force on the gear considering the Lewis Factor of 0.48 approximates to 265 kN. The maximum value of the forces (radial/tangential) is considered to calculate the maximum contact pressure i.e., 3390 N/mm^2. The lubricating film formed in between the gear contacts therefore, has to bear heavy impact loads and remain adsorbed onto the surfaces so as to avoid undue wear resulting into scuffing and scoring of gear teeth. Scuffing manifests itself as, sudden failure of lubricating films in mechanical equipment operating under extreme conditions of load and/or speed [1]. The gear teeth often fail due to scoring/scuffing and as a result of this the entire system is exposed to maintenance in every 4–6 months of operation. The failures of these gear boxes impact on the production loss and maintenance costs are usually high. Typical failures of the gear surfaces are photographed in Fig. 2. Visual inspection of used gears shows rigorous scoring and scuffing marks on the gear teeth in the rolling direction. These observations reveal the failure of lubricating films within the gear contact. The failure of lubricating film