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Prediction of the thermo-fluids of gearbox systems

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ABSTRACT

The thermal and multiphase flow physics are among the leading determinants of the performance, durability and life of gearbox systems and are crucial to devising optimization guidelines to minimize the power loss associated with their operation. Despite the significance of the thermo-fluids of the gearbox system, limited capability has been developed to predict and characterize these phenomena in a wholistic fashion. This is, in part, due to the significant complexity and multi-scale nature of the physical phenomena involved. This paper demonstrates the application of a recently developed modeling methodology (Yazdani and Soteriou, 2014) for predicting the time-dependent thermo-fluid state of the gearbox system on a set of three interlocking gears that serve as a validation vehicle. Along with the modeling approach which is applicable to any multiscale problem of this nature, a novel gridding methodology is presented for simulating interlocking gears with substantially improved accuracy. The long-term (time-independent) temperature distribution within the gearbox is obtained through the proposed simulation method and compared to the experimental measurements. In addition, the model is shown to be able to reproduce the temporally cyclic temperature oscillations once the system reaches the stationary-state condition.

1. Introduction

Efficiency of geared systems, such as power, propulsive systems and rotorcraft or aircraft applications, has been a renewed interest and increasingly important research topic due to more stringent energy economy requirements and environmental concerns. Given the sheer amount of power transferred through gearbox systems, very small inefficiencies (i.e. typically $\leqslant 1\%$) translate into megawatts of power loss [2]. Power losses in a gearbox can be classified into two basic groups: load-dependent (mechanical) power losses and load-independent (spin) power losses, both of which are largely driven by the thermal-fluid characteristics of the system. For instance, the thermal state, i.e. temperatures of gears, bearing and lubricant are critical parameters affecting the life and performance of these components. In addition, the local oil flow distribution in the vicinity of the gears meshing region along with its temperature-dependent properties are amongst the primary variables controlling the liquid film thickness, hence the contact-generated heat.

As the result of the importance of thermal characteristics of the contact zone, large number of research studies have been focused on developing modeling capabilities for the temperature rise preceding the scuffing phenomenon, which is known to be as one of the most common surface failure modes observed at lubricated contacts [3,4]. This has led researchers to focus on the submicron phenomena to predict the friction-oriented, instantaneous temperature rise at the contact zone [5], a practice commonly referred to as the "thermal" extension of the Elastohydrodynamic Lubrication theory or TEHL [6–9]. While the model and its variations have been extensively used to simulate and predict the thermal phenomena of the highly loaded surfaces in general (e.g. [10–12]) and the contact regions of gear surfaces in particular (e.g. [13–16]), they all require a critical simplification of the so-called "bulk" events, that is the large-scale thermal-fluid phenomena inside the gearbox (e.g. [17–19]). That, in effect, decouples those large-scale physics from the contact region. Further simplifications include solving for transient heat-conduction inside the solid components while representing the fluid side by some heat-transfer coefficient approximation (e.g. [20–22]).

The windage-induced power-loss and ways of its minimization are extensively studied in the past decade mainly towards experimental measurements and semi-empirical models for isothermal fluid flow characteristics of the gearbox (e.g. [23,24,2]). More recently and thanks to the advances in computational techniques and resources, several researchers have paid special attention to modeling and simulation of fluid flow in gearbox systems using Computational Fluid Dynamics (CFD) (e.g. [25–27]). Again, the

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primary focus of these efforts is to characterize the single-phase iso-thermal fluid mechanics and the resulting windage losses inside the gearbox which serves as a guideline towards its design optimization.

Despite the rather extensive, yet separated realms of research in the area of macro-scale fluid-mechanics and micro-scale contact modeling, the interaction of the two was not a subject of investigation due, in large part, to the immense separation of length and time-scales that exists between the two phenomena. Recently, the real-time thermo-fluid analysis of gearbox system was studied by Yazdani and Soteriou [1] where a novel numerical approach was proposed to resolve the time-dependent stationary state of the thermo-fluid of the systems at the expense of bypassing the initial transient (e.g. warm-up) of the gearbox. Although only valid in certain regimes of operation, it was shown that the approach can apply to a wide range of aerospace and automotive applications where the gearbox is expected to operate at a certain set of operating conditions for an extended period of time (> few minutes). The numerical approach was verified against a full-fidelity thermo-fluid simulation of a simplified system and demonstrated on a jet-lubricated single rotating gear subject to a hypothetical heat source. This paper presents the implementation of a this approach to predict the thermal and flow characteristics of a gearbox comprised of three interlocking gears in which experimental data was obtained for validation. The validation includes the instantaneous oil flow distribution within the gearbox along with local temperature distributions at several locations both at the steady- and stationary-states of the gearbox.

2. Formulation

The theoretical model is adapted from [1] were the formulation was presented for a generic gearbox with turbulent flow regime and is simplified here for a laminar flow case as per operating conditions (see Section 5). This results in the following set of dimensionless equations governing the thermo-fluids of the gearbox,

$$\nabla \cdot \mathbf{u}_f = 0 \tag{1}$$

$$\epsilon \frac{\partial \mathbf{u}_f}{\partial t} + \mathbf{u}_f \cdot \nabla \mathbf{u}_f = -\nabla p + (\epsilon^2)\nabla^2 \mathbf{u}_f \tag{2}$$

$$\epsilon \frac{\partial \alpha_k}{\partial t} + \mathbf{u}_f \cdot \nabla \alpha_k = 0 \tag{3}$$

$$\epsilon \frac{\partial \Theta}{\partial t} + \mathbf{u} \cdot \nabla \Theta = \left(\frac{\epsilon^2}{\mathrm{Pr}}\right)\nabla^2 \Theta + q(\mathbf{x}, t) \tag{4}$$

where $u_{\mathrm{ref}} = \Omega D, p_{\mathrm{ref}} = \rho \Omega^2 D^2$ and $q_{\mathrm{ref}} = \rho c_p T_{\mathrm{ref}} \Omega$ are selected as normalization factors for velocity, $\mathbf{u}$, pressure, p, and heat generation rate, $\dot{q}$, respectively. The spatial coordinates are normalized by characteristic length, D while a composite time scale, $\tau = \sqrt{D^2/\nu\Omega}$, is selected as the reference time-scale (the reader is referred to [1] for the justification of this non-dimensionalization approach). The parameters, Ω and D refer to the angular velocity of a reference gear and its diameter, respectively and the properties, ν and γ stand for the kinematic viscosity and the thermal diffusion of the corresponding phase (i.e. solid, or fluid; air/liquid). Finally, Θ is the dimensionless temperature defined as $\Theta = \frac{T - T_{\mathrm{ref}}}{T_{\mathrm{ref}}}$. The dimensionless numbers and their expected quantitative range are given as,

$$\epsilon = \frac{1}{\mathrm{Re}^{1/2}} = \sqrt{\frac{\nu}{\Omega D^2}} \ll 1$$
$$\mathrm{Pr} = \frac{\nu}{\gamma} \sim O(10^0) \tag{5}$$

Note that while the flow equations (1)–(3), apply only in the fluid region, the energy conservation equation applies across the entire domain so as to govern the temperature distribution inside the solid and fluid components. Therefore a heaviside step-function is imposed onto the physical properties to account for the property jump across solid–fluid interface,

$$\gamma = \gamma_f H(\mathbf{x} - \mathbf{x}_s) + \gamma_s H(\mathbf{x}_s - \mathbf{x}); \tag{6}$$

$$\gamma_f = \alpha \gamma_{\mathrm{oil}} + (1 - \alpha)\gamma_{\mathrm{air}} \tag{7}$$

where subscripts f and s indicate fluid and solid zones, respectively. In addition, the velocity, $\mathbf{u}$, is expanded in a similar fashion,

$$\mathbf{u} = \mathbf{u}_f H(\mathbf{x} - \mathbf{x}_s) + \mathbf{u}_s H(\mathbf{x}_s - \mathbf{x}) \tag{8}$$

Here, $\mathbf{u}_s$ is the solid body rotation velocity for the gears which at the surface (i.e. fluid/solid interface) becomes $\mathbf{u}_s(\mathbf{x} = \mathbf{x}_s) = 0.5$. Finally, the last term on the right hand side of Eq. (4) represents the total heat-generation per unit volume (i.e. windage heating and contact heating).

The approach for solving this set of equations can be summarized as follows: The temporal evolution of the two-phase fluid mechanics is resolved through direct time-marching of Eqs. (1)–(3) from their initial conditions,

$$\mathbf{u}_f(\mathbf{x}, t = 0) = 0 \tag{9}$$

$$\alpha(\mathbf{x}, t = 0) = H(y - y_0) \tag{10}$$

where y_0 is the initial level of oil within the gearbox. For the thermal field and in the first step of a two-stage process, the following equation is solved at every fluid-dynamic time-step to obtain the intermediate temperature solution, Θ^i, within the solid and fluid components,

$$\mathbf{u} \cdot \nabla \Theta^i = q^i(\mathbf{x}, t) \tag{11}$$

Note that the solution of the energy equation obtained at this step does not have any strict physical definition. The integration over one cycle, however, leads to the time-averaged of the thermal stationary state, $\Theta_0(\mathbf{x})$,

$$\Theta_0(\mathbf{x}) = \frac{1}{\epsilon} \int_0^{\epsilon} \Theta^i dt \tag{12}$$

The second step provides the stationary state of temperature distribution through the solution of the following system of equations over one gear cycle,

$$\epsilon \frac{\partial \Theta^t}{\partial t} + \mathbf{u} \cdot \nabla \Theta^t = \left(\frac{\epsilon^2}{\mathrm{Pr}}\right)\nabla^2 \Theta^t + q$$
$$\Theta^t(\mathbf{x}, t = 0) = \Theta_0(\mathbf{x}) \tag{13}$$

The reader is referred to [1] for further details of the implementation of this approach and the associated limitations and assumptions.

2.1. Boundary conditions

The boundary conditions corresponding to the fluid mechanics are the standard no-slip and impermeability conditions,

$$\mathbf{u}_f(\mathbf{x}_o, t) = 0; \quad \mathbf{u}_f(\mathbf{x}_s, t) = 0.5 \tag{14}$$

where subscript o refers to the outer and gear-surface extend of the gearbox and s corresponds to the fluid/solid interface. The following boundary condition is imposed on the outer extend of the gearbox,

$$\nabla_n \Theta(\mathbf{x}_o, t) = \mathrm{Nu}_{\mathrm{conv}}(\Theta - \Theta_\infty) \tag{15}$$

where n is the surface normal at the boundary and $\mathrm{Nu}_{\mathrm{conv}}$ and Θ_∞ characterize the ambient conditions in which the gearbox is operating. Note that the so-called "thin layer approximation" (TLA) which was proposed in [1] as a means to reduce the computational cost associated with solving the energy equation inside the solid components is not implemented in this work. Rather and given the modest