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Aircraft landing gear greased slider bearing steady-state thermo-elastohydrodynamic concept model

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ABSTRACT

This paper presents a steady-state concept model for studying the thermal behavior of a greased aircraft landing gear lower slider bearing. Structural damage has been reported as a consequence of excessive heat generated by the high loads induced by rough runways on the bearings, and by the high sliding velocities of the piston. The goal of the model is to enable fundamental understanding of the frictional heat generation. The governing equations are adapted for grease flow and special attention is given to the underlying algorithm of the developed numerical framework used to efficiently solve the governing equations. The developed numerical code is validated against existing results. Numerical results indicate fundamental differences in fluid flow behavior between greased and oil-lubricated bearings.

1. Introduction

In civil aviation, aircraft are required to operate on a variety of runways. As a result, investigations into aircraft maneuvering on rough runways are necessary. High loads on the shock absorber bearings and high sliding speeds induced by rough runways lead to excessive heat generation at the slider bearings, eventually causing structural damage.

The root cause of the reported overheating issues has been postulated to take place at the lower bearing–piston sliding interface of the main landing gear (MLG) (see Fig. 1).

Since the derivation of the Reynolds equation for very thin fluid films more than a century ago, elasto-hydrodynamic (EHD) studies have focused on the isothermal performance in order to design high efficiency bearings. For more than 30 years, lubrication studies have been extended to include temperature effects [2]. Many of the thermo-elasto-hydrodynamic (TEHD) studies were steady-state and focused on the performance of the bearings [3–8]. Recent literature on the performance of slider bearings is very scarce [9].

Transient TEHD analyses are the exception and only a few studies have focused on vibrations or oscillating machinery parts [10]. Non-Newtonian lubricants have been the objective of several studies, but were very often restricted to lubricants that are shear thinning/thickening [11–15]. One of the most common lubricants, namely grease, has not been considered very frequently [16–18].

Currently, the focus of numerical TEHD studies has shifted to computational fluid dynamics (CFD). The differences between solving the complete Navier–Stokes equations by CFD and solving

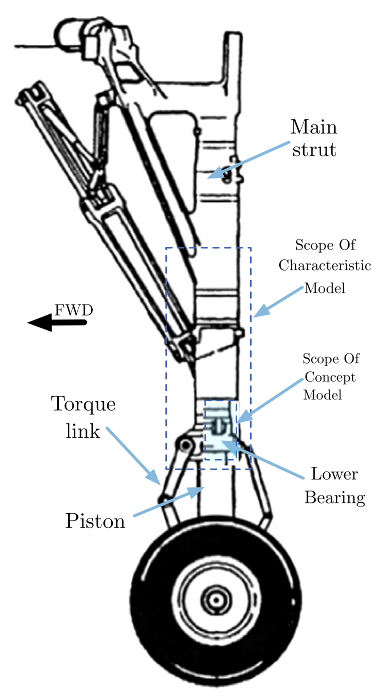

Fig. 1. Main landing gear (MLG) [1].

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Nomenclature

$\mathbf{A}$	matrix of linear system
$A_{1,2}^{+,-}$	velocity field parameters
$A_{(i,j)}$	matrix coefficient at row i and column j
A_k	velocity parameter k
α	thermal expansion coefficient (α_s: solid)
B, L	characteristic width and length of structure
$\mathbf{b}$	right-hand-side vector
c	bearing lubrication profile
c'	plug flow thickness
c_0	static minimum clearance
c_p	lubricant's heat capacity
δc	radial displacement of the plug from centerline
$\mathcal{C}$	bearing configuration set
$\tilde{C}_k^x, \tilde{\tilde{C}}_k^x$	integral factor (simple and double integral)
∂	border to domain Ω: $\partial\Omega = \Gamma$
$\tilde{\Gamma}_1, \tilde{\Gamma}_2, \tilde{\Gamma}_3$	Reynolds equation parameters
$\dot{\gamma}$	lubricant shear rate
$\dot{\boldsymbol{\gamma}}$	shear rate tensor
$\mathbf{d}$	displacement vector field
$\mathbf{D}$	stretching tensor
e	lubricant's energy
$\mathbf{e}$	numerical error vector
$\mathbf{E}$	stiffness matrix
ϵ	ratio of c_0 to L_c
$\boldsymbol{\epsilon}$	strain tensor
$\mathbf{f}$	external force vector
$\mathcal{F}_i$	functional i
F_x, F_y	horizontal and vertical forces
h	heat transfer coefficient
η	lubricant's viscosity (η^*: base oil viscosity)
η_0	lubricant's viscosity at ambient temperature
$\mathcal{H}$	viscosity functional
θ	temperature field (θ_g: drop-point)
i, j, k	dummy indices
λ	lubricant's thermal conductivity (λ_s: solid)
λ'	second viscosity parameter
L_c	bearing length
μ	bearing friction coefficient
m	order of power law for lubricant's shear stress
M_z	bending moment
$\mathbf{n}$	normal unit vector to Γ
N	integer for vector/matrix size
p	pressure field
R	internal radius of piston
$\mathbf{r}$	residual vector
r_i	vector coefficient at row i
ξ	volumetric fraction of thickener and oil
ρ	lubricant's density (ρ_s: solid)
s	slope of thrust portion of bearing
$\dot{s}$	shock absorber relative sliding speed
$\boldsymbol{\sigma}$	stress tensor
$\mathbf{T}$	Cauchy stress tensor
τ	shear stress field
τ_0	initial shear stress of grease
$\tilde{U}$	characteristic velocity
$\mathbf{v}$	fluid film velocity vector field
v_y'	plug flow velocity
v_{y_+}, v_{y_-}	velocity in $\oplus$ region (velocity in $\ominus$ region)
$\tilde{\mathbf{v}}$	approximate solution vector
$\hat{\mathbf{v}}$	numerical solution vector
w_x, w_y, w_y'	normal, tangential and shear loads
x, y	radial and vertical positions
x_+, x_-	plug flow boundaries
y_0	length of thrust portion
$\boldsymbol{\Omega}$	spin tensor
Ω^k	domain k

the Reynolds equation are generally small (within 1%) [19]. The general aim of this study is to develop a TEHD model that can be solved in the most efficient way with minimum computational effort. The goal of this paper is to highlight the mathematical derivation of the governing equations which enabled quantitative assessment of the overheating problems. Although a steady-state regime never occurs when an aircraft is maneuvering on a rough runway, it may be modeled as a sequence of steady-state simulations, each with initial conditions of the previous run that results in a fully transient simulation.

2. Methodology

To the best of our knowledge, slider bearings of landing gear (LG) have not yet been analyzed from a TEHD point of view, since no immediate need has existed thus far. Very high accuracy such as that provided by full CFD is not sought for lubricated contacts in slider bearings of aircraft LG. The main characteristics of the TEHD model for aircraft LG are very fast computation times for nearly exact solutions, including the freedom of using different rheological models and integration to an existing framework in industry.

2.1. Model definition and assumptions

The *Concept Model* shown in Fig. 2 uses an adapted Reynolds equation and is strongly coupled to the energy equation. The goal of the *Concept Model* is to derive the modified Reynolds equation for a 2D thermal analysis involving vertically moving solids. In addition, the aim of the *Concept Model* is to accurately calculate the temperature field in the fluid and structure domains. The temperature field in domain $k = 1,2,3$ is defined as $\theta^k : \Omega^{(k)} \rightarrow \mathbb{R}$ such that $(x, y) \mapsto \theta^k(x, y) \in C^2(\Omega^{(k)})$.

The assumptions are in line with the standard assumptions of the Reynolds lubrication theory. Throughout the present analysis, a line contact with no side leakage is assumed. The surfaces in contact with the lubricant are assumed to be smooth. The grease itself is modeled as a Bingham plastic with a power law of order $m = 1$. The viscosity of the base oil is dependent only on pressure and temperature, and the lubricant is assumed to be entirely incompressible ($\rho_{,p} = 0$ and $\rho_{,\theta} = 0$). In the derivation of the modified Reynolds equation, all the body forces, surface tension forces, as well as the lubricant's inertia forces are omitted ($D\mathbf{v}/Dt = 0$). The structural components surrounding the lubrication gap are considered to be linear and isotropic. The shock absorber oil and surrounding air temperatures are constant. All fluid velocities are $O(1)$ ($\overline{v}_{\overline{y},\overline{x}} \geqslant \epsilon^2 \overline{v}_{\overline{x},\overline{y}}$) and the pressure derivatives are not of equal order: $\overline{p}_{,\overline{y}} \geqslant \overline{p}_{,\overline{x}}$. To simplify the Navier–Stokes equations, higher order velocity derivatives across the gap are ignored: $\overline{\eta}_{,\overline{y}} \overline{v}_{\overline{y},\overline{x}} \geqslant \overline{v}_{\overline{x},\overline{x}\overline{x}}$. The thermal conductivity λ of the fluid film is constant ($\lambda_{,\theta} \sim 0$) for the considered temperature range. Convective cooling along the film is ignored and the flow within the lubrication gap is considered to be laminar.

The boundary l of the domain k is denoted as Γ^{kl}. The fluid–structure interfaces between the domains k and k' are denoted as $(\forall k, k' : k \neq k')\ \Gamma^{k/k'} = \Gamma^{k'/k}$. The normal vector to the interface