NEW PROPOSAL TO THE TURBOMACHINERY RESEARCH CONSORTIUM

A COMPUTATIONAL MODEL FOR INTEGRAL SQUEEZE FILM DAMPERS AND EXPERIMENTAL VERIFICATION

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SIGNIFICANCE

Squeeze film dampers (SFDs) reduce amplitudes of rotor motion and improve rotor system stability while also allowing to isolate mechanical elements [1].Common SFDs have a continuous 360^o film land geometry, integrate either O-rings or piston rings as end seal elements, and are often mounted with an elastic cage, weight bearing, for centering of a rotor. The squirrel cage stiffness can be tuned to place the rotor-bearing system critical speeds away from the operating speed range. However, the cage requires too much length, detrimental for an aircraft engine application.

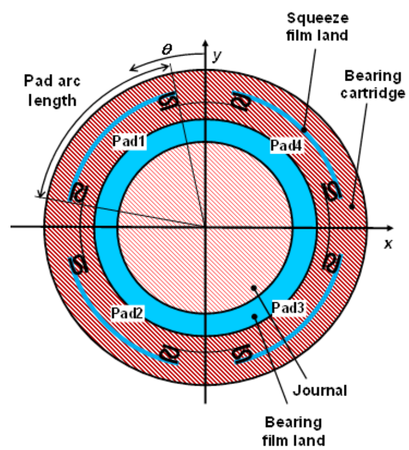

Fig. 1: Schematic view of a 4-pad ISFD (exaggerated clearance**)**

An integral squeeze film dampers (ISFD), as shown on Fig.1, offers the advantage of a lower number of parts, a shorter axial span, a lighter weight, split manufacturing, and higher tolerance precision [2]. An electro discharge machining (EDM) manufacturing process produces individual arcuate damper film lands with *S*-shape flexural springs.

Although ISFDs have been around for over 20 years, their application is limited to certain types of process gas compressors and as a drop-in retrofit to displace system critical speeds away from a certain operating speed range. In this last case, ISFDs offer mainly structural isolation. Application of ISFDs into aircraft engines has taken longer than usual, in spite of their significant advantages. Their time is ripe, however; air breathing engine manufacturers search for ultra-short damper configurations (L/D~0.2), five to seven in a typical engine, as the savings in axial space are a premium to shorten overly long, multiple shaft flexible rotors.

In the mid 1990's, Dr. San Andrés and students [2-3] conducted imbalance response tests on a heavy three-disk rotor supported on ball bearings in series with ISFDs, and demonstrated their capability to produce viscous damping, amplified by the proper selection of end plate seals restricting the axial flow. At the time, Dr. San Andrés developed a proprietary code (SFDFLEX®) for prediction of force coefficients in a segmented pads damper, i.e., an ISFD.

Much later in 2011, Delgado et al. [4] conduct dynamic load experiments with an ISFD having four 60^o arcuate pads with clearance c=0.58 mm, diameter D= 169 mm and length L=44 mm. The S-spring stiffness equals 8.8 MN/m. End plates, extending 9.5 mm and with 25 μm gap, seal the axial sides of the damper lands. The authors find a test damping coefficient at about 50% of their prediction and a large virtual mass coefficient. Note that ISFD manufacturers typically ignore the (significant) virtual mass effect.

In 2012, Agnew and Childs [5] identify the force coefficients of a flexure pivot tilting pad bearing in series with an ISFD, see Fig. 2. This element is used to control the support stiffness towards reducing system critical speeds. The four pads damper (73^o arc length) has D=158 mm, L=76 mm, c=0.356 mm, and end plate seals with a large gap, 2.4 mm. The test results demonstrate the series mechanical device has a lower direct stiffness and added mass compared to those of the tilting-pad bearing alone. The damping coefficient remains practically unchanged, however

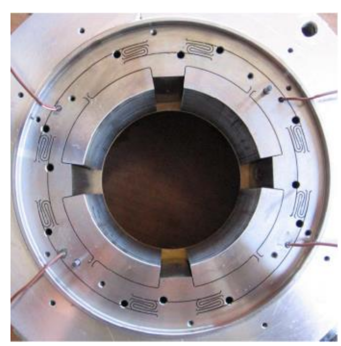
Fig. 2: ISFD & TPJB [5].

In 2018, Ertas et al. [6] present measured and predicted damping and inertia coefficients for an ISFD, with clearance c=560 μm, hosting end plate seals with distinct gaps: 114 μm, 116 μm, 191 μm, and also open ends. The damper, L=56 mm in axial length, has four 54^o arc extent film lands at D_{out}=141 mm, S-springs at the arc ends with stiffness ~220 klb$_f$/in, and connected to four short 19^o *film* lands at

D_{in}=127 mm. ISO VG32 oil at 49°C and 2.4 bar is supplied thru orifices (ϕ=2 mm) at the center of each 54° film land. A finite element model solves the extended Reynolds equation for evaluation of the pressure field under squeeze film actions. As expected, the ISFD damping and inertia coefficients dramatically decrease as the end seal gap increases. The predicted added mass coefficients are thrice the test magnitude whereas the predicted damping coefficient correlates well with the test coefficient. The authors speculate the discrepancy is due to ignoring fluid inertia convective terms. Just as likely, the overly tight end seals not only restrict flow but also trap (incompressible) fluid in the film lands to amplify the test system inertia. [7]

PROPOSED WORK 2018-2019

The main objective is to further analysis and revamp a computational tool, legacy finite element (FE) code SFDFLEX®, for prediction of the dynamic performance of both SFDs[1] and ISFDs. The tool will include:

(a) Integration of elastic S-springs' stiffness for estimation of the static sag due to a load (a fraction of rotor weight) and the (change in) nominal film clearance for each of the arcuate film segments.
(b) Include an end plate seal coefficient based on its geometry (length and gap).
(c) Add sources of constant pressure such as a feedhole (orifice with flow resistance) or discharge hole.
(d) Solve the extended Reynolds equation including temporal fluid inertia effects and deliver (I)SFD force coefficients for (a) small amplitude motions about a static eccentric position, and (b) representative of a whirl orbit with a specified amplitude and over a frequency range. The coefficients will include the effective action of the S-Springs and damper (journal) mass.
(e) Use the tilting pad bearing test rig to accommodate the existing ISFD [5], see Fig. 2, and conduct dynamic load tests to obtain ISFD force coefficients for ready comparison and validation of the model.

The TL research program on SFDS has a long outstanding tradition as it has produced fruitful quantification of SFD forced performance and their integration into engineering practice. The revamped computational program in XLTRC[2] will serve the current needs of TRC members.

BUDGET FROM TRC 2018-2019

Support for graduate student (20 h/week) × $2,400 ×12 months	$ 28,800
Fringe benefits (2.4%) and medical insurance ($422/month)	$ 5,755
Tuition & allowable fees for three semesters	$ 9,886
Rig revamping ($4,000) + ancillary equipment ($759) + paper publication charges ($800)	$ 5,559
Total Cost:	**$ 50,000**

REFERENCES

[1] San Andrés, L., 2012, *Modern Lubrication Theory*, "Squeeze Film Dampers (SFDs)," Notes 13, Texas A&M University Digital Libraries, http://oaktrust.library.tamu.edu/handle/1969.1/93197 [April 5th, 2018].

[2] De Santiago, O., Oliveras, J., and San Andrés, L., 1997, "Imbalance Response of a Rotor Supported on Open-ends Integral Squeeze Film Dampers," Research Progress Report to the TAMU TRC, TRC-SFD-2-97, College Station, TX, May, 1997.

[3] De Santiago, O., San Andrés, L., and Oliveras, J., 1998, "Imbalance Response of a Rotor Supported on Open-ends Integral Squeeze Film Dampers," ASME J. Eng. Gas Turb. Pwr., **121**(4), pp. 718-724.

[4] Delgado, A., Cantanzaro, M., Mitaritonna, N., and Gerbet, M., 2011, "Identification of Force Coefficients in a 5-pad Tilting Pad Bearing with an Integral Squeeze Film Damper," 10th EDF/Pprime Poitiers Workshop, Poitiers, France, October 6-7.

[5] Agnew, J., and Childs, D., 2012, "Rotordynamic Characteristics of a Flexure Pivot Pad Bearing with an Active and Locked Integral Squeeze Film Damper," ASME Paper No. GT2012-68564.

[6] Ertas, B., Delgado, A., and Moore, J., 2018, "Dynamic Characterization of an Integral Squeeze Film Bearing Support Damper for a Supercritical CO_2 Expander," ASME J. Eng. Gas Turb. Pwr., **140**, 052501.

[7] San Andrés, L., Koo, B., and Jeung, S.-H., 2018, "Experimental Force Coefficients for Two Sealed Ends Squeeze Film Dampers (Piston Rings and O-rings): An Assessment of their Similarities and Differences," *Proc. of ASME Turbo Expo 2018*, GT2018-76224, Oslo, Norway, June 11-15.

[1] Note XLTRC[2]® includes too simplistic SFD codes, not updated since the Dissertation of Dr. San Andrés in 1985.