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Conservative one-dimensional finite volume discretization of a new cavitation model for piston–ring lubrication

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A B S T R A C T

This paper presents a conservative numerical implementation of a new cavitation model that is well suited for lubrication problems with cavitated regions in which the fluid film is attached to just one of the participating surfaces, as happens for instance in piston–ring assemblies. This new model was recently proposed by Buscaglia et al. (2011) and is the first successful attempt at modifying the Elrod–Adams model considering a physically realistic value for the lubricant transport velocity in the incomplete-film region in those cases. In this work we show first the reasons for previous attempts to have failed, which come from a loss of uniqueness of the associated exact mathematical problem. Then, the new model is briefly recalled and a one dimensional numerical implementation by means of a finite volume scheme is presented together with several test-case results.

1. Motivation

The main function of piston–rings is to seal the space between the piston and the liner, acting as slider bearings subjected to alternating motion. These systems have been thoroughly studied before (see for instance [1–5]). Among the many reciprocating components present in internal combustion engines, piston–rings/liner contacts are responsible of an important part of the total power loss due to friction; therefore, it is of great concern whether the friction can be diminished, for instance, by texturing the surfaces with microtextures, that are now a days produced with well defined sizes and shapes using different techniques available in the industry. At the theoretical level, the effect of textures on the performance of lubricated devices is not fully understood. Though some experimental data are available (see, e.g., [6–10]) suggesting that a friction reduction can be achieved, at least in the mixed lubrication regime, numerical studies are more difficult to find (e.g. [11–13]).

The key issue in the simulation of these lubricated devices is the correct treatment of cavitation phenomena by means of incorporating appropriate mass-conserving conditions at the *unknown* cavitation boundaries. By simple inspection of the Reynolds equation, it can be noticed that the phenomenon of cavitation may take place: due to insufficient feeding, due to a positive squeeze (i.e., a transient variation of the gap between the lubricated surfaces) or as a result of a divergent film geometry and consequently at microtextures (microcavitation).

Two models are predominantly used in hydrodynamic lubrication: the Reynolds model and the Elrod–Adams model [14]. The former, easier to implement numerically, though being non-conservative, gives reasonable predictions in many cases and is thus still used in engineering practice. In the Elrod–Adams model, the JFO conditions proposed by Jacobson and Floberg [15] and Olsson [16] are applied at the cavitation boundary to enforce mass conservation. However, due to the highly non-linear nature of the problem, numerical implementations of this model are more prone to numerical instabilities. Implementations of conservative algorithms can be found for instance in [17–19]. The importance of using a conservative model has been shown by means of several numerical examples in [20,21] for problems including transient effects and/or microtextures.

Piston–ring/liner systems need special consideration, however. In the cavitated or non-pressurized region the amount of available oil is, logically, insufficient to fill the entire separation between the surfaces. For the particular case of the piston–ring/liner pair, the lubricant film remains essentially adhered to just one of the lubricated surfaces (the liner), at least far away from rupture boundaries. This is a fundamental difference with respect to other reciprocating components such as journal or thrust

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bearings in which the available oil cannot be thought as attached to one specific surface of the two. This is related to the concept of streamers (see [22,23]). The mathematical and numerical modeling of piston–ring/liner systems becomes thus a challenge, since the already mentioned mathematical models do not account for this fact. Mathematically speaking, the Elrod–Adams model yields a lubricant transport velocity in the cavitated (non-pressurized) region that is half of the physically realistic value in the case of piston–rings. Now, we aim to illustrate this situation.

For this purpose, the two-phase Navier–Stokes (N–S) equations are solved and the results are compared to the Elrod–Adams (E–A) model as done in [24]. A two-dimensional implementation of an interface capturing technique is used to track the material surface separating the lubricant fluid phase from the gas phase and the N–S equations are solved on each phase. The numerical formulation adopted here is the one presented in [25], but, with surface tension effects neglected.

We consider one single ring of parabolic shape (the fix upper surface) and a flat liner (the lower surface) moving from left to right relative to the ring with a constant sliding velocity of 10 m/s as seen in Fig. 1. A viscosity equal to 2×10^{-2} Pa s is used for the lubricant phase and 2×10^{-5} Pa s for the gas phase. Densities are equal to 900 kg/m^3 and 1 kg/m^3. The minimum and maximum separations between surfaces are 6.5 μm and 50 μm, respectively (for additional details refer to [24]). As shown in the figure, at the initial time the fluid film is flat and in contact with the ring just in the central region (thin dashed line in pink color). The film profile evolves from this initial condition and at a later time ($\sim$60 μs) the result is the one drawn with thick continuous line, in blue color for the two-phase N–S formulation and in red color for the Elrod–Adams formulation. The differences are quite remarkable. First, the reformation discontinuity of the film profile (to the left of the minimum thickness point) travels faster to the left in the N–S solution than in the E–A one. Second, and perhaps more important, at the rupture point P (to the right of the minimum thickness point), the N–S formulation predicts a discontinuity of the profile (rupture meniscus) that is absent in the E–A model. Therefore, the size of the pressurized region, and thus the lift and friction forces, will significantly differ from one model to the other. Specifically, the lift force corresponding to the N–S solution is 358.5 N/m and that corresponding to the E–A solution is 247.7 N/m.

Previous attempts at modifying the E–A model so as to improve the agreement with Navier–Stokes results have lead to ill-posed mathematical problems (see [24,26]). A new lubrication model that successfully addressed the problem has been introduced in [27]. The purpose of this paper is to propose a conservative finite volume method for the new model, and to show some of the model's results in non-trivial situations.

By way of outline, after this introduction, the mathematical model and the governing equations for the new model are briefly recalled. After that, the numerical procedure for the one-dimensional case is presented. We restrict ourselves to the case of constant sliding velocity for the sake of simplicity. In the Results section, several problems are presented: first, a case with known exact solution. It consists of a single ring on a smooth liner and is solved to show convergence of the numerical predictions as a function of the grid size. Second, the case of a moving texture on the liner with two rings is simulated. This case is important because the ring upstream can influence the one downstream. Comparisons to the Elrod–Adams model are presented for this case. Finally, an example with a transient applied load is presented, in which the dynamical equilibrium equations governing the evolution of the ring are simultaneously solved with the new model equations.

2. Mathematical model

We consider a domain $\Omega\subset\mathbb{R}^d$ ($d=1$ or 2) divided through a cavitation boundary Σ into two regions: the pressurized region (or full-film region) and the cavitated region (or incomplete-film region) as shown in Fig. 2. In piston–ring/liner systems, the velocity profile in the incomplete-film region is planar as a consequence of having the lubricant film attached to just one of the participating surfaces. In the pressurized region, on the other hand, the velocity profile is linear or parabolic (i.e., a Poiseuille/Couette flow) depending on the pressure gradient.

The two subdomains in which Ω is divided are labeled as Ω_+ and Ω_0 and defined as follows:

$$\Omega_+(t)=\{\vec{x}\in\Omega, p(\vec{x},t)>0\} \tag{1}$$

$$\Omega_0(t)=\{\vec{x}\in\Omega, p(\vec{x},t)=0\} \tag{2}$$

where p is the pressure field. The two regions are coupled through the conservation conditions (sometimes named as the Rankine–Hugoniot conditions) at the cavitation boundary, whose position

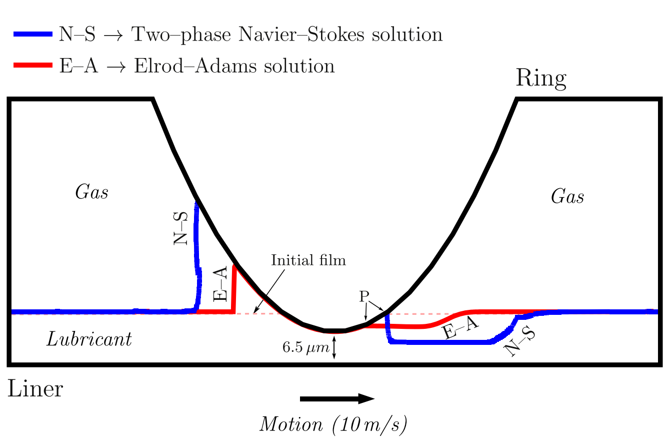

Fig. 1. Comparison of the two-phase Navier–Stokes solution with the Elrod–Adams solution for a single parabolic ring. (For interpretation of the references to color in this figure caption, the reader is referred to the web version of this article.)

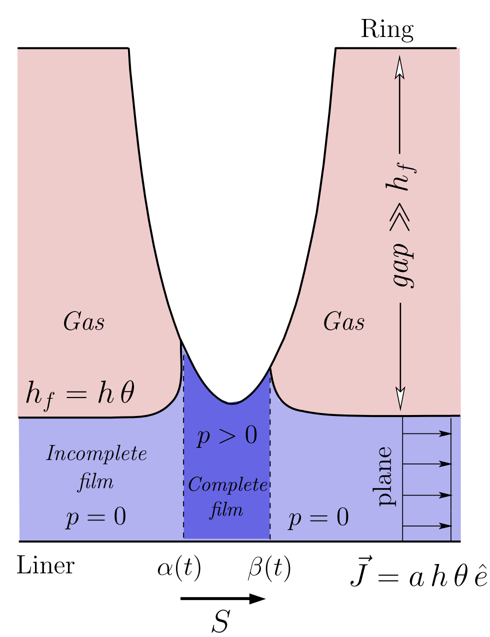

Fig. 2. Problem setting of the piston–ring/liner system.