The Development of A Squeeze Film Damper Parametric Model in the Context of a Fluid-structural Interaction Task

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Abstract. The article considers the work of some squeeze film damper with elastic rings parts. This type of damper is widely used in gas turbine engines supports. Nevertheless, modern analytical solutions have a number of limitations. The article considers the behavior of simple hydrodynamic damping systems. It describes the analysis of fluid-solid interaction simulation applicability for the defying properties of hydrodynamic damper with elastic rings ("alison ring"). There are some recommendations on the fluid structural interaction analysis of the hydrodynamic damper with elastic rings.

1. Introduction
The one of the most common devices to deal with the vibration in rotary systems is a damper that is mounted inside of the support assembly. The classification of different forms and designs for these devices are represented in [1]. In that particular case will be considered several hydraulic damper operational aspects: structural-fluid simulation fidelity on the basis of the flexible part of the elastic ring (figure 1,4). The function of the flexible element is presented by a thin-wall ring that has equiangularly spaced smooth pedestals. Depending on whether are used a single or a pair of rings pedestals will be found on both sides of the ring in a check-board arrange (figure 1,a,2) or on one side but in the same criss-cross pattern with pedestals from another ring (figure 1,b,3).
Figure 1. Damping support with the elastic ring: a – single elastic ring, b – doubled elastic element, 1 – rigid case (elasticity modulus > 1*10^15 Pa), 2 – elastic ring with pedestals on both sides (inside and outside), 3 – elastic ring with pedestals on one side (inside), 4 – flexible part of the damping element, 5 – gap for the oil flow.

Therefor there is always a gap for the oil between ring and between rings and casing (figure 1,a,b,5). The amount of pedestals and other geometry of elements is subjected to [2]. The height of pedestals on two sides could be different. The damping is provided by the squeezing process of the lubricant fluid through gaps. There are many patents (i.e. [3]) related to the design of this type damper. Both present and further investigations in the field of SFDs are highly influenced by the work of Adolfo Delgado and Luis San Andreas the actual head of the rotordynamics laboratory in Texas A&M University [4].

This type of damper is fairly widespread and has many advantages, including small size and ability to be designed with predetermined rigidity and anisotropy, high damping capacity, possibility for the tuning from the resonance frequency. At the same time there are several major deficiencies: the required high precision of tolerance values, angle taper, etc., adjustment on the working mechanism due to the narrow range of existed industrial standard on the elastic ring geometry and the combined action if there are several.

Apart from this IS there is a couple of papers [5,6] on defying the stiffness of these rings. In the articles rings are presented as set of coupled beams attached from their ends. Moreover there is an assumption that in reality there is only a one half of the ring working. In [7] has been studied the effect from dampers on GTE dynamics. To estimate hydraulic forces and reactions from them there are also some analytical research, [8,9], where are mentioned approaches to predict elastic rings flexibility. Thus, in these papers the damping is considered on the basis of theory for fluid oscillations between flat plates that could be found in the S.I. Sergeev monography [9], and the elastic element as a curved beam with the stiffness equals C. In reality the damper is a structure where the rigid part with a complex design (ring with pedestals) is mounted between support casing and the rotor and all gaps are filled with a lubricant. During oscillation of the rotor takes place the mutual dynamic effect between oil and elastic parts of the ring between pedestals. In that case the research of that type of the damper requires the solution of so called fluid-structure interaction task.

2. Hydraulic-elastic interlinked task

2.1 Analytical model

To solve that kind of the task it is required the special damper model that however should be revised by proven analytical solutions. For this reason the model development should be started from the study.
of a single elastic ring section. In the figure 2 is presented an aligned section of the elastic ring. In that case the oil flow in the gap could be considered as flow between two parallel plates, oscillating by the harmonic law. To defy the damping value it is possible to use some analytical expressions from the Sergeev monography [9].

The common scheme is represented in the figure 2. Where $L$ - an overall size of the damping element; $h$ - the initial gap between vibrator and stator; $\zeta$ - harmonic oscillation speed; $b$ - element width, $P(x)$ - pressure distribution during the vibration, $Vx(z)$ - fluid flow speed distribution over the gap height.

The reaction force on the damper vibrator is defined by the expression (1)

$$P = P_l = B \int_{l}^{c} p \, dx = -\mu \Lambda B F \zeta(\alpha, \tau) \dot{\zeta} = -C \dot{\zeta},$$

$$F \zeta(\alpha, \tau) = (1 - \alpha \sin \tau)^{-3}, \alpha = \frac{\alpha}{h}$$

(1)

where $\Lambda = \frac{L}{h}$,

By (1) has been performed several calculations the results of which are represented in figure 3. In figure 3 the vertical axis has a logarithmic scale. According to the diagram it is apparent that with the increase of the gap size (figure 1,5) the reaction force value tends toward zero, and if the gap become smaller the force reaction tends toward the infinity. Therefore the analytical approach is reasonable only for the oscillations with the amplitude value within the size of the gap, i.e. without surfaces contact. Whereas in contrast the interactional numerical computation allows to estimate the damping force even in case there is a rigid parts interaction.

![Figure 2](image_url)

**Figure 2.** The section of a damper with a plain vibrator making oscillations in the normal direction to the stator surface.
Figure 3. The force reaction on the surface of the plain damper defined according to Sergeev analytics for different gap sizes. Other parameters are taken from the table 1.

2.2. Finite-element model
The finite element model is presented by a parametrical 2-way fluid-solid interaction model (figure 4) consists of 4 blocks: solid body part - "Transient Structural"(figure 4,a), fluid body part - "Fluid Flow (Fluent)"(figure 4,b), parameterization block - "Parameter Set" (figure 4,c), solution and results transmission block - "System Coupling" (figure 4,d).

Figure 4. 2-way-FSI analysis block scheme: a-structural, b-liquid, c-structural/liquid coupling, d-parameters

2.3. Principle assumptions and boundary conditions
The model represents an interaction model of a hydraulic elastic task. There are two main parts - mechanical and fluid.

For the solid part have been selected following options:
- fixing of the lower body by the bottom surface in all direction
- fixing the upper surface of the top body in one tangential direction to prevent the sliding
- fixing the upper surface of the top body in another tangential direction by periodic symmetry border to prevent the sliding
- applying the oscillation on the top body by the function $y = a \sin \omega t$
- The global mesh size has the maximum of 1 mm
For the fluid part have been selected following options:

- heat transmission off
- flow stream is laminar
- the gap is closed by walls from two sides
- from two other sides are applied two flow supply borders "pressure-inlet" with the pressure value \( P = 0 \)
- the mesh size has the maximum of 1 mm along the surface and 0,15 mm by height
- the analysis is performed only for several cycles to exclude false results due to the unsteady-state process in the beginning (5 cycles to minimize the processing time)

Both models has borders there they are interact each other. Those are two surfaces from the solid part and two from the fluid (figure 5).

The parameterization is made for all geometry, mesh sizes, fluid body layers and pressure value from both inlets.

2.3.1. Convergence analysis. As long as the model has a lot interacting parts and is in unsteady state the processing time will increase. Has been performed a group of calculations two estimate the acceptable level for mesh size and fluid layers and oscillation cycles number. Initial conditions are listed in the table 1.

![Figure 5. Boundary conditions scheme](image)

| Parameter                          | Value  |
|-----------------------------------|--------|
| Overall length \( L \), mm        | 50     |
| Gap height \( h \), mm             | 0,4    |
| Oscillations amplitude \( a \), mm | 0,03   |
| Oscillations frequency, Hz        | 50     |
| Oil viscosity \( \mu \), kg/ms     | 0,022  |
| Oil density \( \rho \), kg/m³      | 800    |
Figure 6. The dependency of the reaction on the flat damper from fluid layers number: "sup" - stationary lower body surface, "dis" - oscillating upper body surface (figure 5)

The increase in the number of fluid body layers leads to the convergence. The difference of the reaction value for 3 and for 13 layers does not exceed 1.5% (figure 6).

In figures 7 is presented the dependency of the reaction force on the mesh size for different amount of layers.

The decreasing of the mesh size leads to convergence. From the figure 7 it is shown that for 3 and for 7 layers the convergence process looks different, so that for 7 layers the optimal mesh size would be 0.4 mm by the edge length and for 3 layers it is element with 0.75-0.8 mm edge length. During the further mesh size reduction process appears the divergence. The relative difference of the control values in any case is not above 1%. As a result, for the time processing economy was taken a decision to use a 3 fluid layers and 1mm size mesh for this analysis.

Figure 7. The dependency of the flat damper reaction on the mesh size for the case of: a - with 7 mesh layers, b - with 3 mesh layers.

In the figure 8 is shown the dependency of the force reaction on the time. It is clear that during a short period of time (before 0.025 s) the process is unsteady. Therefore for the meaningful comparison with analytical calculations should be considered results from the relatively steady part that is appeared only after 0.03 s. It is also should be mentioned that the total value of the reaction is not steady and cyclically drops to the zero. It means that the comparison should be made for maximum values.
2.3.2. Analysis. With applying parameters that have been received previously is defined the effect of the gap sizes and oscillations amplitude over the force reaction in the damper. For both dependencies processing results have been compare with analytical solution by Sergeev [9]. All of the received data is represented in figure 9.

3. Discussion and Conclusion
The processing results are in a good correspondence with the analytical data. On the basis of this solution it is expected to perform the further analysis including full damper elastic ring models in the context of the 2way fluid-solid interaction task. That is required further preparation by analysis of cylindrical dampers behavior in single-axial and double-axial oscillation, and prism-type.

- Has been developed the flat damper model in the context of interconnected elasticity-hydrodynamic cross-simulation.
- Has been defined optimum mesh parameters for better convergence
- Has been conducted the validation of the developed fluid-solid interaction model by comparison with analytical solution
- Has been defined further steps to study squeeze film damper in the context of a fluid-solid interaction task.

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