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An Experimental Investigation of the Effects of Air Ingestion on Subsynchronous Damping Coefficients in an Open Ends SFD

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ABSTRACT

Squeeze film dampers (SFDs) utilize a thin oil film between a non-rotating journal and stationary housing to reduce vibrational amplitudes while crossing critical speeds and improve stability. SFDs may experience air ingestion with certain end seal designs and motion amplitudes, leading to a reduction in damping coefficients. Most research into SFDs focuses on circular, centered orbits (CCOs), meaning the applied excitation consists of one frequency component; however, turbomachines frequently experience excitations containing both synchronous and subsynchronous frequency components. These experiments use a dual-frequency excitation with the smaller frequency varying from 9 to 42 Hz and being labeled as "subsynchronous." A higher frequency is held constant at 100 Hz and is labeled "synchronous." One test condition holds the subsynchronous amplitude constant with an amplitude to clearance ratio (r/c) equal to 5%, and the synchronous amplitude ranging from r/c = 5 – 60%. A second test condition sets both amplitudes to equal values ranging from r/c = 5 – 30%. Preliminary CCO tests are performed using the synchronous amplitude range outlined above and a frequency range of 10 – 100 Hz. The 100 Hz tests are used for evaluating the dynamic pressure. All testing is performed using an open ends configuration with a relatively large radial clearance of 279 μm (11 mils). Additional analysis is provided by calculating the average squeeze velocity due to both frequency components and plotting against damping coefficients. The results show a large increase in subsynchronous damping as synchronous amplitude increases, despite the onset of air ingestion. Synchronous amplitude provides the most significant contribution to the squeeze velocity, and the results show that subsynchronous damping increases with significant increases in squeeze velocity.

Keywords: Turbomachinery, squeeze film damper, lubrication, rotordynamics, vibrations.

NOMENCLATURE

A_{ij}	Acceleration Matrix
a	Excitation Amplitude
C_{ij}	Damping Matrix
c	Radial Clearance
D	Damper Diameter
D_{ij}	Displacement Matrix
H_{ij}	Dynamic Stiffness Matrix
K_{ij}	Stiffness Matrix
L	Damper Length
L_{ij}	Load Matrix
M_{BC}	Housing Mass
M_{ij}	Mass Matrix
Q_{oil}	Oil Feed Flow Rate
r	Excitation Radius
v_{avg}	Average Squeeze Velocity
γ	Squeeze Parameter
2π	Pure Oil Film
τ	Excitation Period
ω	Excitation Frequency

Abbreviations

CCO	Circular Centered Orbit

gpm	Gallons per Minute
GVF	Gas Volume Fraction
KCM	Stiffness, Damping, Mass
l.c.m.	Least Common Multiple
lpm	Liters per Minute
r/c	Clearance Ratio
SFD	Squeeze Film Damper

INTRODUCTION

Squeeze film dampers (SFDs) are implemented in rotordynamic systems requiring additional damping as introduced by Cooper in the 1960s [1]. SFDs are commonly found in aircraft engines due to their usage of rolling element bearings with high stiffness and low damping; however, SFDs may also be used in conjunction with fluid film bearings. SFDs are implemented to reduce vibration levels at natural frequencies and to counteract destabilizing forces as discussed by Zeidan and San Andrés in 1996 [2]. Della Pietra and Adiletta [3,4] provide a detailed outline of SFD advancements from the 1960s to the early 2000s. Advancements include further research on the effects of damper geometries and air entrainment/cavitation.

Zeidan and Vance [5] describe five distinct cavitation regimes. The onset of each regime relies on squeeze velocity and SFD geometric features such as clearance and end seals. The first regime describes a 2π-film with no air ingestion or cavitation. The second regime occurs when increased speeds cause negative pressures to fall below atmospheric pressure, causing the formation of an air bubble in the low-pressure region of the damper. In the third regime, as speeds increase further, more air becomes entrained into the system, causing the bubble from the second regime to enter the high pressure region and break up into smaller bubbles, creating an oil-air mixture. The fourth regime describes vapor cavitation which requires the SFD to have tight end seals. Vapor cavitation occurs when the negative pressure in a sealed SFD falls below the vapor pressure, creating a vapor cavity in the low pressure region. The fifth regime occurs under the same conditions as the fourth regime without tight end seals. In this regime, the negative pressure allows for the formation of a vapor cavity. However, once the pressure rises back above the vapor pressure, the cavity collapses. Once this occurs, air is drawn into the damper until the pressure rises above atmospheric pressure. Zeidan and Vance also mention that supply pressure theoretically has no effect on dynamic pressures which directly impact force coefficients other than shifting the onset of cavitation.

In 1998, San Andrés and Diaz [6] showed that supply pressure has a minimal effect on air ingestion at high whirl frequencies due to the high peak to peak pressures relative to the supply pressure. In 1999, San Andrés and Diaz [7] use experimental data to show the effects of air entrainment on the pressure profile of an SFD, identifying a uniform pressure region. In 2003, the same authors [8] provide images form a high-speed camera that show the presence of non-homogeneous gas cavities that coexist with a homogeneous bubbly oil matrix. The authors state that the formation and extent of gas pockets within the uniform and high-pressure zones of the dynamic pressure field is responsible for the reduction of the damping performance. The authors also show that damping forces decrease steadily as the whirl frequency increases. In all their experiments, San Andrés and Diaz use an open-end damper configuration to promote air ingestion. Overall, increasing whirl frequency increases air ingestion resulting in a reduction of the damping capability of the SFD. In 2004, San Andrés and De Santiago [9] investigated the force coefficients in SFDs while controlling the gas volume fraction (GVF) of the oil. The results show a negative highly linear relationship between the GVF and damping coefficients, up to a GVF of 80%.

Jeung [10] shows the effects of large orbit motions (clearance ratios up to 0.76) on the force coefficients in an open-end SFD and proposes an alternative method for estimating the force coefficients of SFDs. These tests reveal that high orbit amplitudes lead to increased direct damping coefficients. Open end dampers are strongly affected by air ingestion; however, the use of end seals can strongly limit the occurrence of air ingestion. Several end seals design can be adopted. In 2019, San Andrés et al [11] compare the performance of piston ring and O-ring end seals for SFDs. Since O-rings are more effective at reducing leakage than piston rings, the O-rings show improved SFD damping performance over piston rings. Since piston rings allow for leakage through slits, some air ingestion may occur with this configuration. In a more recent study, San Andrés and Rodriguez showed that air ingestion can still affect SFDs sealed with O-rings in case of large amplitudes of motions [12]. With end seals installed, vapor cavitation is observed. In 2019, Jeung and San Andrés [13] show that increasing the feed pressure helps eliminate both vapor cavitation and air ingestion for SFDs sealed with piston rings.

Since past experimentation has shown that air ingestion strongly affects the dynamic performances of SFDs, numerical models have been developed to quantify air ingestion and characterize the dynamic properties of SFDs operating outside of 2π conditions. In 2000, San Andrés and Diaz [14] developed an SFD model to predict whether air entrapment is possible based on operating conditions. The paper defines a squeeze parameter, $\gamma = Q_{oil}/\pi D L a \omega$, where Q_{oil} is the feeding oil flowrate, D and L are the damper diameter and length respectively, a and ω are the excitation amplitude and velocity. γ represents the ratio of the oil entering the damper to the volume change due to the squeezing motion in the SFD and can be used to predict the onset of air entrapment. For $\gamma > 1$, air entrapment does not occur since the SFD volume is entirely occupied by oil while for $\gamma < 1$, air entrapment is possible. In 2020, San Andrés and Koo [15] developed a similar model to characterize tightly sealed SFDs with piston rings supplied with a bubbly mixture. Both predicted and experimental damping coefficients don't show a significant change with the air concentration in the supply mixture up to a GVF of 50%. Contrarily, both numerical and experimental mass coefficients decrease significantly as a function of the supplied GVF.

SFDs are often adopted to counteract the destabilizing forces in turbomachines which are usually translated in subsynchronous vibrations of the shaft. The studies cited previously are all carried out considering single frequency excitation for either circular centered orbits (CCO) or statically eccentric circular orbits and do not consider multifrequency excitations. San Andrés and Diaz [16] excite a bubbly oil SFD with sine sweep and impact loads. The damping coefficients obtained for the impact loads increase with the air volume fraction while the sine sweep damping coefficients remain constant in magnitude up to 0.5 of air volume fraction. The tests show that the loading mechanism and journal kinematics affect the shape and motion of the air bubbles, causing a different response in the SFD. Delgado and San Andrés [17] test an open ends SFD supplied with a feeding groove with sine sweep excitations, generating noncircular, multi-frequency motions. The experimental damping coefficients lie between the predicted values for circular centered orbits and small amplitude motions about an eccentric journal position. San Andrés provides comprehensive overviews of SFD research beyond that of Della Pietra and Adiletta [3,4] including the 2000s [18,19].

In this study, a large clearance open-ends SFD is tested with single frequency CCO and with dual frequency excitations. The main objective of the study is to evaluate if and how air ingestion affects the SFD damping capability on the subsynchronous vibration, therefore compromising its ability to suppress destabilizing forces. The SFD is tested with a fixed synchronous frequency component of 100 Hz with variable amplitudes ranging from r/c = 5% to 60%. Subsynchronous frequency components range from 9 to 45 Hz and are operated at either r/c = 5% or at the same amplitude as the synchronous frequency. Force coefficients are calculated for the subsynchronous frequencies to measure the effects of air ingestion caused by the synchronous frequency. Based on the findings of San Andrés and Diaz [6], the air ingestion is expected to reduce the damping performance within the SFD. However, the large motions provided by the main frequency cause higher overall squeeze velocities despite the small subsynchronous amplitudes. These large motions are expected to increase the damping performance as predicted by Jeung [10].

TEST RIG DESCRIPTION

Figure 1 depicts the test rig comprising a rigid, non-rotating journal and an outer housing (damper cartridge). The journal is fixed to a pedestal bolted on a steel table. The damper cartridge is connected to the pedestal using four flexible support rods. A pair of orthogonal shakers attached to the damper cartridge excite the system at prescribed amplitudes and frequencies. This configuration mirrors standard SFD operation since the relative motion is typically transmitted from the journal due to excitations from the rotor/bearing. This mirrored configuration eliminates complexities regarding the proper sealing of shaker attachment points.

A modular journal design allows for testing at various geometries. The journal is attached to the pedestal using a central bolt, which also acts as the anti-rotation mechanism for the test rig. Oil (ISO VG2) is supplied into the journal central feed hole and then enters the damper land through three, evenly spaced, 5.8 mm (0.2 inch) diameter supply holes located at the center of the 25.4 mm (1 inch) damper land. Oil flows through either the upper and lower side of the damper land into basins above and below the journal, where the oil is returned to the storage tank after flowing through a deaerator. Table 1 provides the SFD dimensions.

Figure 1

Table 1

Instrumentation

The damper cartridge is excited by a pair of orthogonally positioned electromagnetic shakers with a load limit of 2200 N (500 lbf). The shaker directions are defined as X and Y, respectively. Load cells attached to each shaker stinger measure the input forces. Accelerometers are attached to the damper cartridge above each shaker connection to provide acceleration readings in the X and Y directions. Two eddy current proximity sensors are threaded through the damper cartridge opposite of each stinger and track the relative displacement between the journal and cartridge in the X and Y directions. Data from the load cells, accelerometers, and proximity probes is used for force coefficient calculations. Six dynamic pressure sensors are set up in two sets of three sensors. Each set has one pressure probe placed over the supply region (P1 and P4), one pressure probe placed over the upper land (P2 and P5), and one pressure probe placed over the lower land (P3 and P6). Figure 2 illustrates the instrumentation layout.

Figure 2

CALCULATIONS

Excitation Model

The experimental procedure involves both circular and dual frequency excitations. A circular excitation is parametrically defined as a pair of two single-frequency sinusoidal excitations where a is the amplitude of excitation, ω is the excitation frequency, and t defines the position within the orbit period where an orbit period is defined as $\tau = \frac{2\pi}{\omega}$.

$$(x, y) = (a \cos \omega t\,,\; a \sin \omega t),\; 0\; \leq t \leq \tau \qquad (1)$$

Dual-frequency excitations follow a similar model to circular excitation by superimposing two separate circular excitations denoted by a subscript. The calculation of the period of a dual-frequency excitation is not straightforward due to the use of two separate frequencies, so a dual frequency period is defined as $\tau = l.c.m.(\frac{2\pi}{\omega_1}, \frac{2\pi}{\omega_2})$. Equation 2 parametrically defines a dual frequency excitation.

$$(x, y) = (a_1 cos\omega_1 t + a_2 cos\omega_2 t,\; a_1 sin\omega_1 t + a_2 sin\omega_2 t), \qquad 0 \leq t \leq \tau \qquad (2)$$

Dynamic Coefficients

The dynamic coefficient calculations follow the methodology outlined by Delgado and San Andrés [17] in 2010 for multiple frequency testing. Load cell, proximity probe, and accelerometer readings are transformed to the frequency domain to calculate the dynamic stiffness, H_{ij}. The dynamic stiffness matrix is calculated as

$$H \begin{bmatrix} D_{xx} & D_{xy} \\ D_{yx} & D_{yy} \end{bmatrix} = \left(\begin{bmatrix} L_{xx} & L_{xy} \\ L_{yx} & L_{yy} \end{bmatrix} - M_{BC} \begin{bmatrix} A_{xx} & A_{xy} \\ A_{yx} & A_{yy} \end{bmatrix} \right) \qquad (3)$$

where D_{ij}, L_{ij}, and A_{ij} are the displacement, load, and acceleration components in the frequency domain, respectively. M_{BC} is the mass of the damper cartridge.

The dynamic stiffness matrix can be approximated using the KCM model through

$$H_{ij} = -\omega^2 M_{ij} + i\omega C_{ij} + K_{ij} \qquad (4)$$

The real part and imaginary parts of this equation are then separated and used to find mass/stiffness and damping coefficients, respectively,

$$Re(H_{ij}) = -M_{ij}\omega^2 + K_{ij} \qquad (5)$$

$$Im(H_{ij}) = C_{ij}\omega \qquad (6)$$

Damping is calculated by subtracting the imaginary dynamic stiffness from a dry test (no lubricant) from the imaginary dynamic stiffness from a wet (lubricant supplied) test and dividing by the excitation frequency as shown in equation 7. A similar process is used for finding mass and stiffness in terms of subtracting out the dry dynamic stiffness term.

$$C_{ij,SFD} = \frac{Im(H_{ij,wet}) - Im(H_{ij,dry})}{\omega} \qquad (7)$$

For a dual frequency excitation, the displacement, load, and acceleration terms are found for the desired frequency.

Two methods can be applied for calculating the damping coefficient. Typically, damping is calculated for a range of excitation frequencies by finding the linear curve fit for the imaginary component of dynamic stiffnesses for the frequency range. This approach is appropriate when comparing damping to excitation amplitude and when damping does not show any frequency dependance; however, for comparison with squeeze velocity, which is dependent on excitation frequency, damping is calculated at each frequency. For this approach, the imaginary dynamic stiffness is directly divided by the excitation frequency, as shown in Eq. 7. Nonetheless, the curve fit for the linear model shows a strong correlation (R^2) coefficient for both the single frequency excitation and the multi-frequency excitation.

Due to the nature of multifrequency excitations, nonlinear effects may arise at larger synchronous amplitudes; therefore, the documented coefficients are based on a time averaged dynamic stiffness calculation by using an array of data points measured over a set amount of time. This is done since the instantaneous reaction forces may vary based on the position in the orbital period; however, the time averaged results provide a reasonable estimate of the reaction forces in the SFD.

Squeeze Velocity

Since excitations consist of two frequency and amplitude components, an additional squeeze velocity analysis helps account for the overall motion in the SFD. For comparison of squeeze velocity to damping coefficients, an average squeeze velocity approach is implemented. The average squeeze velocity is defined as

$$v_{avg} = \frac{\int_0^\tau |a_1\omega_1 \cos(\omega_1 t) + a_2\omega_2 \cos(\omega_2 t)| dt}{\tau} \quad (8)$$

where τ is the period of the excitation as defined in Eq. 2.

EXPERIMENTAL RESULTS AND DISCUSSION

CCO Orbit Force Coefficients

The SFD is initially tested using circular centered orbits (CCO) for 3.0 and 5.7 lpm (0.8 and 1.5 gpm) flow rates. Damping and added mass coefficients are calculated for both flow rates in the orthogonal X and Y directions. Excitation amplitudes range from radial clearance ratios (r/c) of 0.05 to 0.6, and excitation frequencies range from 10 to 100 Hz.

Figure 3 displays the damping coefficients for the CCO tests. Damping coefficients show slight growth from r/c = 0.05-0.4, increasing from ~ 400 N-s/m to 500 N-s/m (2.3-2.9 lbf-s/in) Damping coefficients show steeper growth from r/c = 0.4 to 0.6, rising from ~ 500 N-s/m to 700 N-s/m (2.9-4.0 lbf-s/in). Damping coefficients are slightly greater for the 5.7 lpm (1.5 gpm) flow rates than for the 3.0 lpm (0.8 gpm) flow rate; however, these increments are marginal compared to the rise in flow rate. This result matches the observations from Zeidan and Vance [5] regarding the effects of flow rate on damping coefficients. Identified added mass coefficients are not shown, as the values of up to 2 kg are an order of magnitude smaller than the housing mass of 15 kg and have a large relative uncertainty.

Figure 3

Dynamic Pressure Analysis

The dynamic pressure profiles are examined for each amplitude of the CCO tests to examine the development of air ingestion, which manifests as a flat region between the peaks of the dynamic pressure [5]. The duration in time of this flat region indicates the severity of air ingestion. CCO pressure profiles are examined at 100 Hz, the synchronous frequency for the dual excitation tests, throughout the entire amplitude range of r/c= 0.05 to 0.6 to determine the levels of air ingestion present in multifrequency testing.

Figure 4 provides the pressure profiles for r/c = 0.2, 0.4, and 0.6 at a 5.7 lpm (1.5 gpm) flow rate. The tests show pressure profiles with increasing levels of air ingestion as amplitudes increase. Small amplitudes (r/c = 0.05 - 0.2) develop smooth, sinusoidal pressure profiles indicating low air ingestion levels. From r/c = 0.3-0.4, a flat spot develops prior to the positive pressure peak, indicating an increasing gas volume fraction in the oil due to air ingestion. For large amplitudes (r/c = 0.5-0.6), the flat spot develops into a full flat region at ambient pressure prior to the peak pressure, indicating large levels of air ingestion. Moreover, there is an asymmetry in the dynamic pressure generation between the lower and upper lands of the SFD. The lower portion of the SFD land is open to ambient air and suffers air ingestion at lower r/c. On the other hand, the oil that exits the upper land is collected in a plenum and shows symptoms of air ingestion at larger r/c values. The dynamic pressure recorded during the multi-frequency tests are similar to those recorded during the single frequency tests. The superimposition of the low frequency excitation to the 100 Hz excitation is responsible of a small oscillation of the pressure peaks. However, the development of the flat region due to air ingestion is similar for both sets of tests.

Figure 4

Dual Frequency Testing

Dual frequency tests are performed using a 100 Hz frequency signal superimposed with a lower frequency signal ranging from 9 to 42 Hz. The 100 Hz signal represents a synchronous excitation, and the lower frequency signal represents a subsynchronous excitation. A first set of tests uses a synchronous amplitude range of r/c = 0.05 to 0.6 while the subsynchronous amplitude is held constant at r/c = 0.05 with force coefficients shown in Figure 5. A second test set imposes equal amplitudes for the synchronous and subsynchronous frequency components, ranging from r/c = 0.05 to 0.3 with force coefficients shown in Figure 6. Both test sets are run for inlet flowrates of 3.0 and 5.7 lpm (0.8 and 1.5 gpm).

Subsynchronous force coefficients are calculated using the curve fit approach for the dynamic stiffnesses corresponding to the subsynchronous frequency. The subsynchronous added mass coefficients for the first test case (subsynchronous r/c = 0.05) are similar to those found in CCO testing (<2 kg) and also exhibit a large error.

Subsynchronous damping coefficients show a significant increasing trend with synchronous amplitude for both the subsynchronous r/c = 5% and equal amplitude test cases. Damping for the subsynchronous r/c = 5% case increases from approximately 400 N-s/m (2.3 lbf-s/in) at synchronous r/c = 0.05 to 1000 N-s/m at r/c = 0.6 for the X direction and 1500 N-s/m (8.6 lbf-s/in) for the Y direction. This discrepancy is likely caused by a slight static eccentricity in the Y, causing the damping in the Y direction to be larger than in the X direction. This effect becomes more noticeable at higher synchronous amplitudes. At r/c = 0.6, the subsynchronous damping is between two and three times that of the damping identified from CCO tests.

The damping values for both cases are similar for the overlapping amplitudes in each respective test matrix, increasing from approximately 400 N-s/m at r/c = 0.05 to 750 N-s/m at r/c = 0.3. This result suggests that the synchronous amplitude is primarily responsible for the increase in damping rather than the subsynchronous amplitude.

Figure 5

Figure 6

Damping coefficients for the 100 Hz synchronous excitation are examined to determine the significance of adding a subsynchronous excitation to both the synchronous damping and air ingestion levels. Figure 7 compares the synchronous damping coefficients for multifrequency testing for the subsynchronous r/c = 5% case and the CCO damping coefficients, both at a flow rate of 3.0 lpm (0.8 gpm). The synchronous damping coefficients and CCO damping coefficients are nearly identical, indicating that the synchronous excitation and associated air ingestion is not significantly impacted by the superimposition of a smaller subsynchronous excitation.

Figure 7

Since dual-frequency tests are dependent on two pairs of frequencies and amplitudes, squeeze velocity provides an accurate overview of the total motion associated with a pair of frequency components. Damping is calculated for each frequency rather than through a linear curve fit and is plotted against the average squeeze velocity. Figure 8 shows the subsynchronous damping coefficients plotted against the average squeeze velocity for the subsynchronous r/c=5% case and for the matched synchronous and subsynchronous amplitudes case. The results show clusters representing each synchronous amplitude since these have a much larger contribution to the squeeze velocity than the subsynchronous frequencies. Both test cases show that damping increases significantly with increasing squeeze velocity. Figure 9 shows a zoomed-in view of the r/c=5% subsynchronous amplitude case clusters corresponding to 10%, 30%, and 50% synchronous amplitudes.

For the 10% amplitude cluster, the subsynchronous damping initially increases from approximately 300 N-s/m to 500 N-s/m with increased squeeze velocity (due to the increased subsynchronous frequency component) then flatlines. This indicates that the larger squeeze velocities are contributing to higher damping values; however, small levels of air ingestion prevent the damping from increasing further. For the 30% amplitude cluster, the subsynchronous damping is relatively unaffected by changes in squeeze velocity due to subsynchronous frequency component. The subsynchronous damping ranges from approximately 500 N-s/m to 800 N-s/m, depending on the flow rate and coefficient direction (X or Y). This result suggests that the subsynchronous damping gains due to slight increases in squeeze velocity associated with the higher frequencies is offset by air ingestion effects due to the higher subsynchronous frequency. At the 50% synchronous amplitude cluster, the subsynchronous damping decreases as squeeze velocity increases. The Y coefficients decrease from over 1300 N-s/m to 1100 N-s/m, while the X coefficients decrease from approximately 1100 N-s/m to 900 N-s/m. This result indicates that at larger subsynchronous frequencies, the effects of air ingestion due to the subsynchronous component are large enough to dominate the damping gains provided by the slightly elevated squeeze velocities. Despite the trends within each cluster, the results indicate that damping increments due to increased synchronous amplitudes is much larger than any damping reductions due to any air ingestion associated with the subsynchronous component.

Figure 8

Figure 9

Figure 10 shows the zoomed-in view for the matched amplitude cases for clusters corresponding to synchronous amplitudes of 10% and 30%.

Figure 10

The 10% synchronous amplitude cluster shows the subsynchronous damping initially increases and then remains mostly constant as subsynchronous frequency causes the squeeze velocity to increase. For example, the Y coefficient for the 5.7 lpm (1.5 gpm) flowrate rapidly increases from 300 N-s/m to 500 N-s/m, then remains mostly constant. Compared to the 10% cluster from the r/c = 5% case, the damping coefficients lie within the same range with a similar behavior. However, the initial increase in damping with squeeze velocity appears steeper, and the flatlining effect occurs sooner. This behavior indicates that the higher subsynchronous amplitude for this case (10%) leads to a greater increase in damping due to higher squeeze velocities, but this increased squeeze velocity leads to an earlier onset of air ingestion causing the flatline to occur at a lower subsynchronous frequency.

The 30% synchronous amplitude cluster shows the subsynchronous damping decreases for both flowrates in the X direction and for the 3.0 lpm (0.8 gpm) flowrate in the Y direction. However, the 5.7 lpm (1.5 gpm) data in the Y direction shows an increasing trend. This result is due to a Y static eccentricity leading to higher eccentricity motions of the damper cartridge.

Overall, the squeeze velocity gains due to increases in synchronous amplitude are much larger than the gains due to increases in the subsynchronous frequency. When comparing all points rather than just individual clusters, damping shows a large positive correlation to squeeze velocity and synchronous amplitude, outweighing any air ingestion effects due to subsynchronous frequency excitation.

CONCLUSION

This paper examines a large clearance, open ends SFD operating with a dual frequency excitation. The effects of a large synchronous frequency excitation on the force coefficients of a smaller subsynchronous frequency component are examined.

Air ingestion has been shown to reduce available damping at synchronous frequencies. In rotating equipment, this is an expected and favorable behavior for frequencies above critical speeds, as the force transferred to the bearings decreases. However, previous research has not shown how air ingestion affects the damping at subsynchronous frequencies, which remains critical for system stability. The following conclusions are drawn from the experimental results:

- The damping associated with the subsynchronous frequency component is not detrimentally impacted by air ingestion. The large squeeze velocities attributed to the overall motion of the SFD cause the subsynchronous damping to grow significantly as a function of the synchronous amplitude.
- An average squeeze velocity approach is introduced to help quantify the effects of squeeze velocity on the subsynchronous damping coefficient. An analysis using the overall motion squeeze velocity shows that damping grows as a function of the squeeze velocity. A closer look into individual data clusters representing different levels of synchronous amplitude indicate a slight damping reduction in damping as the subsynchronous frequency increases; however, this effect is overcome by the large gains in subsynchronous damping as squeeze velocity increases.
- The synchronous damping coefficients are compared to the circular orbit coefficients at 3.0 lpm (0.8 gpm). The results show no significant difference between the synchronous damping from the dual frequency testing and the damping generated in circular tests with the same flow rate. This indicates that the added subsynchronous component has no notable effect on the synchronous damping coefficient

These results indicate that while air ingestion reduces damping at synchronous frequencies, providing a desired reduction in bearing force transmissibility, the subsynchronous damping increases as the synchronous amplitude increases. While increased synchronous amplitude leads to higher levels of air ingestion, the larger squeeze velocities associated with the synchronous excitation cause the subsynchronous damping to increase.

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Table 1: Squeeze Film Damper Dimensions

Dimension	Value
Damper Length	25.4 mm (1 in)
Radial Clearance	0.267 mm (0.0105 in)
Journal Diameter	126.62 mm (4.985 in)

Figure 1: Partial cut view of test rig highlighting main components and flow path.

Figure 2: Test rig top and side views highlighting sensor and shaker configurations.

Figure 3: CCO damping coefficient in X and Y directions for 3.0 and 5.7 lpm (0.8 and 1.5 gpm) flowrates.

Figure 4: CCO Pressure Profiles at r/c = 0.2 (top), 0.4 (middle), and 0.6 (bottom) at 100 Hz with a 5.7 lpm (1.5 gpm) inlet flow rate.

Figure 5: Subsynchronous damping coefficients in X and Y directions for 3.0 and 5.7 lpm (0.8 and 1.5 gpm) flowrates for variable 100 Hz (synchronous) amplitude and a constant subsynchronous amplitude (r/c = 0.05).

Figure 6: Subsynchronous damping coefficients in X and Y directions for 3.0 and 5.7 lpm (0.8 and 1.5 gpm) flowrates for matched synchronous and subsynchronous excitation amplitudes.

Figure 7: Synchronous damping coefficients for subsynchronous r/c=5% case vs circular (CCO) damping coefficients with 3.0 lpm (0.8 gpm) inlet flowrate

Figure 8: Subsynchronous damping coefficients vs squeeze velocity for subsynchronous amplitude of r/c = 5% (top) and matched subsynchronous and synchronous amplitudes (bottom). Each cluster is associated with a given synchronous vibration amplitude value.

Figure 9: Subsynchronous damping coefficients vs squeeze velocity for 5% subsynchronous amplitude zoomed in on cluster representing 10% synchronous amplitude (top), 30% synchronous amplitude (middle), and 50% synchronous amplitude (bottom)

Figure 10: Subsynchronous damping coefficients vs squeeze velocity for equal subsynchronous and synchronous amplitudes zoomed in on cluster representing 10% synchronous amplitude (top) and 30% synchronous amplitude (bottom).