STLE 74th Annual Meeting & Exhibition, Nashville, Tennessee, May 1-23, 2019
Control ID: 3076595

On The Effect of Journal Kinematics on the Force Coefficients of a Test Squeeze Film Damper Supplied With an Air in Oil Mixture

TRACK OR CATEGORY

Fluid Film Bearings

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INTRODUCTION

Squeeze film dampers (SFDs) aid to reduce shaft vibrations and enhance stability in rotating machinery [1]. Operation at a high squeeze velocity, the product of an amplitude of rotor motion times a whirl frequency, draws air into the film to make a bubbly mixture that produces notable changes in the SFD force coefficients [2]. During dynamic load experiments to quantify the effect of air ingestion in a test SFD, an air in oil ISO VG 10 mixture of known gas volume fraction (GVF) and low pressure [0.1 bar(g)] is supplied at the damper top end plenum, it flows through the thin film land over a short length, and exits to atmospheric conditions at the damper bottom end. The damper diameter D=127 mm, axial length L=0.36 D, and radial clearance c=0.18 mm. Multiple single frequency loads and impact loads exerted on the SFD housing serve to identify the damper force coefficients (stiffness, damping and added mass). In the tests, the amplitude of damper motion is $r \sim 0.08c$ and the excitation frequency ranges from 20 Hz to 100 Hz, in steps of 20 Hz. The maximum squeeze velocity equals 9.4 mm/s. When supplied with a pure liquid, the SFD shows a direct dynamic stiffness reducing with excitation frequency thus evidencing a significant added mass effect. When supplied with an air/oil mixture, the direct dynamic stiffness increases with frequency, i.e., a hardening effect. The damping coefficient (C) identified from periodic (single frequency) loads decreases monotonically as the GVF increases from 0 to 1, whereas the coefficient estimated from impact load tests first increases with GVF to ~ 0.4 and then drops continuously with a further increase in GVF. The test results demonstrate the kinematics of journal motion affect the force coefficients of a damper operating with air ingestion.

DESCRIPTION OF TEST RIG

Figure 1 shows a schematic view of the test rig adapted from an annular seal vertical rig [3]. The test section comprises a rigid journal and a bearing cartridge supported by a steel pipe atop. The pipe serves both to deliver lubricant into the test section as well as a means to provide structural lateral stiffness to the bearing. Two flow meters record the lubricant and air flow rates. A sparger element (with pore size 2μm) mixes the oil and air streams to generate an air in oil mixture whose gas volume fraction (GVF) ranges from 0 to 1, i.e. all liquid to all gas, at the supply condition. At an operating temperature of 28 ºC, the ISO VG 10 oil has viscosity μ_l =13.5 cP and density ρ_l=830 kg/m^3, whereas air has viscosity μ_a =0.020 cP and density ρ_a = 1.2 kg/m^3 at 1.1 bar(a). Note the large difference in material properties for the mixed fluids; i.e., $\rho_l / \rho_a >> 1$, $\mu_l / \mu_a >> 1$.

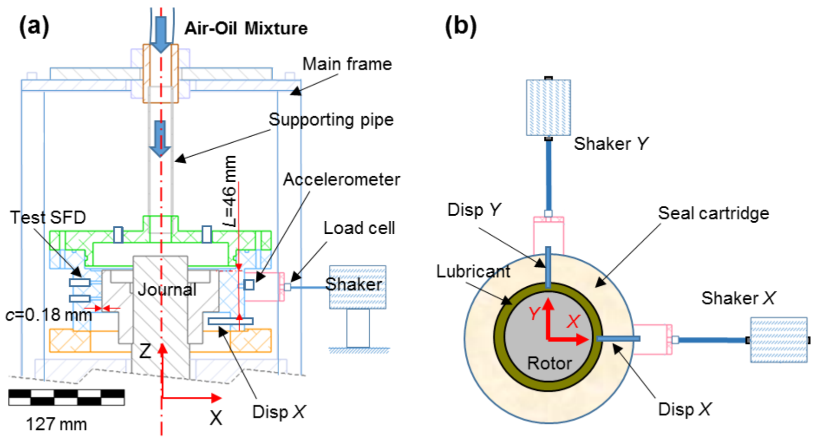

Fig. 1 Schematic view of annular seal/SFD open end test rig; (a) cross section and (b) top view [2].

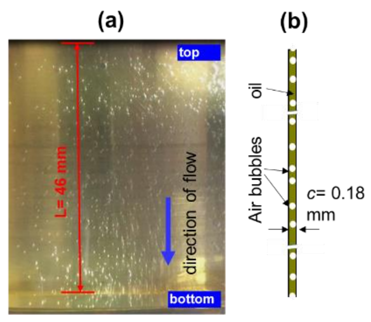

Fig. 2 (a) Photograph of bubbly mixture flowing through squeeze film land. (b) A schematic view showing gas bubbles in the clearance. Inlet gas volume fraction =0.5. Supply pressure P_s= 0.1 bar (g), exit pressure P_a= 0 bar (g).

In the current tests, the mixture flows into the plenum section atop of the SFD section at a pressure of just 0.1 bar above ambient. The flow condition represents a damper with one end flooded and the other end exposed to ambient. Recall the damper axial length, diameter and radial clearance equal to L=46 mm, D=127 mm, and c= 0.18 mm. Figure 2 depicts a photograph of a bubbly mixture with an inlet GVF= 0.5 whose corresponding gas mass fraction = 0.998. The average diameter of a gas bubble is 0.6 mm ±0.2mm, ~ 3.3 c. Note for GVF=0 the flow of oil is 0.2 LPM.

San Andrés and Lu [4] detail the test procedure, data analysis in the frequency domain, and the identification method for extraction of mechanical parameters from the real and imaginary parts of a complex dynamic stiffness H constructed from the transfer function of force/displacement. When the test rig is not lubricated by either oil or a gas in oil mixture, independent dynamic load tests produce the test system structural stiffness K_s = 0.69 MN/m, remnant damping C_s = 0.2 kN s/m, and mass M_s = 7 kg. The *dry* test structure natural frequency $\omega_n = \sqrt{K_s / M_s}$ =50 Hz and its damping ratio $\xi = C_s / \sqrt{K_s M_s}$ ~ 4.5%.

Next, the damper is supplied with either pure lubricant or a mixture on known GVF at the inlet plane. In some tests, single-frequency dynamic loads are exerted on the bearing via two shakers and stingers, orthogonally mounted. In other tests, without the stingers attached, impact hammers deliver loads onto the bearing to excite its motion. Two load cells, four eddy current displacement sensors, and two accelerometers record applied dynamic loads, the relative displacements between the bearing and the journal, and the absolute accelerations of the bearing. A data acquisition system records the sensors' signals at 12,800 samples/second.

RESULTS AND DISCUSSION

For both types of dynamic load tests, the maximum amplitude of bearing displacement (e_u) is kept at 15 μm (= 0.083 c). For tests with periodic loads, along X and Y directions, the excitation frequency ($f= \omega/2\pi$) ranges from 20 Hz to 100 Hz, in steps of 20 Hz. For motions about a centered condition, the SFD element shows identical dynamic direct stiffnesses, $H_{XX}= H_{YY}$, and negligible cross-coupled coefficients, $H_{XY}= H_{YX}$ ~0. The following discussion takes H to represent $H_{XX}= H_{YY}$.

Figures 3 and 4 show the real and imaginary parts of the complex stiffness (H) versus excitation frequency as obtained from unidirectional single frequency loads. The graphs depict the instances of a mixture, each with a known inlet GVF. The symbols represent the test coefficients with bars denoting the standard deviation from five independent tests. For all flow conditions, in Figure 3, Re(H) → 0 as $\omega \to 0$, hence the damper produces no static stiffness coefficient, i.e. K_{SFD}=0. For operation with a pure oil condition (GVF= 0), Re(H_{SFD}) → - $M_{SFD}\,\omega^2$, since the oil having a large density induces a significant added mass coefficient, M_{SFD} ~ 6.4 kg. A predicted added mass $M = \frac{1}{12}\pi\rho\left(\frac{D}{2c}\right)L^3 = 7.5$ kg is just 14% larger than the identified magnitude. For operation with a GFV=0.2, the SFD produces a Re(H) >> 0 and growing with frequency. The result evidences a significant hardening of the squeeze film! Note that as GVF > 0.2 → 0.9, the dynamic stiffness reduces significantly at a given frequency. Even for GVF as large as 0.95, the damper produces a stiffness hardening which would lead to the estimation of a (physically implausible) negative added mass.

Figure 4 depicts Ima(H)$_{SFD}$ increasing with excitation frequency, as expected; while it also decreases with an increase in inlet GVF. It is remarkable that for all flow conditions, either a pure liquid or a mixture, Ima(H) ~($\omega\,C_{SFD}$), where C_{SFD} is a viscous damping coefficient, shown in Fig. 5. Here, C_{SFD} decreases as the GVF increases since the effective viscosity of the mixture decreases. Refs. [3,4] report similar findings for tests with a *wet gas* annular operating with a bubbly mixture. Presently, for operation with pure oil (GVF=0), the short length open ends SFD model predicts $C_= \frac{1}{2}\pi\mu(L/c)^3 D$=44.9 kNs/m that agrees very well with the test data, $C_{SFD=}$42.6 kN.s/m.

Figure 5 compares damping coefficients extracted from single-frequency loads and impact loads. For the tests with periodic loads C_{SFD} drops linearly with an increase in GVF. Supplied with a GVF as large as 0.9, the damper still provides 60% of the damping as if supplied with pure oil. Note that for GVF=0.9, the liquid mass fraction LMF = $\dot{m}_l / \dot{m}_m$ = 0.99! For GVF > 0.9, C_{SFD} drops to a negligible magnitude as the effective viscosity drops quickly since $\mu_a / \mu_l = 1.4\ 10^{-3}$.

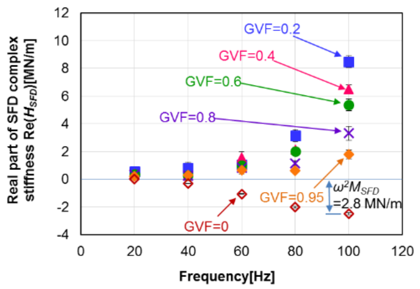

Fig. 3 Imaginary part of SFD complex stiffness (H_{SFD}) vs. excitation frequency (f). Inlet GVF = 0 to 0.95. Supply pressure P_S = 0.1 bar(g) and exit pressure P_a = 0 bar(g).

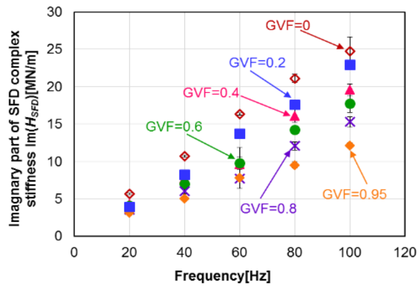

Fig. 4 Imaginary part of SFD complex stiffness (H_{SFD}) vs. excitation frequency (f). Inlet GVF = 0 to 0.95. Supply pressure P_S = 0.1 bar(g) and exit pressure P_a = 0 bar(g).

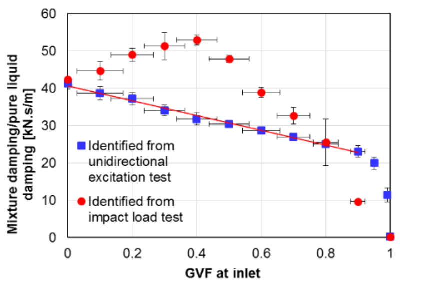

Fig. 5 SFD damping (C_{SFD}) vs. inlet GVF. Derived from single-frequency loads and impact loads. Supply pressure P_S= 0.1 bar(g), discharge pressure P_a= 0 bar(g).

However, as also shown in Fig. 5, impact load tests produce a damping coefficient that actually increases for operation with an air in oil mixture whose GVF increases from 0 to 0.4. At GVF= 0.4 the damping coefficient is 1.3 times that for a pure oil condition. As the GVF increases further towards the pure gas condition, C_{SFD} drops continuously. Recall that in 2001 Ref. [5] presented similar results and discussed the great differences in SFD dynamic load performance, from impacts and periodic loads, induced by the air content in the lubricant. The current results validate the earlier findings and call to attention that journal kinematics alter the physical behavior of bubbly mixtures flowing through a small clearance.

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KEYWORDS

Squeeze film damper, force coefficients, bubbly mixture