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TESTING AND ANALYSIS OF A SIMPLIFIED NONLINEAR HORIZONTAL STABILATOR OF AN AIRCRAFT

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Abstract: The horizontal stabilator (H-Stab) of a fighter aircraft develops freeplay as the control surface mechanism ages. Modal tests on a simplified model of a H-Stab structure were performed to investigate the behavior of frequency and damping ratio of the spindle and bushing interface on different levels of wear on the structural dynamics. Testing on this simplified model successfully revealed critical changes in its dynamic properties.

The H-Stab structure was represented by a rectangular aluminum block (exciter plate), attached to a steel rod as a spindle fixed to a mast. A set of bushings with varying diameters was used at the center of the plate to simulate different levels of freeplay, which was controlled in the order of 0.034° to be comparable with the MIL-A-8870 standard. Steel bushings were fabricated that represent three freeplay cases: tight fit, nominal clearance, and double clearance.

The parameters of the first three modes of the structure were identified for each of the three bushing fits, and each configuration involved testing using two excitation loadings, namely a burst random and a sweep sinusoidal input. Comparing modal frequencies and damping ratios revealed that the spindle and bushing interface demonstrated nonlinear dynamic behavior. Horizontal bending mode showed a clear reduction in modal frequency when going from the tight bushing to the nominal bushing in the order of 3% to 4%. Vertical bending showed a 1% reduction in modal frequency resulting from the nominal bushing compared to the tight bushing and with no further change from the loose fit bushing. This suggest that this particular mode is rather insensitive to the freeplay. Torsional mode displayed a large decrease when going from the tight fit to nominal freeplay bushing, in the order of 15% to 16%. However, going to the loose fit bushing, the reduction of modal frequency is only a further 1%.

In summary, with the spindle freeplay within the MIL-A-8870 standard range, the identified modal frequencies of the H-Stab major modes were observed to decrease with increased spindle freeplay and the type of excitation. These results indicate that tracking the modal parameters of the H-Stab during the aircraft's service may be a promising approach in determining when the structure has undergone excessive wear resulted in freeplay at the hinges.

1 INTRODUCTION

Control surfaces inherently have mechanical freeplay at their hinges. This freeplay has to be carefully managed because it has negative aeroelastic implications. One type of control surface is represented by an aerodynamic surface that is simply supported by a spindle at the root, and is controlled by rotating the orientation of the control surface. Such a mechanism has two freeplay modes. The first is the angular motion from the driving mechanism and the second is freeplay in the spindle due to the space between the spindle surface and bushing. The focus of this investigation is the wear that occurs on the spindle hinge mechanism of the aircraft horizontal stabilator that connects to the fuselage. Maintenance activities related to in-service wear and excessive freeplay is a considerable effort, however spindle wear is particularly expensive to correct.

One potential simple solution to reduce maintenance activities is to relax the freeplay criteria. Aircraft designers often take the standard freeplay allowances described in MIL-A-8870 [1], and do not necessarily investigate what the aircraft can tolerate. As aviation require authorities require evidence to support a relaxation of freeplay clearance, credible analytical models and ground vibration testing are necessary to demonstrate safe freeplay limits. Therefore, in collaboration with the National Research Council, L3Harris initiated a modal analysis and testing on a highly simplified representative structure to identify changes in dynamic properties due to variation in the freeplay of the spindle collar, and mass added on to the wing section. This analysis and testing on a simplified structural mechanism provided relevant data to investigate suitable actions, validate analytical models and provide recommendations to minimize the cost of repair.

The objective is to investigate key parameter using the simplest analysis and test methodology to demonstrate the influence of freeplay on the spindle, primarily modal frequencies and mode shapes, of the simplified H-Stab structure.

2 STRUCTURAL DESIGN

The configuration of the simplified H-Stab was a rectangular aluminum block attached to a steel rod spindle which was fixed to a fixture mast, Figure 1. A set of bushings with varying diameters was used in the inner edge between the exciter plate and spindle to generate different freeplay. The freeplay was in the order of 0.034 degrees to be compatible with MIL-A-8870 [1, 2]. The accurately machined steel bushing was used to simulate freeplay conditions in the order of a few thousands of an inch or a few minutes of angle.

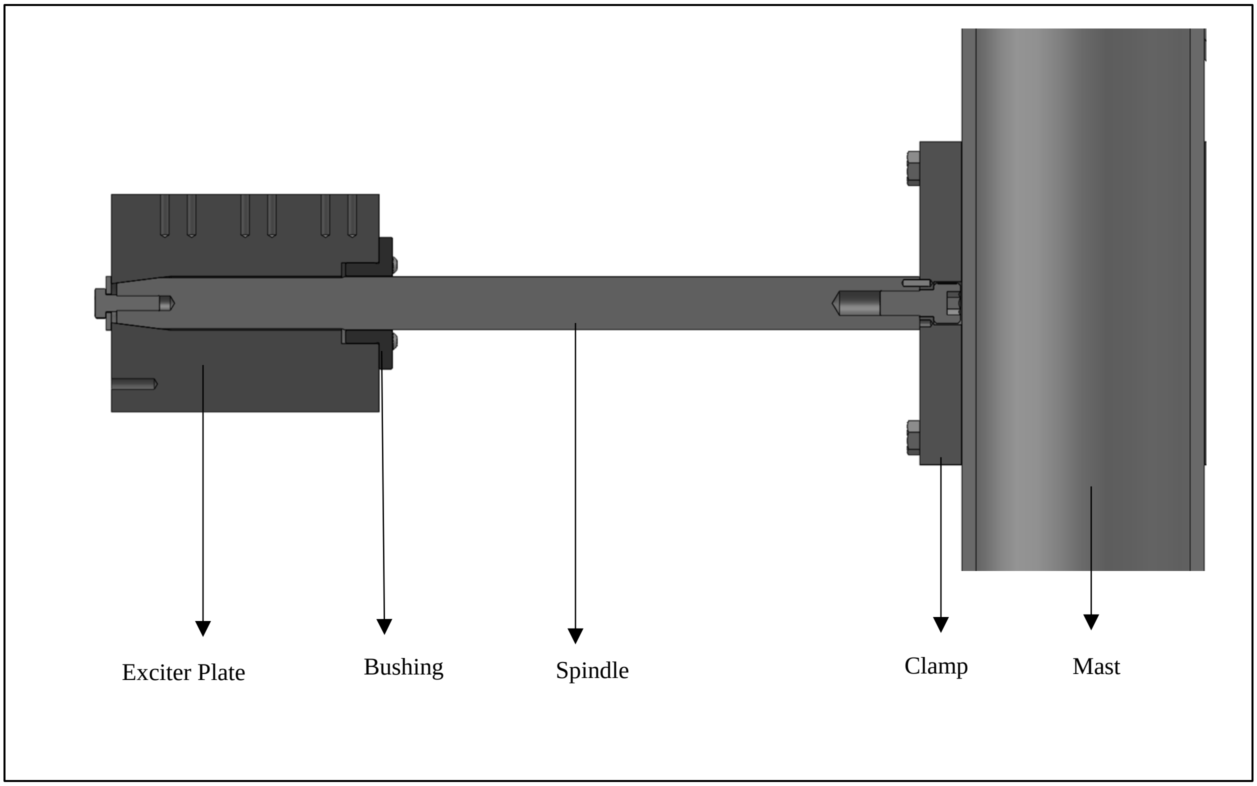


Figure 1: CAD of the Simplified H-Stab Structure

3 FINITE ELEMENT ANALYSIS

A simple finite element model of the structural concept was developed to investigate the modal frequencies and mode shapes of the baseline design. The simplified design is scaled such that the expected modes of interest have well separated modal frequencies and are all under 50Hz.

A three-dimensional finite element model was developed to perform the modal analysis, as shown in Figure 2 and 3. This simplified H-Stab model is composed of two main parts: exciter plate and spindle which had a fixed boundary condition applied at one end. The material properties used are listed in Table 1. The structure was modeled using approximately 205,200 solid elements with six degrees of freedom at each node.

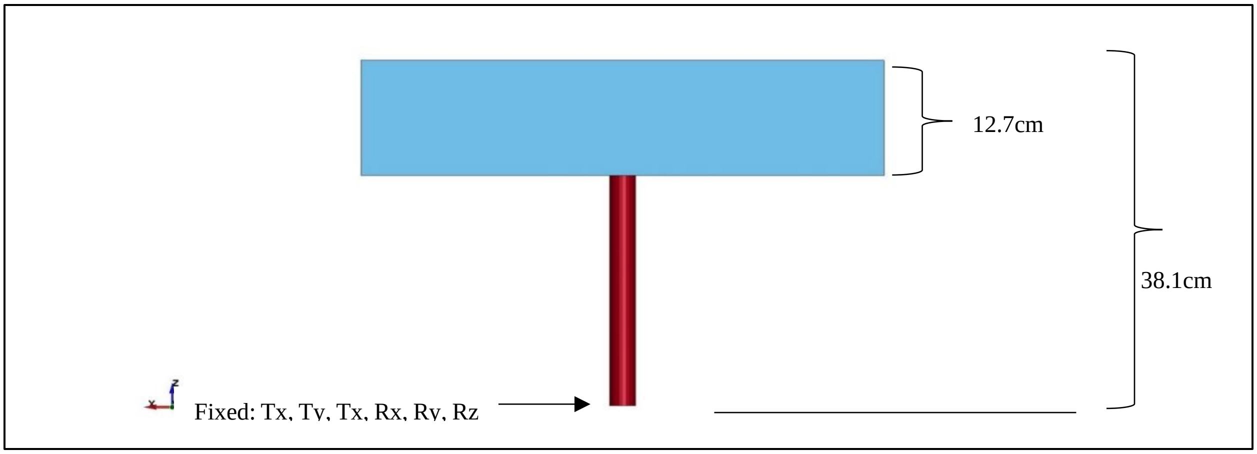


Figure 2: Simplified H-Stab (Top View)

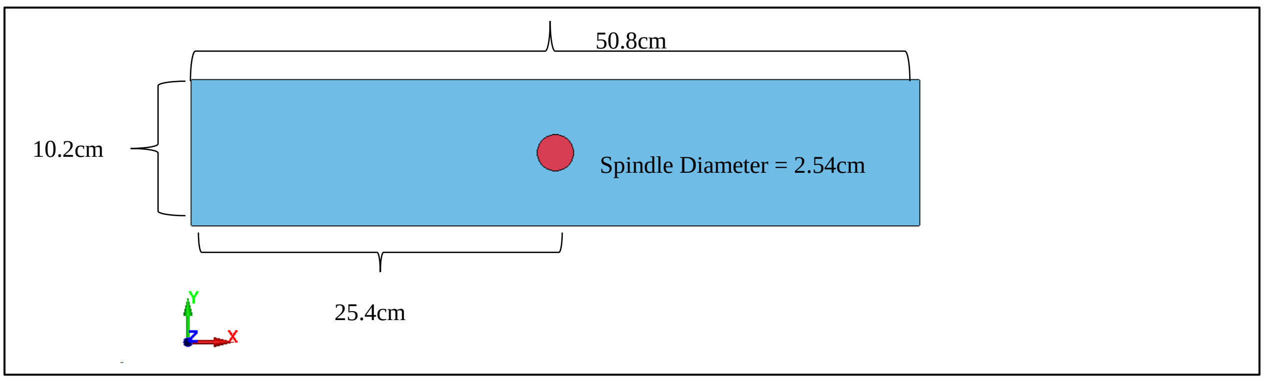


Figure 3: Simplified H-Stab (Front View)

Table 1: Exciter Plate and Spindle Material Properties

Part	**Material Model**	**Density [kg/m^3]**	**Young's Modulus [Pa]**	**Poisson Ratio [-]**	**Mass [kg]**
Exciter Plate	Elastic	2700	6.89×10^{10}	0.33	17.5
Spindle	Elastic	7850	2.96×10^{11}	0.29	1.5

3.1 Modal Analysis

Using the FEA model outlined in Section 3, the natural frequencies of the structural plate were calculated using an eigenvalue analysis. The mode shapes associated with the frequencies were also obtained, as illustrated in Figure 4 through Figure 6.

1) Horizontal bending mode (18.9Hz)
2) Vertical bending mode (22.6Hz)
3) Torsion mode (27.8Hz)

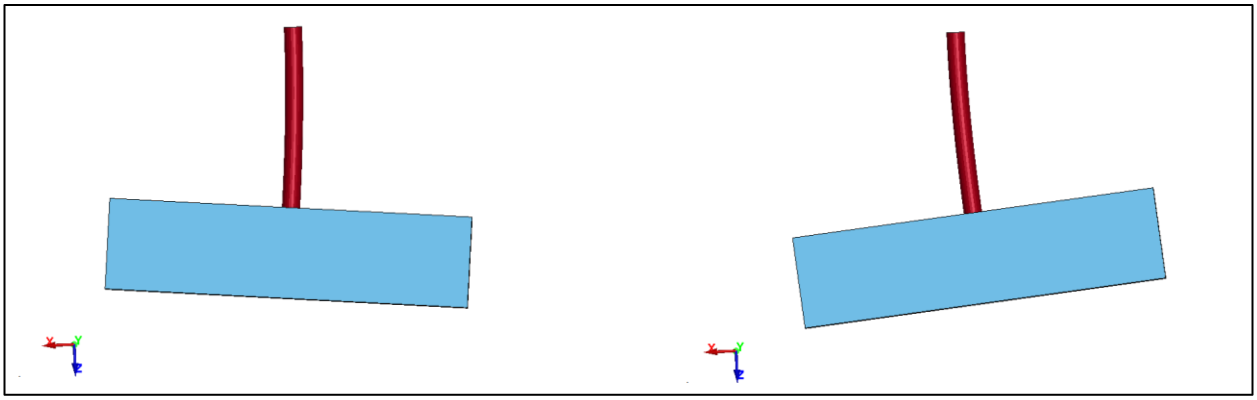


Figure 4: First Horizontal Bending Mode (18.9Hz)

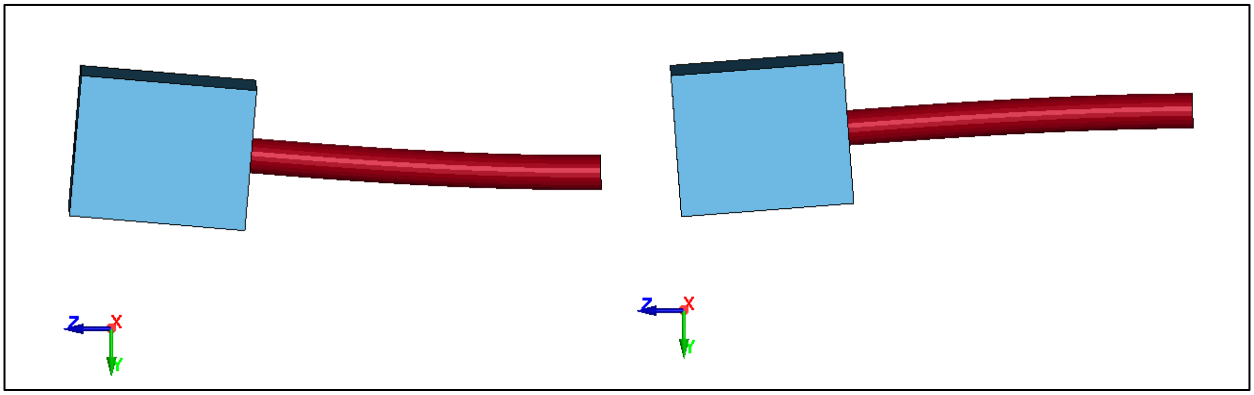


Figure 5: Vertical Bending Mode (22.6Hz)

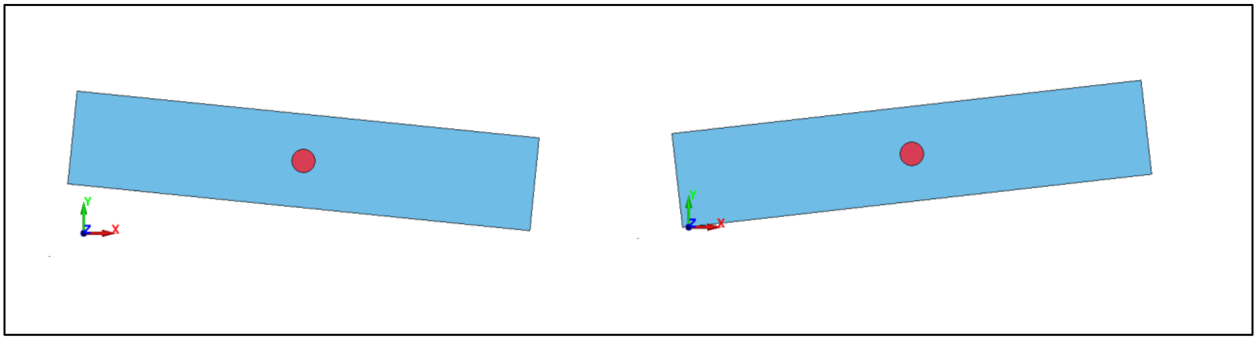
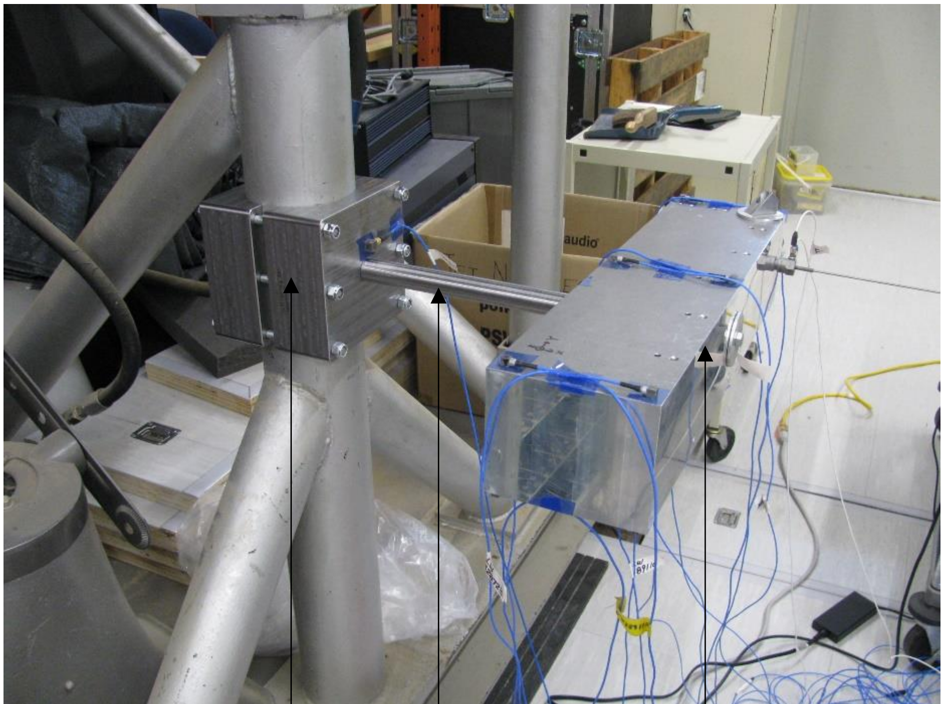


Figure 6: First Torsion Mode (27.8Hz)

4 MODAL EXPERIMENTAL TEST CONFIGURTION

The simplified H-Stab model parts were fabricated, assembled, and attached to a sturdy support structure through a rigid clamp system to simulate the fixed boundary condition. The installed, simplified H-Stab structure is shown in Figure 7.



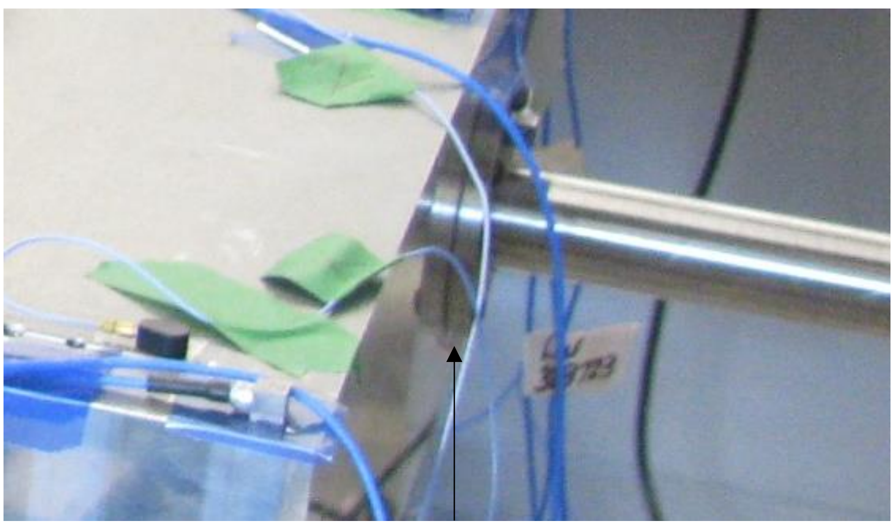


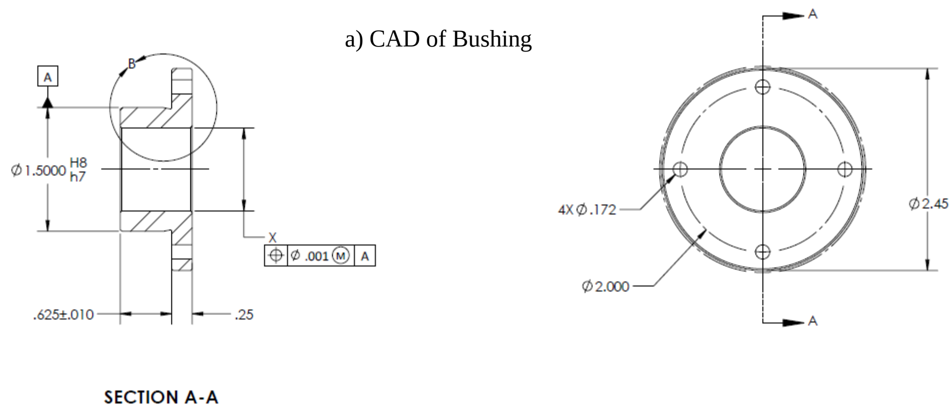
Figure 7: Simplified H-Stab Model Test Set Up

The simplified H-Stab block was attached to the spindle through a bushing mechanism to investigate the behavior of the structure related to the bushing. Three bushings with the same nominal dimension were fabricated, but each was designed with different clearances. The actual dimension of the three bushings were precisely measured before the modal test. Bushing 1 represented a tight fit with the spindle; bushing 2 had a nominal clearance and bushing 3 with double of the nominal clearance, as listed in Table 2. A CAD model of the bushing is shown in Figure 8, along with an image of the actual bushings.

Table 2: H-Stab Bushing and Tolerance

Bushing #	**Clearance**	**Tolerance [in]** *(X in Figure 9)*
1	Tight Fit	0.9847 / 0.9843
2	Nominal Fit (Nominal Clearance)	0.9905 / 0.9895
3	Loose Fit (Double Clearance)	0.9963 / 0.9853

a) CAD of Bushing



b) Aluminum Bushing

Figure 8: CAD Schematic of Bushing used to Mount H-Stab to Exciter Plate

4.1 Experimental Setup

A PCB 2100E11 model shaker was used to provide controlled broadband excitation load to the simplified H-Stab model structure. The shaker was suspended to a sturdy stand using two bungee cables. A PCB Model 288D01 ICP type impedance head was attached to the exciter plate with the flexible shaker stinger rod screwed in tightly to measure the shaker input load and the local acceleration simultaneously. After a few initial trials, it was determined that a stinger input with an orientation of 45 degrees relative to the exciter plate in the X, Y and Z directions was effective in exciting all the H-Stab structural modes of interest.

A total of 14 ICP type PCB 356A01 tri-axial accelerometers were bonded to the simplified H-Stab structure using adhesive in order to measure the responses under the shaker excitation loading. Among them, 12 tri-axial accelerometers were bonded to the exciter plate, and 2 tri-axial accelerometers were bonded to the two ends of the spindle structure. The sensor location and geometrical model of the H-Stab structure is shown in Figure 9.

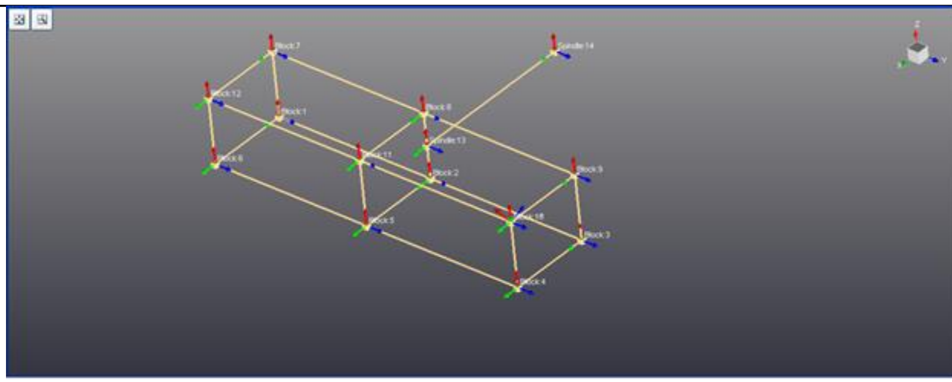


Figure 9: Sensor Locations on the simplified H-Stab Model

A Siemens SCADAS III data acquisition frontend installed with both input and output modules was used to generate the shaker excitation signals while simultaneously recording all accelerometer responses. The sampling rate was set at 800Hz to sufficiently cover the frequency range of interest. The Siemens TestLab R18 software [4] was used to control the shaker input and record the time domain responses during the modal testing. The modal parameters of the simplified H-Stab structure were extracted through post data processing using the POLYMAX Plus module of the TestLab software [5].

To characterize the behavior of the bushing and the spindle fittings on the simplified H-Stab structure, for each test configuration, the structure was excited by a burst random signal and a sweep sinusoidal input signal separately, and at multiple load levels. The modal parameters in each configuration were extracted. By comparing the modal parameters of the identified structural modes, the variation in modal frequencies and damping ratios with comparable input load levels were studied for the various bushing configurations.

A total of 3 test configurations of the H-Stab structure were tested and analyzed, Table 3. In each test configuration, the structure was repeatedly excited by a burst random input, or a sweep sinusoidal input, at multiple load levels. The burst random input covered the frequency range up to 200Hz, while the sinusoidal signal was a log scaled sweep-up signal that ranged from 3Hz to 50Hz.

Table 3: Test Configurations

Bushing	**Shaker Input Orientation**
Tight	45 degrees relative to X, Y & Z
Nominal Clearance	45 degrees relative to X, Y & Z
Double Clearance	45 degrees relative to X, Y & Z

For each test configuration and excitation input condition, the time traces of the excitation load and acceleration responses were recorded in synchronization at the sampling rate of 800Hz. Test results including the Frequency Response Function (FRF) and coherence curves were derived for each test input case. For the Burst Random excitation, a total of 10 averages were used to derive the FRF curves, and the frequency resolution was 0.05Hz. For the sinusoidal excitation condition, a total of 3 averages were used.

4.2 Modal Test configuration

For each of the burst random and sinusoidal input conditions, modal parameters for the simplified H-Stab structure under various input load levels were analyzed through post-analysis. It is important to note that the sweeping rate of the sinusoidal input may have introduced minor modal parameter discrepancies when compared to the burst random excitation conditions, mainly due to the dynamically varied input signal used in the test method. The three major modes of interest for the structure were clearly and consistently identified in all test conditions and all input load levels, as shown in Tables 4-6.

Table 4 Modal Parameters in the Tight Test Configuration

Run 1: Burst Random, Low Level		
Modes	Frequency (Hz)	Damping Ratio (%)
Mode 1	15.24	0.99
Mode 2	16.75	0.80
Mode 3	27.23	1.48
Run 2: Burst Random, High Level		
Mode 1	15.02	1.38
Mode 2	16.67	0.81
Mode 3	26.938	1.54
Run 3: Low Sinusoidal		
Mode 1	14.88	1.34
Mode 2	16.53	0.65
Mode 3	26.58	1.88

Table 5: Modal Parameters in the Nominal Clearance Test Configuration

Run 1: Burst Random, Low Level			
Modes	Frequency (Hz)	Damping Ratio (%)	Freq.Δ Rel.Tight [%]
Mode 1	14.68	1.56	4
Mode 2	16.50	0.67	1
Mode 3	22.87	1.68	16
Run 2: Burst Random, High Level			
Mode 1	14.63	1.31	3
Mode 2	16.47	0.61	1
Mode 3	22.76	1.77	16
Run 3: Low Sinusoidal			
Mode 1	14.51	1.37	3
Mode 2	16.35	0.62	1
Mode 3	22.55	2.07	15

Table 6: Modal Parameters in the Double Clearance Test Configuration

Run 1: Burst Random, Low Level			
Parameters	Frequency (Hz)	Damping Ratio (%)	Freq.Δ Rel.Tight [%]
Mode 1	14.49	1.72	5
Mode 2	16.52	0.52	1
Mode 3	22.57	1.33	17
Run 2: Burst Random, High Level			
Mode 1	14.3	1.25	5
Mode 2	16.46	0.64	1
Mode 3	22.41	1.38	17
Run 3: Low Sinusoidal			
Mode 1	14.23	2.17	4
Mode 2	16.34	0.68	1
Mode 3	22.19	1.68	16

The experimental results highlighted the non-linear nature of freeplay, and the change in frequency is not proportional to the change in diameter. Mode 1 (horizontal bending) demonstrated a clear reduction in modal frequency when going from the tight bushing to the nominal bushing in the order of 3% to 4%, however the additional reduction in modal frequency when considering the loose bushing is only between 1% to 2%.

Mode 2 (vertical bending) appears to be rather insensitive to the bushing diameter. There was a 1% reduction of frequency resulting from the nominal bushing compared to the tight bushing and with no further change from the loose bushing. This suggests that this particular mode is rather insensitive to the freeplay.

Mode 3 (torsional) frequency displayed a large decrease when going from the tight to nominal freeplay bushing, in the order of 15% to 16%. However, when considering to the bushing with the bushing with greater freeplay, the reduction in modal frequency was only a further 1% to maximum of 17%.

Damping ratios throughout the various configurations and test conditions did not vary considerably and did not show a significant correlation to any particular changes in the experimental parameters other the mode in question. Similarly, the type of excitation loading did not substantially change the frequencies or the damping ratios for any of the three configurations.

Within the spindle freeplay range of the MIL-A-8870 standard, the identified modal frequencies of the simplified H-Stab major modes were observed to decrease with increased spindle freeplay as well as the type of excitation. These experimental results indicated that the modal parameters of the H-Stab during the aircraft's service life may serve as a tracking index to identify when the structure has undergone excessive wearing, resulting in loose clearance of the bushing.

The experimental results from this test indicate the challenges in analytically modeling systems with free play. The change in modal parameters obtained from this experiment are not intuitive. For example, there are significantly different characteristics between the vertical and horizontal bending. Furthermore, the large change in the torsional mode is (~15% change in modal frequency vs <5% for the horizontal bending) was not anticipated to such an extent.

5 CONCLUSIONS

Experimental modal tests on the simplified H-Stab structure were performed to investigate the nonlinear dynamic behavior of the spindle and bushing interface related to wearing of hinge mechanisms after extended usage. The hinge wearing process was simulated by three specially fabricated bushings that represented tight fitting, nominal clearance and double clearance conditions, respectively. The modal parameters of the first three modes of the simplified H-Stab structure were identified for the 3 freeplay configurations. Modal parameters were identified using a burst random and a sweep sinusoidal excitation spectrum in each freeplay configuration, and each spectrum included several different input load levels. The key modal parameters of the structure have been identified for the major modes of interest.

By comparing the identified modal frequencies and damping ratios for the corresponding modes in the three different freeplay conditions, it was clearly shown that the spindle and bushing interface demonstrated consistent nonlinear dynamic behavior. In general, the modal frequency of the major modes tended to decrease with increase in freeplay. The extent of the frequency difference varied depending on the actual freeplay boundary condition at the interface and it is not linearly related to the freeplay value.

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