

Ground Vibration Testing of the TiltRotor Aeroelastic Stability Testbed

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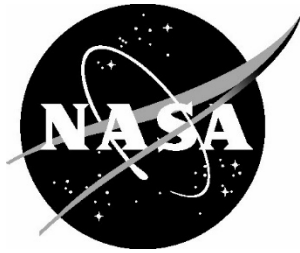
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This report is a summary of the ground vibration testing of the TiltRotor Aeroelastic Stability Testbed (TRAST). The U.S. Army has developed this new tiltrotor model system to validate whirl flutter stability analysis, a critical instability mechanism in this type of aircraft. As such, component-level vibration testing is critical to creating and validating analytical models. This report includes experimental vibration testing results from component level to the full model system. In addition, experimentally measured stiffness values are reported for key components of the model.

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Nomenclature

Model – refers to the physical test apparatus

Analytical Model – refers to the mathematical description of the physical model

TRAST – TiltRotor Aeroelastic Stability Testbed

GVT – Ground Vibration Testing

FRF – Frequency Response Function

TDT – Transonic Dynamics Tunnel

1. Introduction

The U.S. Army DEVCOM Army Research Laboratory (ARL), in collaboration with NASA and DEVCOM Aviation and Missile Center (AvMC) has developed a new tiltrotor testbed for conducting fundamental research on aeroelastic stability, specifically whirl flutter. Whirl flutter is an instability that arises when rotor aerodynamics couple with flexibility in the wing and/or pylon during flight of propeller-driven aircraft and tiltrotors in airplane mode (Ref. 1-3). The stability of whirl flutter is a critical design criterion for tiltrotors as it dictates the wing design to be shorter and thicker than what might be aerodynamically optimal. Due to the critical nature of this instability, which could result in the loss of an aircraft, conservative analyses and large safety margins are typically employed when considering the whirl-flutter boundary. This is partly due to older aircraft being designed with simple aeroelastic analyses using modal approximations for the wing and rotor, and prescribed kinematics for the rotor dynamics. More advanced designs, such as those presented in Ref. 4, will require improved analysis techniques.

As part of the Army's Joint Multi-Role program aimed at reducing the risk of Future Vertical Lift vehicles, there was an initiative to obtain experimental data to validate aeroelastic analyses for tiltrotors. The 8-ft diameter TiltRotor Aeroelastic Stability Testbed (TRAST) was developed as the research wind-tunnel model to obtain this data, as shown in Figure 1. This model is the first new tiltrotor model developed and tested in the Transonic Dynamics Tunnel since the JVX program tested what became the V-22 and derivatives of this 1/5th-scale aeroelastic model that was known as the Wing and Rotor Aeroelastic Test System (WRATS). One of the design objectives was to create a testbed that not only reflects current tiltrotor designs but can also be adapted to replicate new tiltrotor configurations. Consequently, TRAST includes features such as interchangeable hubs, wing extensions/tips outboard of the pylon, interchangeable spars, springs, and tuning masses. Though representative of tiltrotor designs, it was desired that TRAST be a generic configuration without proprietary concerns for sharing validation data.

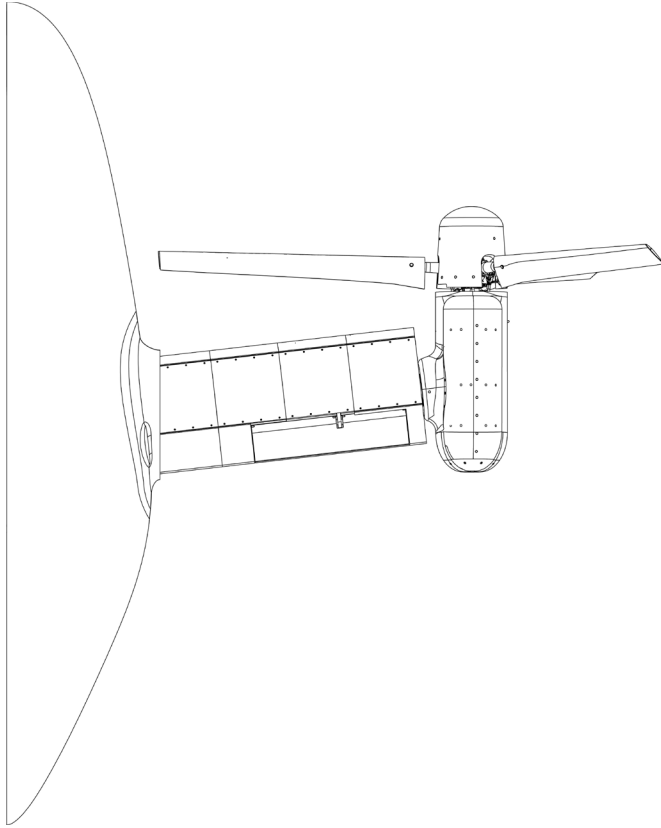


Figure 1. TRAST model in airplane mode, viewed from above.

Ground vibration testing is a crucial component in any dynamic system, particularly for flutter models. Vibration testing allows for a comparison between the analysis and the experimental setup without the additional complication of aerodynamic loads on the model. This ensures that analytical models accurately capture the structural dynamics prior to wind-tunnel testing. This report summarizes the ground vibration testing of the TRAST preceding and subsequent to wind-tunnel test T662, which was the first entry of the model in the Langley Transonic Dynamics Tunnel (TDT), Ref. 5.

1.1 System Layout

Since the TRAST was designed to be used for test-analysis correlation studies, it was designed with a relatively straightforward load path that could be modeled in structural analysis and comprehensive analysis codes. As shown in Figure 2, the rotor is supported by a steel rotor shaft that transmits the torque from the gearbox. The other forces and moments are transmitted to the gearbox housing via bearings at the forward and aft ends of the housing. The gearbox housing serves as the primary load-bearing structure for the pylon with all other pylon components mounted directly or indirectly to it, including the bulkheads, skin panels, control system, and future wing extensions.

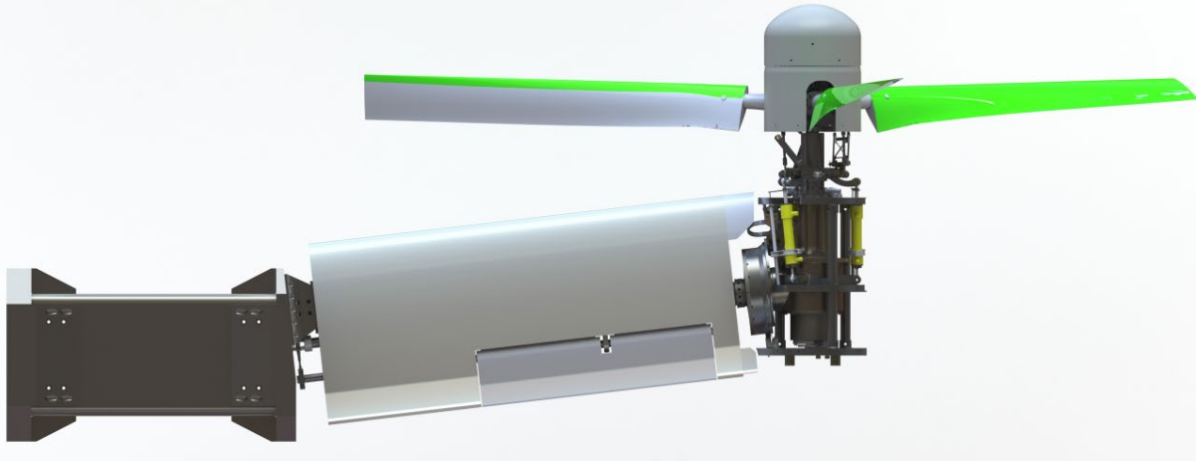


Figure 2. CAD representation of the TRAST without the fuselage fairing and pylon skins.

The gearbox housing is supported by a conversion bearing that permits rotation about the input driveshaft axis, but directly transmits other forces and moments to the wing via a tunable diaphragm spring. Rotation about the conversion axis (pylon pitch) is restrained by a pitch spring (Figure 3) that connects the wing-side of the conversion bearing to the forward pylon bulkhead. The genesis for this component was to simulate the conversion actuator stiffness of a flight vehicle. The conversion actuator would be at its highest stiffness when fully converted to airplane mode and locked onto the “downstop,” which is why previous models often referred to this component as conversion actuator spring but colloquially “downstop spring”. The terms “on-downstop” and “off-downstop” were used to denote the two different stiffnesses. For TRAST, various pitch spring stiffnesses are used that do not correlate to any particular flight hardware. The conversion angle for the pylon can be set by connecting the pitch spring to different mounting holes around the circumference of the conversion bearing, allowing the angle to be set in 10° increments from airplane mode to helicopter mode. The wing-side of the conversion bearing bolts to a six-segment diaphragm spring that transfers loads to the wing and adds tunable flexibility between the pylon and wing.

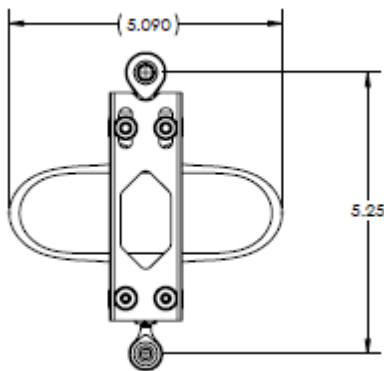


Figure 3. Pitch spring.

The primary wing structure is a carbon-fiber tubular spar cantilevered from a spar support structure that orients the wing to 6.5° forward sweep and 2° dihedral (Figure 4). In the hover cell this support structure is mounted directly to a backstop. When the TRAST is installed in the TDT test section, the support structure connects to the Oscillation Turn Table (OTT) via a strut mount adapter that extends 29 inches out from the wall (Figure 5). In the TDT, a fuselage fairing covers the strut mount adapter and is attached to the adapter so that it pitches with the adapter when the wing

angle-of-attack is changed. Neither of these mounting locations can be considered perfectly rigid relative to the TRAST wing, and each slightly alters the system frequencies by its flexibility.

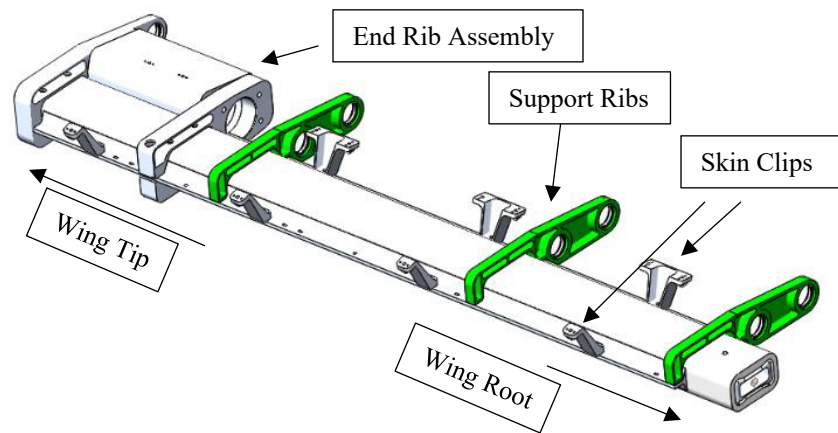


Figure 4. Wing spar assembly, including the spar, support ribs, end rib assembly and skin clips.

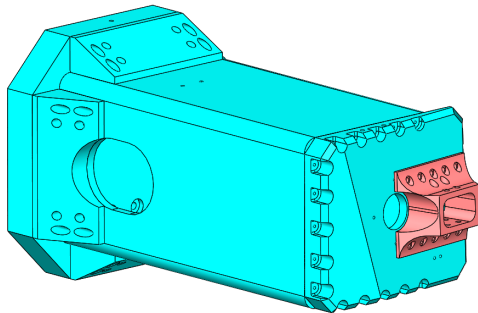


Figure 5. Strut mount adapter with spar support flange.

Aluminum flanges are bonded to the leading and trailing edges of the spar, increasing the chordwise stiffness and providing mounting points for the driveshaft support ribs and skin panels. Since the driveshaft, and hence the conversion axis, runs aft of the wing spar, an aluminum end rib assembly mounts to the outboard end of the spar and provides a structural connection for the diaphragm spring that is offset from the spar axis. The end rib assembly (Figure 6) consists of an end rib, a gearbox support rib, a C-shaped rib channel, and a diaphragm spring (Figure 7) that provides a connection between the wing spar and the pylon. All loads from the pylon pass through the end rib assembly to the wing spar, except for drive-shaft torque. In addition to supporting the driveshaft, the wing support ribs also support an aileron shaft that permits the aileron to be set to a specific pitch angle. Each wing panel is mounted by a pair of chordwise points such that the wing panels do not contribute to the stiffness of the wing.

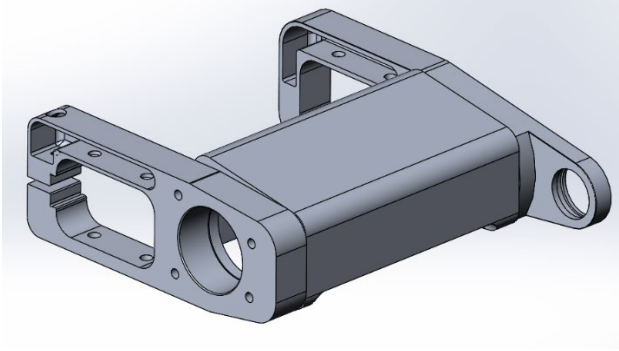


Figure 6. End rib assembly.

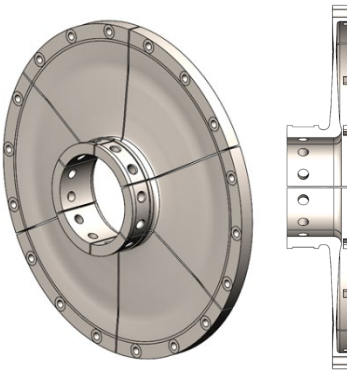


Figure 7. Diaphragm spring.

The rotor is driven by an electric motor that is mounted inboard of the wing support structure. A 5:1 reduction gearbox is mounted directly to the motor output and connects to the driveshaft by a flexible coupling that can be disconnected to permit the rotor to windmill. Between the driveshaft and the input pinion to the pylon gearbox, a short intermediate driveshaft with flex couplings on each end prevents pylon motion from creating large bending strains in the main driveshaft. The pylon gearbox has a 7:3 reduction in addition to orienting the main mast to a 96° incident angle relative to the driveshaft.

The rotor hub mounts to splines on the rotor shaft. The baseline rotor hub for the TRAST is gimbaled, with a constant velocity joint used to transfer torque and internal spherical-bearing surfaces to transfer shear forces (Figure 8). A set of springs is used to establish a tunable amount of stiffness for the gimbaled connection. The spindles extend from the hub along the pitch axis for each rotor blade. A pitch case containing the pitch bearings slides over the spindle and rotates in pitch with the blade. The blade spar fits over the pitch case with a tight fit at the same radial stations as the pitch bearings. Centrifugal load is reacted using a tension-torsion (TT) strap. The inboard end of the TT strap is held by a pin through the spindle. The outboard end is held by a pin passing through both the blade spar and the pitch case, thus allowing centrifugal loads to be transmitted directly from the blade spar to the spindle and torsion loads to be transmitted from the spar to the pitch case. The tight fit of the blades with the pitch case at the pitch bearings results in blade-bending moments being transmitted as shear forces directly through the bearings to the spindles.

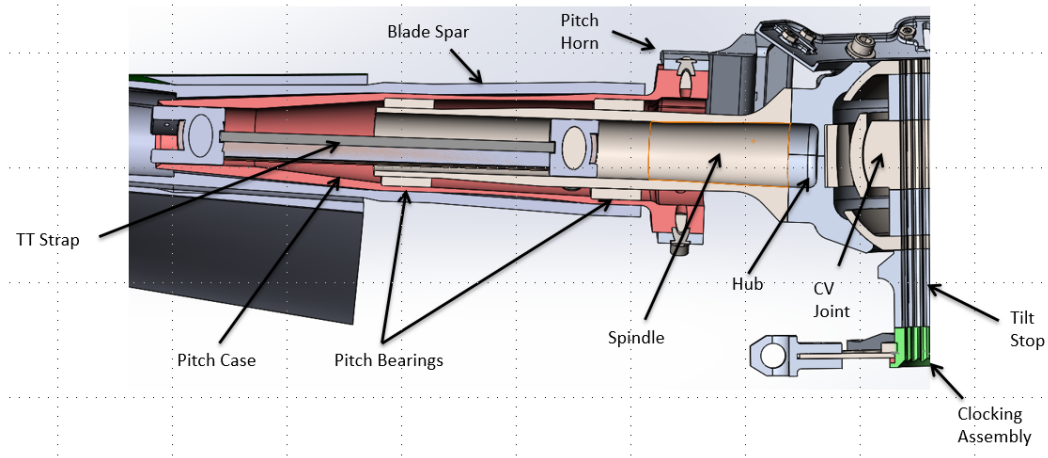


Figure 8. TRAST baseline hub and blade configuration.

Blade pitch is controlled by a rise-and-fall swashplate with pitch links connected to a pitch horn on each pitch case. Flap-pitch coupling, also known as δ_3 , can be adjusted by a combination of spacers at the pitch horn and clocking of the rotating side of the swashplate. This system permits the δ_3 coupling to be changed while keeping the pitch links vertical. In addition, the pitch horn can be swapped from trailing to leading, to induce a positive δ_3 . Swashplate position is controlled by a set of three hydraulic actuators mounted to the sides of the pylon gearbox with 120° spacing.

The vibration testing and results reported in this document are broken down into four major assemblies: wing, pylon, hub, and rotor. Each of these four systems was evaluated separately with careful mass accounting. The wing and rotor blades, which are critical to the dynamics of the testbed, had additional vibration testing and structural stiffness checks compared with finite-element modeling. The structural stiffness checks presented in this document are system-level measured displacements for various static loads and not experimental measurements of component stiffness properties. The pylon is considered to be a relatively rigid system, but it has key features like the diaphragm and pitch spring that add flexibility to its connection to the wing.

2. Weight Tracking

An essential and fundamental check for any dynamic system is to verify that the total system mass is captured. However, in the case of the TRAST system, there are challenges due to the existence of mounting hardware or wires that extend beyond the model. The paper will provide a summary of masses for the major components, along with the best estimate of the total system mass in relation to the model's dynamics. Table 1 presents a summary of the masses for each system, and a more detailed breakdown is provided for the wing and pylon total mass in Tables 2 and 3. The masses reported for the rotor blades include the instrumentation cable, but do not include the blade bolts (0.036 lbm/each) as this was accounted for in the hub mass.

Table 1. Major assembly masses.

Component	Mass (lbm)
Wing	49.174
Pylon	80.248
Hub	8.20
Rotor Blades	1.62/blade

The wing serves as the most logical starting point for weight tracking. The bare wing assembly, depicted in Figure 9, has a weight of 29.42 lbm, encompassing components permanently attached to the spar, like the root flange, end rib

assembly, and driveshaft support ribs. For the complete wing assembly, including all components listed in Table 2, the total weight amounts to 49.174 lbm.



Figure 9. Mass of the bare wing assembly.

Table 2. Mass of the full wing assembly.

Description	Weight (lbm)	Running Mass (lb/in)
Wing Spar Assembly (Figure 9)	29.42	
Aileron Shaft with snap ring	0.920	
Drive Shaft Assembly	3.176	
Flange bolts and washers (10)	0.560	
Diaphragm spring (6)	3.240	
Diaphragm to wing screws (12)	0.272	
Diaphragm spring keys (18)	0.072	
Accel Mounting Block	0.020	
Aileron (without shaft)	1.080	
Hydraulic Hose	1.776	0.035
Instrumentation cables	2.030	0.040
Skin Panel 1	2.016	
Skin Panel 2	1.784	
Skin Panel 3	1.492	
Skin Panel 4	1.316	
Wing Total	49.174	

The pylon mass measurements include a significant number of parts not listed individually. However, most assemblies have undergone validation through CAD or FEA analysis. The major pylon assembly is comprised of the gearbox, bulkheads, control system, and the other components shown in Figure 10. The measured mass for this assembly is 61.35 lbm, which includes hydraulic hose caps removed during installation but is missing the mass of a small amount

of hydraulic oil. Another measurement of 79.99 lbm was obtained for a more complete assembly, and Table 3 shows how this compares when the measured mass of the missing components is added to the mass of the subassembly. The small discrepancy between the two measurements is likely from changes in hydraulic or gearbox oil, since these measurements were obtained on different dates. The addition of the pylon skins and spinner results in a total pylon and hub mass of 88.448 lbm, which corresponds to the sum of the hub and pylon masses presented in Table 2.

The hub assembly includes the rotating and non-rotating parts of the gimbal, along with all components mounted above the gimbal, such as the CV joint, gimbal housing, gimbal springs, spring tower, spindles, pitch case, blade bolts, instrumentation, and wiring, with a total weight of 8.20 lbm. Note that this does not encompass the spinner (2.23 lbm) and spinner arms (already included in the bare pylon weight), which have been accounted for separately.

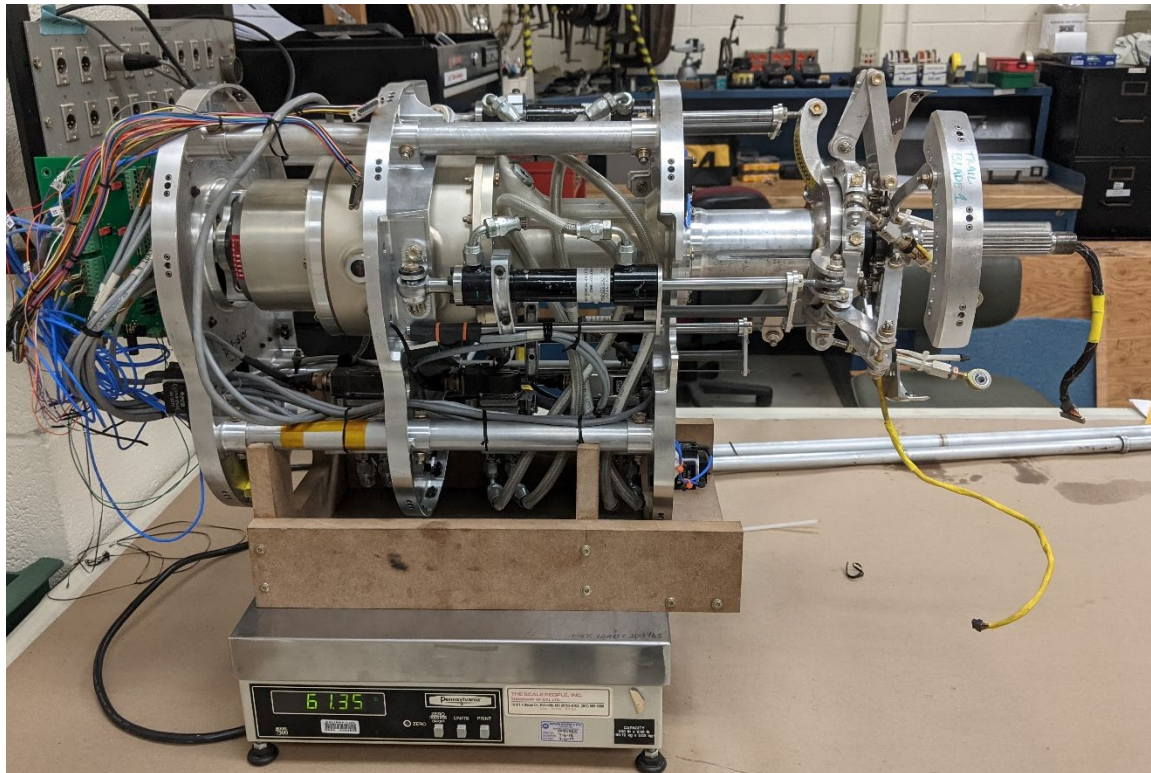


Figure 10. Bare pylon assembly weight.

Table 3. Pylon mass.

Component	Mass (lbm)
Bare Pylon Measured (Figure 10)	61.35
Hydraulic caps	-0.11
Missing Hyd Oil (air in lines)	0.167
Conversion Bearing plus Lug	8.500
Pitch Spring, 8k	0.778
Brake with Cable	1.010
Hub Assembly	8.200
Mast Nut	0.062
Subassembly Weight Check	79.900
Pylon Skin Total	6.318
Spinner	2.230
Total	88.448

3. Blade Testing

Validation data for elastic analytical models of the TRAST rotor blades was obtained through a series of tests. First, free and cantilevered frequencies were measured for each blade, and subsequently, a free-free full rotor test was conducted. The blades are referenced based on their tip color as White, Blue, Red and Green. For the wind-tunnel test, White, Blue and Red were installed as blades 1, 2 and 3, respectively.

3.1 Free-Free Blade Testing

To achieve a free boundary condition, each blade was suspended with a thin flexible line through the blade-bolt hole and excited by using an instrumented hammer for both the flapwise and chordwise hits. Four accelerometers were strategically positioned on the blade and repositioned depending on the direction of the hammer impact. For flapwise hammer strikes, the four accelerometers were oriented in the flapwise direction: three spaced spanwise along the quarter chord and one at the trailing edge to capture blade torsion. Conversely, for chord hammer strikes, three accelerometers oriented in the chordwise direction were installed along the blade span, and one accelerometer was oriented in the flapwise direction.

Natural frequencies for each blade were derived from the measured Frequency Response Functions (FRFs) using the Rational Fraction Polynomial-Z modal identification algorithm. This technique takes advantage of the multiple accelerometers and hit locations to produce accurate estimates of the modal parameters. Modal estimates were produced separately for the flapwise and chordwise accelerometer configurations. For modes where a robust modal response was produced by both the flapwise and chordwise hits, the estimated frequencies were averaged. The resulting natural frequencies for each blade are presented in Table 4. Modal damping was between 0.65 and 0.95 percent for all modes.

Table 4. Free-free rotor blade frequencies from vibration testing.

Mode	Blade Frequencies (Hz)			
	White	Blue	Green	Mean
Flap 1	46.08	46.60	44.74	45.81
Flap 2	118.1	120.9	117.2	118.7
Chord 1	195.2	202.1	196.7	197.4
Torsion	236.0	233.9	231.8	233.9
Flap 3	241.2	247.5	243.7	243.4
Flap 4	386.5	390.1	382.6	386.4
Torsion 2	436.5	432.6	429.1	426.4

3.2 Cantilevered Rotor Blade Testing

Cantilevered vibration measurements serve as a crucial validation method for the TRAST rotor blades' analytical model, as this test condition isolates the rotor blade from the rest of the wing and pylon system while producing similar modal deformations. Due to the lack of an ideal location for a cantilevered constraint on the TRAST rotor blades, an aluminum dummy pitch case was manufactured to secure the blade. The dummy pitch case features a flange that is bolted to a T-slot floor fixture (Figure 11), effectively creating a cantilevered configuration for the blade. In this setup, the blade is attached to the dummy pitch case using a single blade bolt, similar to its installation on the hub. However, as there is no TT-Strap in this test configuration, torsional rotation is constrained by the blade bolt. Forces and moments are transmitted from the blade to the dummy pitch case through the two bearing surfaces, mimicking the actual pitch case behavior. Considering the relatively significant stiffness of the TRAST rotor blades, it is essential to include the dummy pitch case in the analytical model, rather than treating it as rigid.

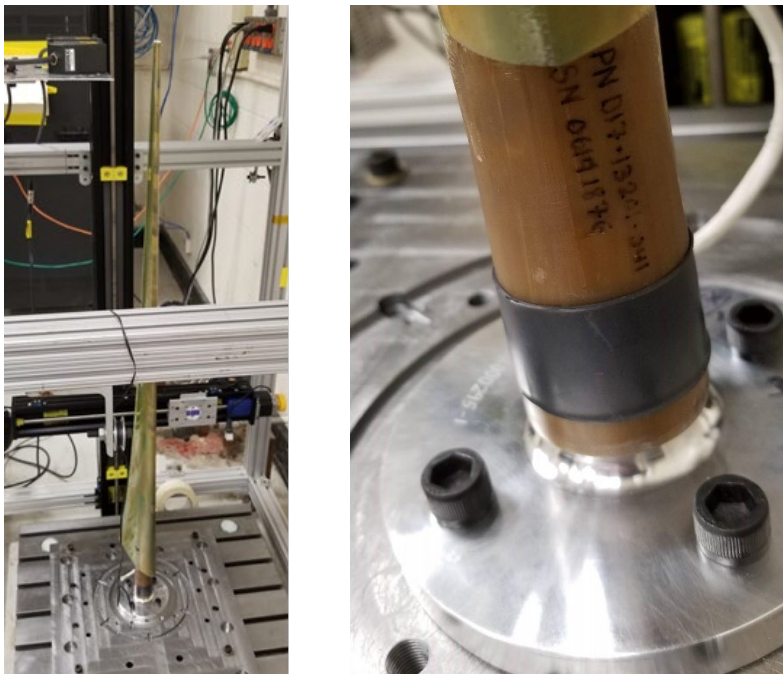
For the cantilevered vibration measurements, three accelerometers were attached to the blade at positions 12.0 inches from the dummy pitch case flange, corresponding to approximately 30% of the full radius. Two accelerometers were oriented in the flapwise direction, located at 1/8 and 3/4 of the chord. The third accelerometer was positioned at the same radial station but oriented in the chordwise direction. Additionally, two laser-displacement sensors were used to measure flapwise motion approximately 0.5 inches from the blade tip.

Experimental measurements of the blade frequencies and mass are presented in Table 5. In general, the flap frequencies were acquired during the flapwise hammer strikes, and the chord frequencies were acquired during the chordwise hammer strikes. However, since the blade is highly twisted a strong coupling is exhibited by some modes, especially the 3rd flap and 1st torsion frequency. Since the Green blade stood as an outlier, being 0.02 lbf lighter and having a 0.5 Hz lower first flap frequency, it was designated as the backup blade.

Additional testing was performed on each blade to assess the influence of the blade bolt and nut torque, due to inconsistencies observed during testing. Due to the geometry of the connection between the blade and pitch case, the blade bolt transfers all of the loads through shear, and the system response should not be affected by the nut. However, this testing showed that when the blade bolt and nut were not properly torqued the FRFs exhibited structural peaks that were skewed and not ideally shaped. Although the modal frequency estimates did not significantly change with bolt torque, torquing the nut to a specified value was found to be necessary for consistent modal parameter estimation.

Table 5. Experimental measurements of the blade cantilevered frequencies (Hz).

	White			Green			Red			Blue		
Mode (Hz)	snug	no-torque	25in-lb	snug	no-torque	25in-lb	snug	no-torque	25in-lb	snug	no-torque	25in-lb
Mass (lbm)	1.622			1.606			1.62			1.626		
1 st Flap	15.4	15.2	15.3	14.9	14.8	14.9	15.6	15.6	15.6	15.5	15.3	15.4
1 st Chord	32.8	32.2	32.3	32.6	31.6	32.8	32.8	32.8	32.9	33.7	33.2	33.5
2 nd Flap	62.7	61.3	62.0	59.6	60.6	60.8	63.4	63.3	63.4	63.2	61.8	62.5
3 rd Flap	144.6	138.7	142.6	136.6	135.2	136.8	145.3	144.3	145.3	144.7	139.7	143.0
1 st Tors.	170.9	167.9	169.2	165.9			169.8	170.0	169.7	168.6	167.0	167.0

**Figure 11. Cantilevered testing of the rotor blades.**

3.3 Static Loading of the Rotor Blades

Blade deflection was measured in conjunction with static loading applied to the blades for strain-gage calibration. To obtain a set of static load-displacement responses for analysis validation, laser-displacement sensors were used to measure displacement while applying the static calibration loads to the blade. However, accounting for the blade's substantial built-in twist and continuously changing airfoil during loading presented a challenge. To address this, a loading cuff was designed and constructed, which was mounted at the blade's tip but aligned to the 75-percent-radius blade cross-section. The local twist angle is defined as being 0° at the 0.75R station for a collective of 0° . Moreover, the loading cuff accommodates the 1° sweep, resulting in a center line of the load fixture offset by 0.2c (0.906 inches) from the leading edge and pitched upward by an angle of attack of 6.61° with respect to the tip airfoil.

The blade was affixed to the same dummy pitch case used for obtaining the cantilevered frequency measurements (Figure 12). Therefore, any comparisons to these results will necessitate analytical modeling of the dummy pitch case in place of the actual hub hardware. The load was applied at the tip of the blade, specifically at blade station of 47.75 inches. Load cells were placed in-line with the load fixture to account for pulley friction and reduce error, rather than relying solely on dead-weight values. The maximum calibration load applied on the blade was approximately 5 lbs, sufficient to deflect the blade beyond the laser range. Any deflection outside of this range was disregarded in determining the load-displacement slope. To induce a torsion moment on the blade, a pair of loads normal to the chord line and separated by 11 inches were applied as a force couple. The measured displacements from the static loading are presented in Table 6.



Figure 12. Static loading of a rotor blade.

Table 6. Displacement measurements for the blades.

	Flap Loading	Chord Loading	Torsion Loading	
Blade	Flap Displacement/Fz (in/lb)	Chord Displacement/Fx (in/lb)	Flap Displacement/Fx (in/lb)	Rotation/MY (rad/in-lb)
White	0.215	0.0359	N/A	0.00146
Green	0.218	0.0360	-0.0243	0.00149
Blue	0.208	0.0358	-0.0255	0.00154
Red	0.207	0.0357	-0.0214	0.00142

3.4 Free-Free Rotor Testing

The free-free and cantilevered rotor blade vibration testing assess the natural frequencies of the rotor blade under two distinct boundary conditions. However, prediction of the blade modes, especially asymmetric chord modes, heavily relies on accurate analytical modeling of the connection between the blade and the hub. Thus, frequency measurements were also desired for the fully assembled rotor, in addition to the individual blade measurements discussed above. At the root of the hub, three components overlap: the blade, the pitch case, and the spindle. When assembled, a multi-load path occurs, in which the flapwise and chordwise-bending moments from the blade are reacted by a pair of high points on the pitch case. These points align with the pitch bearings that interface at the pitch case and the spindle (Figure 8).

To measure the rotor frequencies independent of the wing modes, the hub and blade assembly was suspended from a single soft spring (Figure 13). Separated from the control system, a mechanism was needed to control blade pitch, a critical variable for the rotor frequencies. To achieve this, a 3D printed fixture was fabricated to hold the blades at the correct pitch angle using nylon rods. Manual adjustment of the blade pitch was performed through a pair of nuts to achieve the desired collective pitch. Hammer and shaker testing were conducted in 10° collective increments. While the shaker was better at exciting the symmetric modes, the hammer excelled in exciting the asymmetric modes. Each blade was instrumented with two single-axis accelerometers at approximately 80% radius, oriented in the flapwise and chordwise directions. In addition, a single triaxial accelerometer was mounted at the center of the hub. For each collective, the hammer hits were applied to each blade in both the flapwise and chordwise directions. The results are compiled and presented as frequency versus collective in Figure 14, with tabulated data available in Appendix A. It is important to note that at 50° collective, there is a frequency crossing between the asymmetric flap and the symmetric lag modes, making precise mode resolution challenging.



Figure 13. Free-free rotor test with the shaker setup.

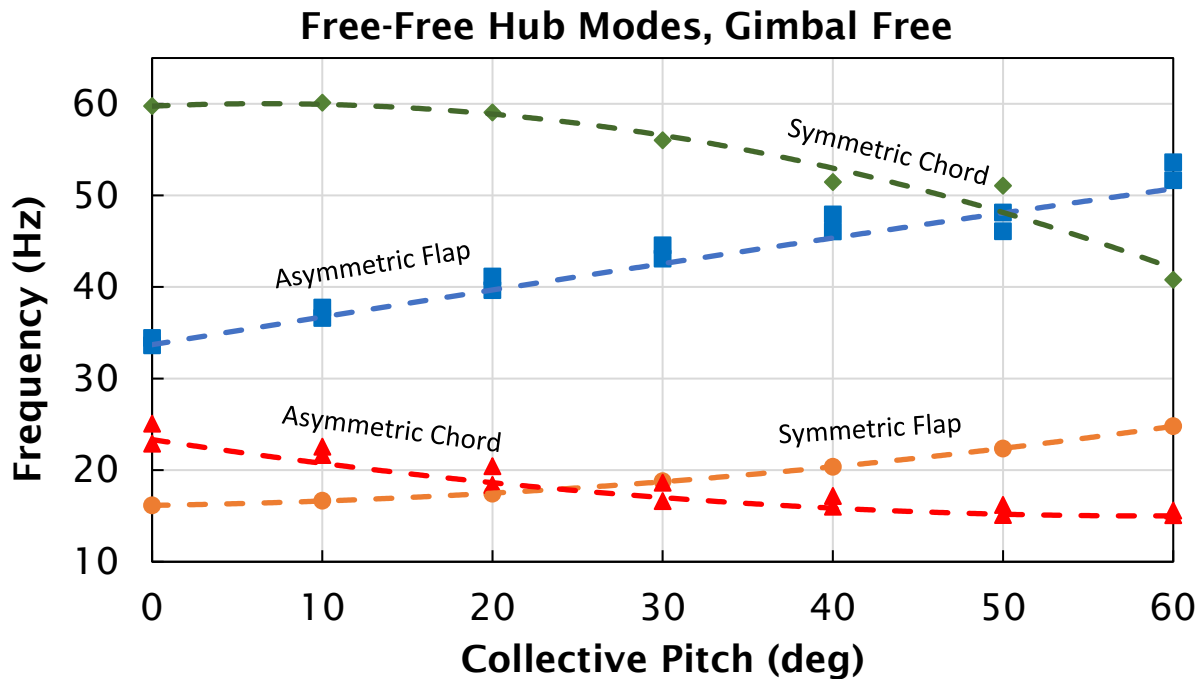


Figure 14. Free-free rotor frequencies versus collective.

4. Wing

The structural response of the wing was verified through testing of individual components as well as larger assemblies. The initial testing involved the bare wing spar before the aluminum flanges were bonded to the leading and trailing edges. However, due to constraints in the manufacturing process, it was not possible to conduct comparable testing of the spar with flanges prior to installation of the end rib assembly, drive shaft support ribs, and other features. An important note on the nomenclature used: ‘bare wing spar’ refers to the spar without the flanges, while ‘wing spar assembly’ includes bonded flanges, the end rib assembly, support ribs, and the spar support flange.

4.1 Bare Wing Spar

Prior to the bonding of the aluminum stiffeners to the leading and trailing edges, the bare wing spar underwent testing. The length of the spar was measured as 44.469 inches, with a height and width of 1.995 and 3.290 inches, respectively. The corresponding design dimensions were 2.00 and 3.30 inches. The design thickness for the leading and trailing-edge walls was 0.29 inches, but measurements showed the actual thickness to be in the range of 0.301-0.303 inches for the leading edge and 0.320-0.323 inches for the trailing edge. The spar cross-section was also observed to be thicker than designed and unsymmetric, as excess composite fabric was used during manufacturing and slight folding was observed in the laminate in the thicker section.

The bare spar was weighed and found to have a mass of 7.85 lbm. Eight holes for the wing root flange were already drilled, with a diameter of 0.37 inches and an average thickness of 0.39 inches. Assuming a composite density of 0.056 lbm/in³, the holes represented a loss of approximately 0.019 lbm. Thus, the running mass for the spar can be computed as $7.87 / 44.469 = 0.1770$ lbm/in.

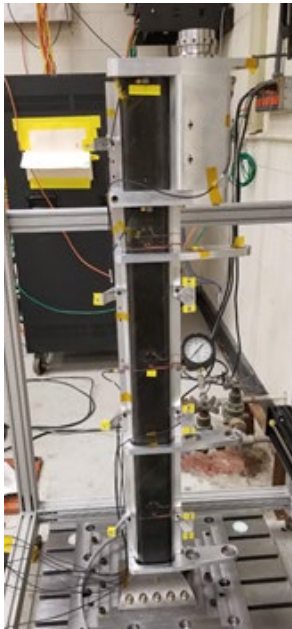
Natural frequencies were measured for the bare spar hung from a flexible line to simulate a free-free boundary condition. The fundamental modes are presented in Table 7. Additionally, the axial frequency of the spar beam was measured to be 2628 Hz.

Table 7. Free-free bare wing modes.

Out-of-Plane Modes		In-Plane Modes		Torsion Mode	
Frequency (Hz)	Damping (%)	Frequency (Hz)	Damping (%)	Frequency (Hz)	Damping (%)
300.1	0.26	389.9	0.28	1534	0.22
800.5	0.30	1039	0.38	3320	0.56
1489	0.33	1947	0.38		
2299	0.33	3058	0.38		

4.2 Wing Assembly Vibration Testing

Vibration testing was performed on the wing for various states of assembly, from the wing spar assembly to the full wing assembly (Figure 15). The vibration testing employed an instrumented hammer to excite the system and accelerometers to measure the wing response. Frequency and damping characteristics were extracted from FRFs using the Rational Fraction Polynomial-Z modal parameter estimation algorithm. The wing was secured to a $\frac{3}{4}$ -inch steel plate, which was mounted to an anchored T-slot floor mount. Tri-axial and single-axis accelerometers were strategically positioned along the spar and on other components to identify any modes where the shape might not be adequately described by beam-like spar motion.



a. Wing Spar Assembly



b. Full Wing Assembly

Figure 15. Configuration of the wing assembly during vibration testing.

The fundamental wing frequencies and modal damping values for the various wing configurations can be found in Table 8. In addition to the bare wing spar assembly, the assembly was tested with the drive shaft install and with both the drive and aileron shafts installed. The full wing assembly consists of the wing spar assembly, both drive and aileron shafts, the aileron, and the wing panels. Note that this full wing assembly excludes instrumentation cables and hydraulic lines, which are installed along the leading edge in the fully assembled TRAST. The distinct modes associated with each of the support ribs are documented in Table 9 for the bare wing spar assembly. Comparable modes were also present when the drive shaft and aileron were integrated, but these were challenging to discern from the measured FRFs.

One concern for the TRAST design was whether the drive shaft and aileron shaft would augment the wing stiffness and potentially increase the modal frequencies. The vibration testing revealed that these shafts did not increase the primary wing frequencies; but rather the frequencies decreased due to the added mass and inertia of the shafts (Table 8). This observation aligns with findings from previous testing involving roller bearings, where a single roller bearing will not constrain the rotational degrees of freedom, and thereby prevent direct transfer of wing bending into driveshaft bending.

Table 8. Measured modal properties of the wing for various states of assembly.

	Wing Spar Assembly (Figure 15a)		Drive Shaft Installed	Drive and Aileron Shafts	Full Wing Assembly (Figure 15b)	
Weight (lbm)	29.4				38.9	
	Freq (Hz)	Damp (%)	Freq (Hz)	Freq (Hz)	Freq (Hz)	Damp (%)
1 st Out of Plane	25.1	0.243	25.1	24.8	22.3	0.456
1 st In-Plane	42.2	0.311	41.8	41.3	37.6	0.536
2 nd Out of Plane	189.7	0.559	178.8	156.56	151.8	2.944
1 st Torsion	253.0	0.761	239.0	188.23	162.0	
2 nd In-Plane	295.8		295.6			
3 rd Out of Plane	471.6					
2 nd Torsion	537.5					

Table 9. Support rib modes.

Mode, Hz	Frequency (Hz)
Support Rib 1 Y	311.1
Support Rib 2 Y	324.4
Support Rib 2&3 Y	338.3
Support Rib 1 Z	494
Flap/2 nd Torsion +	537.5 (same as
Support Rib 2&3 Z	Table 8)
Support Rib 3 Z	592.3

4.3 Wing Assembly Static Displacement Testing

During the wing strain gage calibration process, additional laser displacement measurements were taken to quantify wing deflection under the static calibration loads. These measurements were utilized to validate the compliance of the wing's finite-element model. Although this single measurement does not yield specific beam stiffness properties, it offers an assessment of the structure's overall stiffness. The application of loads to the wing was facilitated by a wing calibration fixture that was affixed to the diaphragm attachment mount. The fixture enabled the application of vertical and chordwise loads to be applied that were aligned with the wing spar's centerline, and a torsion load through points applying a 15-in force couple about the conversion axis. Lasers were placed to measure the principal deflection for each loading condition. Three distinct load cases were evaluated and shown in Tables 10 thru 12:

Case 1: A force in the wing normal (Z^w) direction;

Case 2: A force in the wing in-plane direction (X^w);

Case 3: A wing torsion moment (MY^w).

For Case 1 and Case 2, the wing was horizontally mounted to the side of a strong back (not the floor fixture used for dynamic testing) and static weights were incrementally hung from the load fixture. The orientation of the wing determined whether the applied load was aligned to the wing normal or in-plane direction and whether it was positive or negative. In both cases, a total weight of 300 lbs was applied to the wing in 50 lb increments. The wing was then

flipped over, and measurements were repeated with the load in the opposite direction. This approach was deemed sufficient, as the load-deflection slopes for opposing loadings matched closely with each other. The load-deflection slopes, representing displacement per unit force, are detailed in Table 13.

For Case 3, wing torsion, the wing was horizontally mounted to the strong back once again. However, pulleys were employed to create a force couple using the static weights and the direction of the force couple was alternated instead of flipping the wing. A maximum moment of 1500 in-lb was applied in both directions. The moment deflection slope, detailed in Table 13 as displacement per unit moment, had the largest error of the curve-fit measurements. This is attributed to the torsion loading producing the smallest measured deflections and uncertainty in the load due to friction in the pulleys.

Table 10. Locations of loads and displacement measurements for Case 1 (normal loading).

Description	X ^w	Y ^w	Z ^w	Orientation
Loading	0.0 in.	45.73 in.	0.0in.	Z
Laser 1	-2.16 in. top 1.41 in. bot	44.3 in.	1.88 in. top -1.38 in. bot	Z
Laser 2	0.60 in. top -1.35 in. bot	44.3 in.	1.88 in. top -1.38 in. bot	Z

Table 11. Locations of loads and displacement measurements for Case 2 (in-plane loading).

Description	X ^w	Y ^w	Z ^w	Orientation
Loading	0.0 in.	45.73 in.	0.25 in.	X
Laser 1	-2.35 in. fore 5.7 in. aft	44.0 in.	-0.1 in.	X

Table 12. Locations of loads and displacement measurements for Case 3 (torsion force couple).

Description	X ^w	Y ^w	Z ^w	Orientation
Loading	-4.0 in. and 11.0 in.	45.73 in.	0.25 in.	MY
Laser 1	0.0 in.	39.2 in.	1.0 in.	Z
Laser 2	2.8 in.	39.2 in.	1.9 in.	Z

Table 13. Measured displacements for static wing loading.

Description	Vertical Load Displacement/F _z	Horizontal Load Displacement/F _x	Torsion (Laser2-Laser1)/M _y
Laser 1 (in/lb)	1.966e-3	7.29e-4	
Laser 2 (in/lb)	1.940e-3		
Difference (in/in-lb)			1.05e-5

5. Component Testing – Pitch Spring and Diaphragm Spring

The design of TRAST incorporates the ability to modify the dynamic pitch response of the pylon through the use of various pitch springs. These pitch springs influence the stiffness of the pylon about the conversion axis and accurate determination of this stiffness is critical to determine the flutter speed of the model. A set of six pitch springs has been manufactured with stiffness values ranging from 3,000-8,000 lb/in. To determine their effective spring stiffness, a bench-mounted rig utilizing a load cell and displacement lasers was used to measure the stiffness. The results of these measurements are detailed in Table 14. Notably, there was some observed variation between compression and tension measurements, this discrepancy is believed to be originating in the rod ends.

Table 14. Measured pitch spring stiffness.

Pitch Spring	Stiffness (lb/in), Tension	Stiffness (lb/in), Compression	Stiffness (lb/in), Average	Mass (lbm)
3k	2926.14	2797.10	2861.6	0.702
4k	4316.59	4267.68	4292.1	
5k	4709.72	4991.18	4850.5	
6k	5697.80	5889.26	5793.5	
7k	6673.02	6826.66	6749.4	0.778
8k	8797.17	9445.58	9121.2	

Another essential tuning spring utilized in the model is the diaphragm spring, highlighted in red within Figure 16. This particular component is unique to the wind tunnel model and serves to fine-tune the pylon dynamic response in yaw. The diaphragm spring is composed of six segments that collectively form a circular disk connecting the pylon to the wing. As a result, adjustments to both the pylon-yaw motion and roll motion can be achieved. For the baseline spring with a nominal stiffness of 20,000 ft-lb/rad in both directions, all six segments are identical. To validate the analytical modeling of the individual diaphragm segments, the frequencies were measured using an instrumented hammer and a single axis accelerometer on a single suspended segment (free-free boundary condition). The measured frequencies of this modal test for each diaphragm segment are presented in Table 15.

Table 15. Free-free natural frequencies for a single yaw spring segment in Hz.

Mode (Hz)	Mean Measured	Segment 1	Segment 2	Segment 3	Segment 4	Segment 5	Segment 6
1	1041	1037	1038	1041	1044	1044	1037
2	1216	1216	1216	1217	1219	1215	1214
3	2478	2472	2482	2475	2478	2486	2475
4	3430	3430	3434	3427	3432	3426	3431
5	4355	4349	4364	4356	4350	4360	4349

An inherent challenge associated with this particular spring lies in its high combined stiffness, which poses difficulties when conducting static testing. To address this, a dedicated loading fixture was fabricated for applying bending moments (pylon yaw and roll) and axial force (wing spanwise) to the full set of six segments. Laser displacement sensors are used to measure the corresponding rotations and axial displacement across the diaphragm spring. Using this method, the yaw stiffness (KM_z), roll stiffness (KM_x) and spanwise stiffness (KF_y) were measured for both the baseline spring and a spring with a target stiffness of only 10,000 ft-lb/rad in the yaw direction. The diaphragm spring is assumed to be effectively rigid in the other three directions. The stiffness of the baseline spring was within 10% of the target stiffness from the detail design phase, with the measured stiffness being 22,000 ft-lb/rad for roll and 21,900 ft-lb/rad for yaw. The 10k/20k spring was not as close to the nominal design target, with the measured stiffness being

around 14,800 ft-lb/rad in the yaw direction and 23,100 ft-lb/rad in the roll direction. Both springs had a measured spanwise stiffness of 7.3E5 lb/ft. The results of the stiffness test are presented in Table 16.

Table 16. Diaphragm spring stiffness.

	Baseline 20k Spring	10k/20k Spring
KMx (ft-lb/rad)	22,000	23,100
KMz (ft-lb/rad)	21,900	14,800
KFy (lb/ft)	7.35E+05	7.31E+05

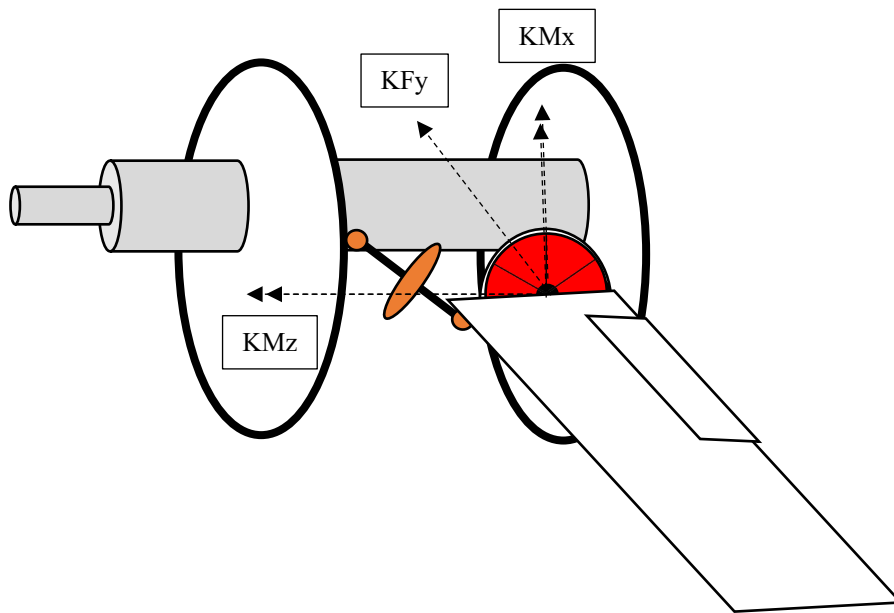


Figure 16. TRAST pitch spring and diaphragm spring arrangement.

6. Full TRAST GVT Configuration and Accelerometer Layout

In addition to the component testing discussed in previous sections, comprehensive vibration testing was undertaken on the complete TRAST model. Broadly, the system configuration was identical to the Test 662 configuration; however, several distinctions warrant mention. Throughout the tests outlined in this section, the rotor system was configured with a δ_3 of -15° . The gimbal was free to rotate, and no dummy blade mass was used. The 8k-pitch-spring configuration was the only one tested in the hover cell, though select results from the wind-tunnel vibration testing of other configurations will be presented. When referencing the full system vibration test, the presence or absence of rotor blades will be explicitly indicated. Another detail is that the slipring underwent substitution from the the Moog AC6355 (0.574 lbm) employed in the 8k pitch spring T662 and GVT configuration with to an Orbex 100247 (0.644 lbm) during T662 during the 4k pitch spring phase of testing.

The TRAST model was installed on the hover stand in airplane mode at zero angle of attack. The pylon's alignment was refined by adjusting the length of the pitch spring rod ends, ensuring the mast tower was horizontal and the bulkheads were vertical. All wing and pylon skins were installed, including the spinner. The aileron was installed and fixed at 0° . No tuning masses were incorporated. The shaft was connected to the motor but not locked, with hub rotation set to have blade 1 oriented down (approximately 180° azimuth). Testing was conducted both with and without blades, and with the gimbal free to rotate.

An assortment of accelerometers was employed to measure the motion of the model accurately. TRAST incorporates eight single-axis DC accelerometers permanently affixed to the pylon bulkheads, locations illustrated in Figure 17, and three more placed on the wing tip. This arrangement enables the comprehensive quantification of the pylon's six-degree-of-freedom rigid-body motion. Furthermore, four triaxial accelerometers were mounted on the hub for the GVT, positioned as depicted in Figure 18. The coordinate system definition and specific locations for all the accelerometers are tabulated in Appendix B.

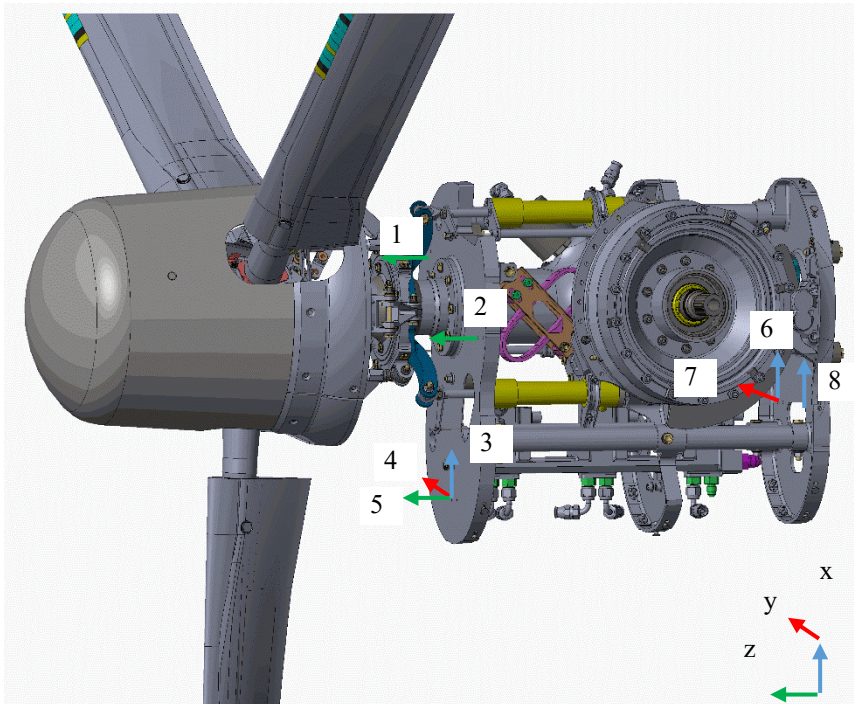


Figure 17. Accelerometers mounted on the pylon.

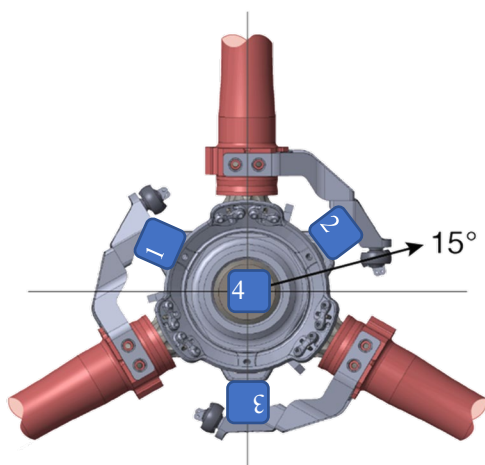


Figure 18. Hub accelerometer layout.

The TRAST airplane configurations were the primary focus for vibration testing and helicopter mode was only tested in the hover cell. Testing in the hover cell utilized both free-decay measurements and shaker excitation, but in the wind tunnel only free decay was used to quantify frequencies and damping. As discussed above, for the majority of testing the 8k pitch spring was installed, and measurements with the 4k pitch spring were only conducted in the wind tunnel. The helicopter mode testing proved particularly effective in identifying the blade frequencies when the swashplate was excited by the hydraulic control system. The breakdown of the test matrix outlined in this section are shown in Table 17.

Table 17. Vibration test matrix completed and documented section.

Test Matrix		Tunnel Free Decay	Hover Cell Free Decay	Hover Cell Shake Test	Swashplate Excitation
Airplane Mode	4k Spring, No blades	Section 6.1			
	4k Spring, With blades	Section 6.2			
	8k Spring, No blades	Section 6.1	Section 6.1	Section 6.1	
	8k Spring, With blades	Section 6.2	Section 6.2	Section 6.2	
Helicopter Mode	8k Spring, No blades		Section 6.3		Section 6.4
	8k Spring, With blades		Section 6.3		Section 6.4

6.1 Airplane Mode Modal Testing without Blades

The estimation of the TRAST model's vibration and modal characteristics was accomplished through three distinct methodologies: FRFs acquired via stepped-sine shaker excitation, FRFs obtained from instrumented-hammer impacts, and free-decay measurements. Each of these three methods offer unique benefits, and all three were employed at different stages of model preparation and wind-tunnel testing. FRFs generated using either a shaker or instrumented hammer produce accurate frequency, damping and mode-shape information when processed with a curve-fit estimator. The Rational Fraction Polynomial-Z method, detailed in Ref. 6, was adopted for the analysis of the test data.

Hammer excitation, while requiring minimal setup, does not always transmit an optimal amount of low-frequency energy to the model. As a result, it is most suitable for investigating blade-hub modes rather than low-frequency, high modal mass, wing-pylon modes.

Shaker excitation is highly effective for controlling excitation frequencies and amplitude, particularly when employing stepped-sine excitation. The drawbacks are that the setup is challenging with the height of the model above the floor, and the potential for inducing undesirable changes to the dynamic response of the system when attaching the shaker due to the absence of built-in connection points. In the testing outlined in this document, a crane was utilized to suspend the shaker and reaction mass from above the model. Since there are no mounting points on the spinner for the shaker to attach to, the weight of the suspended system was used to hold the end of the sting in contact with the spinner at the desired location. This is accomplished directly from the weight of the shaker system for vertical excitation or by a pendulum restoring force during horizontal excitation. This technique limits the magnitude of the shaker force that can be applied, but it was determined to be sufficient for the TRAST model (refer to Figure 19 for shaker setup).

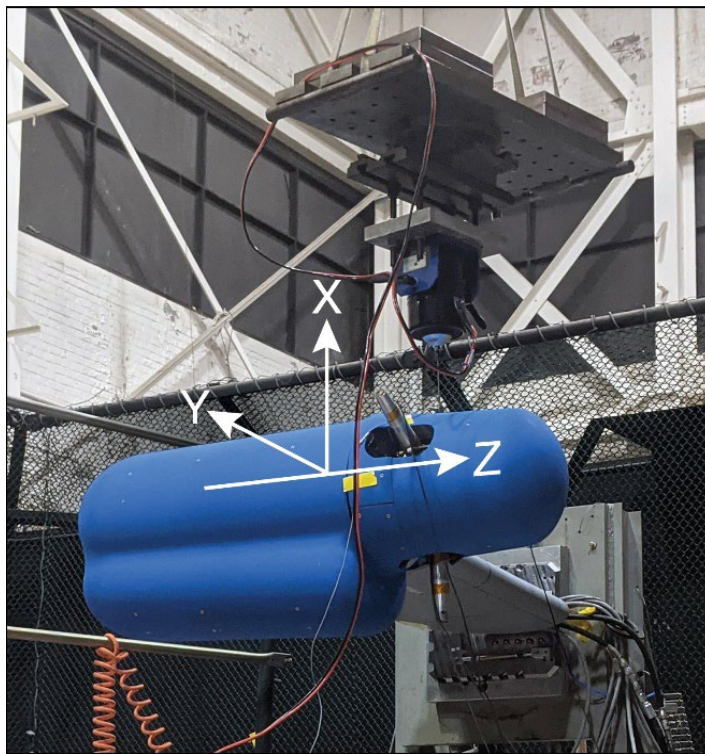


Figure 19. TRAST shaker GVT setup without blades.

The free decay method is the most effective approach for identifying the system damping and offers the advantage of not requiring additional instrumentation or hardware setup. Typically, the response measured by the wing-bending strain gages and the pitch-spring tension gage are used, but the response of the pylon accelerometers has been shown to work as well. The measured time history is analyzed using either a moving-block or Prony analysis to extract frequency and damping information. This method is employed after any configuration change during wind-tunnel testing to promptly assess the frequency and damping of the primary system modes. It is important to note that due to the absence of a measured forcing function, extraction of the complete modal description of the model is not achievable with this method.

The four primary modes of the wing-pylon system are most effectively visualized by examining the modes extracted using the finite-element model in airplane mode, as presented in Figure 20. The first mode is the wing vertical bending mode, akin to the first natural mode in a cantilevered beam. However, the presence of the pitch spring and the offset of the pylon center of gravity introduce a substantial rotational component to this mode. The second mode corresponds

to wing in-plane bending, primarily driven by the chordwise motion of the wing spar with a minor contribution from the diaphragm spring. The third mode is wing torsion, sometimes referred to as a pylon pitch mode due to the significant rotational contribution of the pitch spring to the mode shape. The final wing mode of interest for flutter is the pylon yaw, which bears resemblance to the second chordwise bending mode but with the wing rotating forward and the diaphragm spring allowing the pylon to rotate outboard.

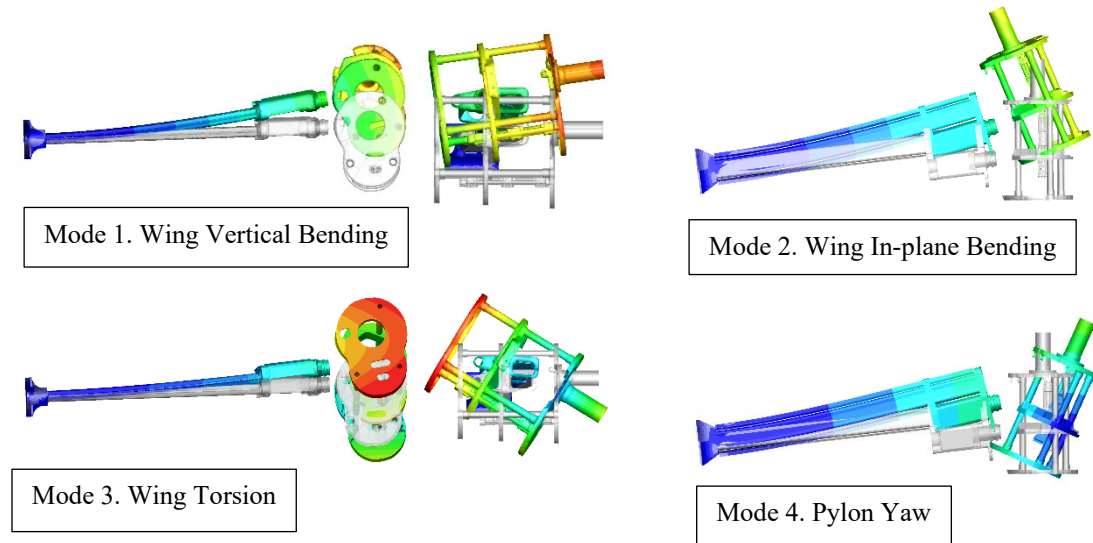


Figure 20. Mode shapes from finite-element model.

To excite these four modes, a variety of excitation locations and orientations were explored. Regardless of the excitation method used, the optimal results were achieved by applying loads to the spinner that were aligned with the center of the hub. Specifically, wing-vertical and wing-torsion modes were excited by a vertical force on the side of the spinner from above, the wing-in-plane mode was excited by a force at the front of the spinner aligned with the shaft axis, and the pylon yaw mode was excited by a horizontal force on the outboard side of the spinner.

A summary of the frequencies and damping values of the baseline model in the hover cell measured using the three excitation methods is presented in Table 18. Generally, the three methods yielded comparable results. The most significant disparity was observed for the torsion mode when excited with the instrumented hammer. The hammer impact at the hub simultaneously excites the wing vertical and torsion modes. Due to the sensitivity of the wing vertical mode and the highly damped torsion mode, the hammer impact did not induce sufficient wing torsion response, resulting in relatively inaccurate modal estimates. Another noticeable observation is that the wing vertical bending damping values differ among the three excitation methods. This discrepancy is attributed to system nonlinearity that results in an amplitude dependency for the damping and frequency of certain modes. Since there are large differences in the peak loads induced by the three excitation methods, this results in different modal estimates being computed from the three measurements. This amplitude dependency is further demonstrated in Table 19, which displays the frequency and damping values for the wing-vertical-bending mode under three different shaker excitation amplitudes. A minor increase in frequency is observed as the shaker excitation amplitude decreases, while a more significant reduction occurs in the damping. In fact, the 0.16% reduction in damping closely resembles the difference in damping observed with hammer excitation.

Table 18. Measured vibration response of the primary modes without blades in the hover cell.

Mode	Shaker		Hammer		Free Decay	
	Frequency (Hz)	Damping (%)	Frequency (Hz)	Damping (%)	Frequency (Hz)	Damping (%)
Wing Vertical Bending	5.383	0.72	5.387	0.57	5.365	0.67
Wing In-Plane Bending	8.310	0.81	8.336	0.93	8.316	0.58
Wing Torsion/Pylon Pitch	11.80	5.4	12.5	5.9	11.71	5.8
Wing 2 nd In-Plane/Pylon Yaw	20.68	0.78	20.57	1.01	20.25	1.35

Table 19. Measured vibration response with shaker amplitude variation without blades in the hover cell.

Mode	Shaker High		Shaker Medium		Shaker Low	
	Frequency (Hz)	Damping (%)	Frequency (Hz)	Damping (%)	Frequency (Hz)	Damping (%)
Wing Vertical Bending	5.383	0.72	5.403	0.68	5.405	0.56

In whirl flutter analysis, one of the crucial factors involves having not only accurate frequency and damping values of the wing vertical mode, but also the specific mode shape at the hub, particularly the rotations in pitch and yaw. This motion can be computed using the accelerometers on the pylon to estimate the rigid body motion of the system. The mode shapes are then extracted from the FRF using the Rational Fraction Polynomial-Z method, which incorporates the resolved motion of the accelerometers and normalizes the shape to unit modal mass (slinch units). For the mode shapes listed in Table 20 the rotation components are calculated from the permanently mounted pylon accelerometers and the hub accelerometers for the displacement, refer to Appendix B for coordinate system definition.

Table 20. Resolved mode shape at the hub node for the 8k pitch spring.

Mode Shapes	Freq (Hz)	X _p (in/in)	Y _p (in/in)	Z _p (in/in)	rX _p (rad/in)	rY _p (rad/in)	rZ _p (rad/in)
Wing Vertical Bending	5.396	2.156	0.2723	0.1515	0.00257	0.01823	-0.0199
Wing In-plane Bending	8.305	0.2511	-1.564	2.000	-0.08857	-0.00205	-0.0067
Wing Torsion	11.58	3.216	-0.277	-0.548	-0.0121	0.2374	-0.0367
Pylon Yaw	20.7	0.2346	3.470	-0.1938	0.1646	0.0165	0.0027

The TRAST model underwent shake testing with both 4k and 8k pitch springs while installed in the wind tunnel. The frequencies and damping for the first four wing modes were measured and are recorded in Table 21 for the corresponding pitch spring configurations. The frequencies observed were found to be consistent with those in the hover cell.

Table 21. 4k-pitch-spring versus 8k-pitch-spring configuration without blades (wind tunnel).

Mode	4k Pitch Spring		8k Pitch Spring	
	Frequency (Hz)	Damping (%)	Frequency (Hz)	Damping (%)
Wing Vertical Bending	5.23	0.74	5.36	0.53
Wing In-plane Bending	8.20	0.87	8.35	0.67
Wing Torsion/Pylon Pitch	9.81	4.40	11.75	4.60
Wing 2 nd In-plane/Pylon Yaw	19.89	1.46	20.50	1.16

6.2 Airplane Mode Modal Testing with Blades

Additionally, some TRAST configurations underwent vibration testing with the blades installed, as shown in Figure 21. These measurements were conducted exclusively with the free decay method, because the blades prevented the shaker from being suspended in close proximity to the model for computing damping. The frequencies and damping for the four primary modes are detailed in Table 22 for the 8k-pitch-spring, airplane-mode configuration. A noticeable discrepancy is observed in the wing-vertical-bending mode, which exhibits significantly higher damping ratio compared to measurements without the blades. This difference is attributed to gimbal friction and is discussed in detail in a subsequent section.

Table 22. Full model free decay measurements with blades, 8k spring, coll = 31.66°.

Mode	Free Decay	
	Frequency (Hz)	Damping (%)
Wing Vertical Bending	5.052	1.81
Wing In-plane Bending	7.749	0.64
Wing Torsion/Pylon Pitch	11.55	5.50
Wing 2 nd In-plane/Pylon Yaw	19.50	1.70



Figure 21. TRAST hover cell GVT with blades installed.

Comparable free decay measurements were carried in the wind tunnel with the wind off, and the results are presented in Table 23. Wind-tunnel flutter measurements are not included in this document; these vibration measurements are presented for illustration of the effect of the difference in mounting stiffness between the wind tunnel and the hover cell.

Table 23. Wind tunnel versus hover cell with blades, 8k pitch spring.

Mode	Hover Cell (Coll = 31.66°)		Wind Tunnel (Coll = 0.1°)	
	Frequency (Hz)	Damping (%)	Frequency (Hz)	Damping (%)
Wing Vertical Bending	5.052	1.81	5.13	2.5
Wing In-plane Bending	7.749	0.64	7.71	0.82
Wing Torsion/Pylon Pitch	11.55	5.5	11.5	5+
Wing 2 nd In-plane/Pylon Yaw	19.5	1.7	18.5	3.3

In addition to employing the 8k pitch spring, the 4k pitch spring was installed and tested in the wind tunnel. The vibration measurements for this configuration are presented in Table 24. The most significant effect of changing to the 4k pitch spring is a reduction of the wing torsion frequency by almost 10%. Though not fully illustrated by these values, it also alters the mode shape of the wing vertical bending mode and results in a slight reduction in the frequency of that mode.

Table 24. 4k pitch spring versus 8k pitch spring with blades (wind tunnel).

Mode	4k Pitch Spring		8k Pitch Spring	
	Frequency (Hz)	Damping (%)	Frequency (Hz)	Damping (%)
Wing Vertical Bending	5.03	2.7	5.13	2.5
Wing In-plane Bending	7.74	1.05	7.71	0.82
Wing Torsion/Pylon Pitch	10.6	5+	11.5	5+
Wing 2 nd In-plane/Pylon Yaw	18.5	3.5	18.5	3.3

6.3 Full TRAST System – Helicopter Mode

For helicopter mode testing, the pylon's orientation was transformed from a horizontal configuration to an approximately vertical position through a 90° rotation about the conversion axis. This axis runs parallel to the swept and dihedral wing axis. Subsequently, the frequency and damping were measured using the same free decay method as previously outlined for airplane mode. One key distinction between helicopter mode and airplane mode is that pylon pitch motion couples with wing in-plane motion in helicopter mode rather than with wing vertical motion. Consequently, wing vertical motion no longer induces significant gimbal motion with the blades installed. Thus, the large increase in damping with blades installed does not occur for the wing vertical mode, but instead appears in the in-plane mode. The frequencies and damping characteristics are detailed in Table 25 for three scenarios: without blades, with blades, and with a different set of gimbal springs, discussed in more detail in Section 6.5. The new springs were an attempt to reduce damping in airplane mode. In helicopter mode, the damping increases with the softer springs, likely due to the softer springs allowing more gimbal motion and therefore allowing friction to remove more energy from the system. It is important to note that this trend only applied to the helicopter mode GVT. The introduction of blades introduces added complexity in determining the yaw frequency due to blade dynamics, which will be elaborated upon in section 6.4.

Table 25. Helicopter mode frequency and damping (8k spring, coll = 32.4°).

Mode	No Blades		With Blades		New Springs (with blades)	
	Frequency (Hz)	Damping (%)	Frequency (Hz)	Damping (%)	Frequency (Hz)	Damping (%)
Wing Vertical Bending	5.2	0.25	4.95	0.25	5.03	0.5
Wing In-plane Bending	7.75	1.05	6.95	2.5	6.95	3.0
Wing Torsion/ Pylon Pitch	11.6	0.6	10.65	1.5	11.3	2.7
Wing 2nd In-plane/ Pylon Yaw	18.78	0.8	See Figure 22		See Figure 22	

6.4 Model testing with Blades – Blade Modes (Helicopter Mode, swashplate excitation)

Beyond the wing-pylon modes, there are rotor frequencies that interact with the wing and pylon system and influence the stability. One mode of particular concern is the regressive lag mode, characterized by this asymmetric mode with predominant motion in the blade's chordwise direction. This mode occurs at a frequency that is the blade lag frequency minus the rotor speed. Further complicating the whirl-flutter predictions, the lag frequency changes as the rotor blades pitch due to the coupling between the hub and blade structural dynamics.

As the rotor-blade modes interact with the wing-pylon modes along with the pitch angle, an attempt was made to capture these modes on the complete TRAST system with the blades in place. Initially, an instrumented hammer was utilized to excite the rotor-blade modes, but that method proved insufficient for adequately exciting all of the blade modes. Instead, low-frequency random noise was applied to the swashplate actuators, and the output from the blade and spindle bending strain gages was recorded as the frequency response. This test was conducted with the TRAST in both airplane and helicopter configurations, although the frequencies in airplane-mode were not as distinctly discernible in the response data. The response was measured in 5° collective increments for 60 seconds and a moving average was applied to calculate the FFTs and cross-spectra of the responses. For each collective angle, the blade frequencies were determined from the peaks observed in the cross-spectra between the blade and spindle bending gages against the pitch link gages.

Figure 22 displays four estimated hub-blade frequencies that are particularly relevant for TRAST in hover mode. The first frequency represents the blade symmetric flap mode (black), where all blades exhibit an in-phase out-of-plane motion. This frequency increases with collective pitch, and as the pitch increases this mode includes a larger component of chordwise blade bending. Similarly, there is a pair of asymmetric lag frequencies (blue and green) that decrease with increasing collective pitch. The fourth frequency is the pylon yaw mode (red), and as discussed previously, changes with collective. As collective increases the two asymmetric lag modes become strongly coupled with the pylon yaw mode. Above 25° collective the coupling is substantial to the point that distinguishing between them in the response measurements become challenging, leading to these three modes being better described as a set of three pylon-yaw/blade-lag modes.

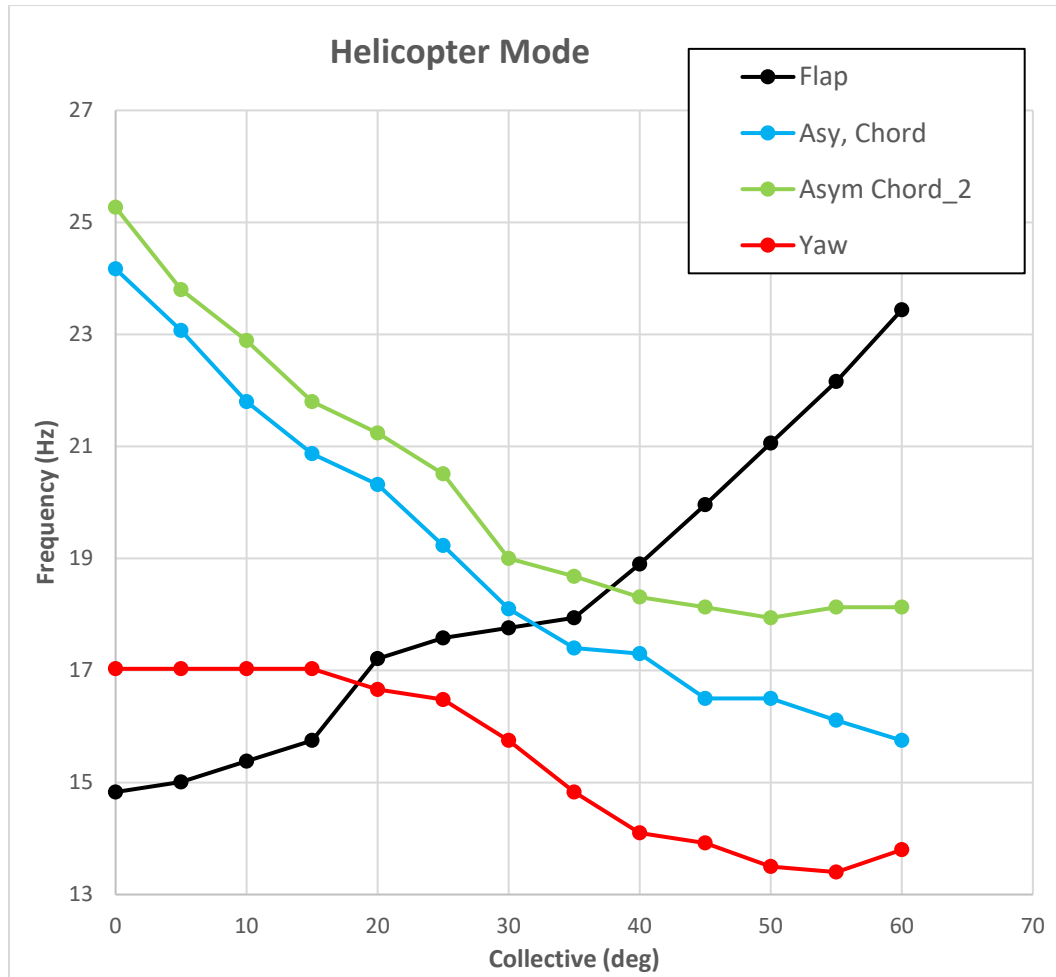


Figure 22. TRAST System Frequencies from 13-27 Hz, in helicopter mode.

6.5 Damping

As previously discussed, it was observed that the measured system damping increased with higher excitation amplitudes, as shown in Table 19. This phenomenon was attributed to friction within the model, particularly in the hub gimbal and the conversion bearing. This hypothesis was substantiated when comparing the measured wing bending to the gimbal motion during free-decay measurements of the TRAST modes. Figure 23 presents the output of one of the vertical wing-bending gages (displayed in black) and the measured pitchwise gimbal motion (displayed in blue) during a free-response measurement of the TRAST with blades installed. In this specific scenario the TRAST was in airplane mode within the TDT, installed with the 4k pitch spring, and the entire system was at an ambient temperature of 70°F before tunnel operations commenced. The gimbal motion can be observed to decay very rapidly, and after 7-8 oscillations stiction occurs, causing the gimbal to reach zero discernable motion. During gimbal motion, the combination of gimbal friction and substantial rotor inertia effectively dissipates energy from the system, resulting in a relatively high damping ratio of the wing and pylon (approximately 2.5 percent) that is noticeable in the wing-bending response. Once gimbal motion ceases, the rate of energy dissipation from the system diminishes, leading to significantly lower wing damping (approximately 0.65 percent), which matches the damping measurement without blades, seen in Table 18. This behavior is indicative of friction rather than viscous damping, and presents a notable challenge when analytically modeling the TRAST in comprehensive analysis codes.

During wind-tunnel testing, a noteworthy observation was made regarding the TRAST's system damping. It was found that after long periods of operation when the model had reached a higher temperature, over 100°F, the system exhibited lower damping compared to the initial state before a test run. Additionally, at these elevated temperatures the gimbal motion took approximately twice as many cycles to decay, and the overall system response more closely resembles viscous damping as depicted in Figure 24.

To investigate this temperature-dependent behavior, a series of tests were conducted in the hover cell to quantify the relationship between temperature and system damping. The TRAST was operated in the hover cell in airplane mode at nominal rotor speed without blades installed. This gradually increased the gearbox temperature from ambient to approximately 130°F. At intervals of 10°F, the motor was stopped, blades were installed, and a set of free-decay measurements was performed. The computed damping values for the wing-bending-mode from this test are presented in Figure 25. Damping values calculated from the early portion of the decay response are distinguished from those computed during the later stages of the decay. It was observed that the damping decreases from an initial damping level of 2.8% to around 2.1-2.2% as the temperature reached 100°F and remains relatively constant above this temperature.

The damping observed from the gimbal during Test 662 was found to be less than ideal due to a significant non-linearity associated with it which ultimately is not ideal for a benchmark test. To address this issue and reduce the damping present in the gimbal, two key changes were implemented in the hub. Firstly, the original grease used to lubricate the constant-velocity gimbal joint was replaced with a lighter-weight grease. This new grease was selected for its lower viscosity at lower temperatures, aiming to improve the gimbal by reducing friction. Secondly, the gimbal springs were substituted with a set of more compliant springs. Since tension springs are employed in the gimbal configuration, they are installed in a manner that draws the hub and upper spherical bearing together, creating a substantial internal force on the joint. It's important to note that the level of gimbal friction is directly proportional to the magnitude of this internal reaction force. By replacing the six original 24.67-lb/in gimbal spring with springs possessing a stiffness of 1.80-lb/in significant reduction in friction losses within the hub were achieved.

The measured damping values following these modifications are presented in Figure 26. Although the temperature dependence of the system damping did not undergo significant changes, the system damping now converges to a value of approximately 1.3-percent damping above 100°F. Based on these results, the plan for future wind-tunnel testing involve briefly running the model to allow the system to warm up to temperatures exceeding 100°F before obtaining system-damping data.

The conversion bearing motion was observed to increase the system damping of certain modes, though this effect was observed to be smaller than that for the gimbal. Unlike the gimbal, this joint did not exhibit any temperature dependency. However, it was found that the damping contribution is a function of the amplitude of the motion. Figure 27 illustrates the damping and frequency characteristics of the wing-vertical mode in relation to the amplitude of the wing-bending moment for the TRAST in airplane mode, where no blades are installed. This configuration minimizes the effect of gimbal motion on the damping. To derive these damping measurements, multiple free-decay measurements were analyzed in 1-second time segments spanning the length of the free-decay. The results clearly indicate an increase in system damping and a decrease in modal frequency as the oscillation amplitude increases. This behavior is indicative of losses stemming from friction, and given that it primarily affects the wing vertical bending and pylon-pitch modes indicates that source is the conversion bearing.

The damping versus temperature trend was observed in helicopter mode and is presented in Figure 28. In contrast to airplane mode with the original springs and blades, there was minimal temperature variation but a slight wing amplitude dependency. The new gimbal springs which were more flexible exhibiting a greater dependency on the amplitude. The hypothesis behind this behavior is that the rotor is no longer offset from the primary vertical motion. Consequently, the springs do not move significantly, resulting in very little damping (approximately 0.2-0.3%). When the new gimbal springs were installed, it took very little moment for the springs to allow gimbaling, which actually increased the damping of the wing-vertical-bending mode. However, this increase in damping was deemed acceptable in order to reduce damping in the airplane vertical-bending mode.

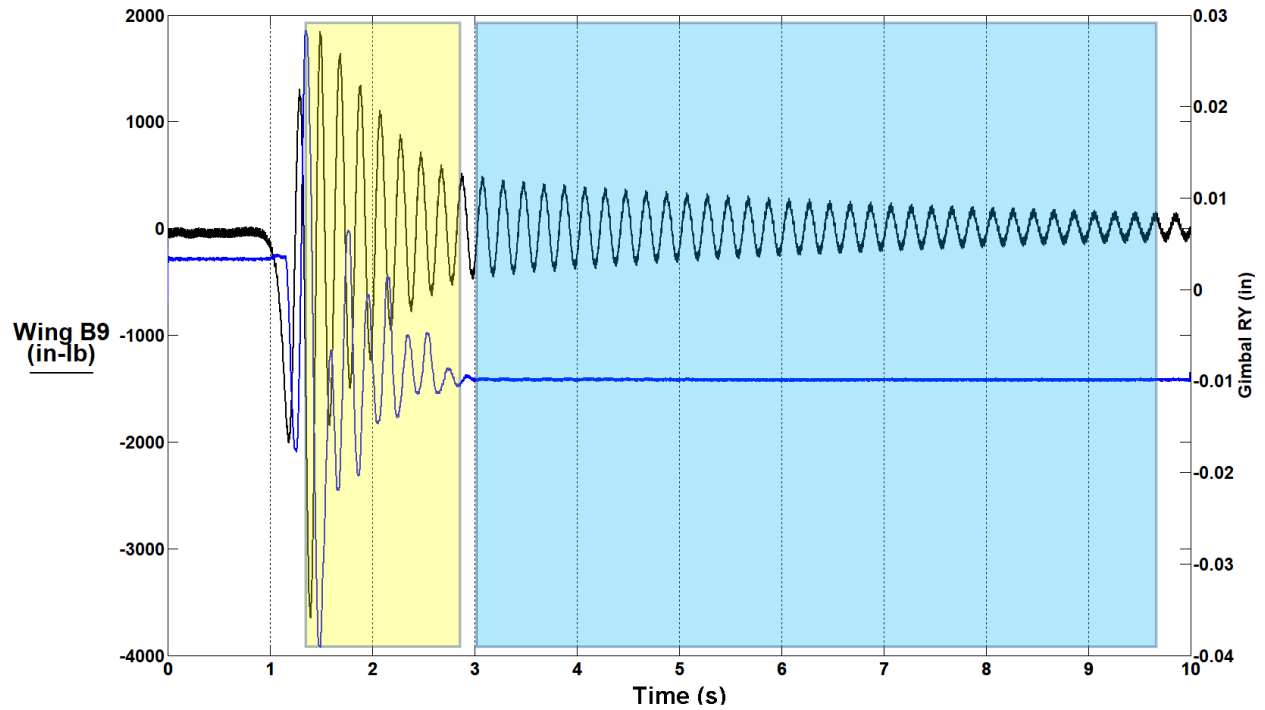


Figure 23. Damping trend of wing bending (black) and gimbal motion (blue) with 4k pitch spring during T662, +70 degF.

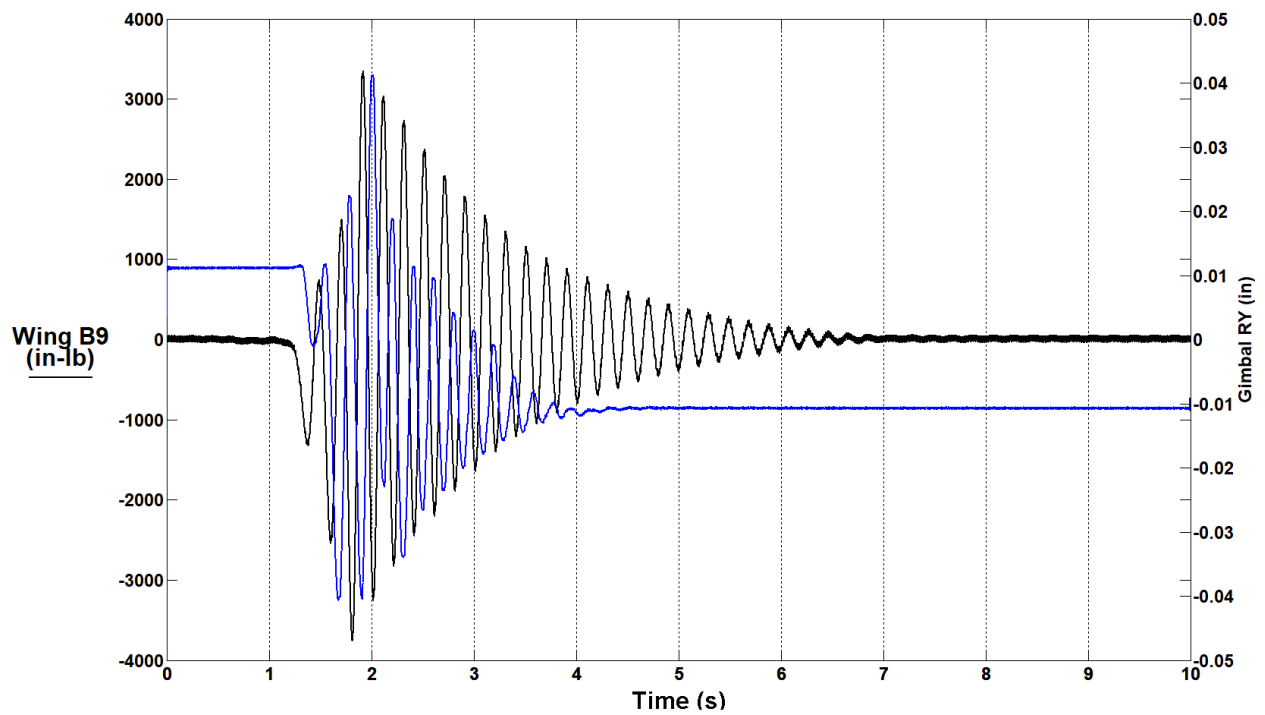


Figure 24. Damping trend of wing bending (black) and gimbal motion (blue) 4k pitch spring during T662, +100°F.

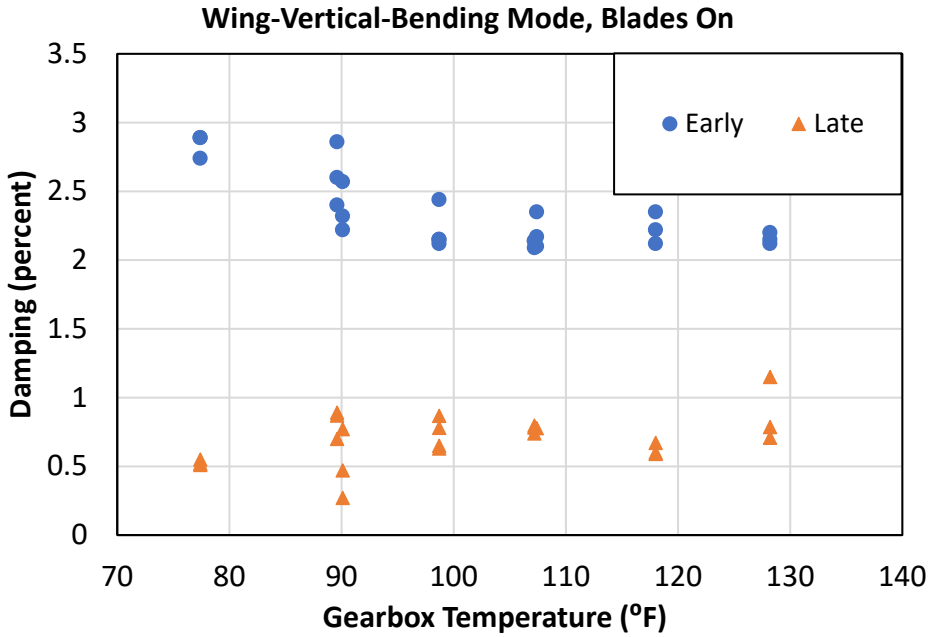


Figure 25. Post-test damping vs gearbox temperature, 8k pitch spring, T662 configuration, airplane mode.

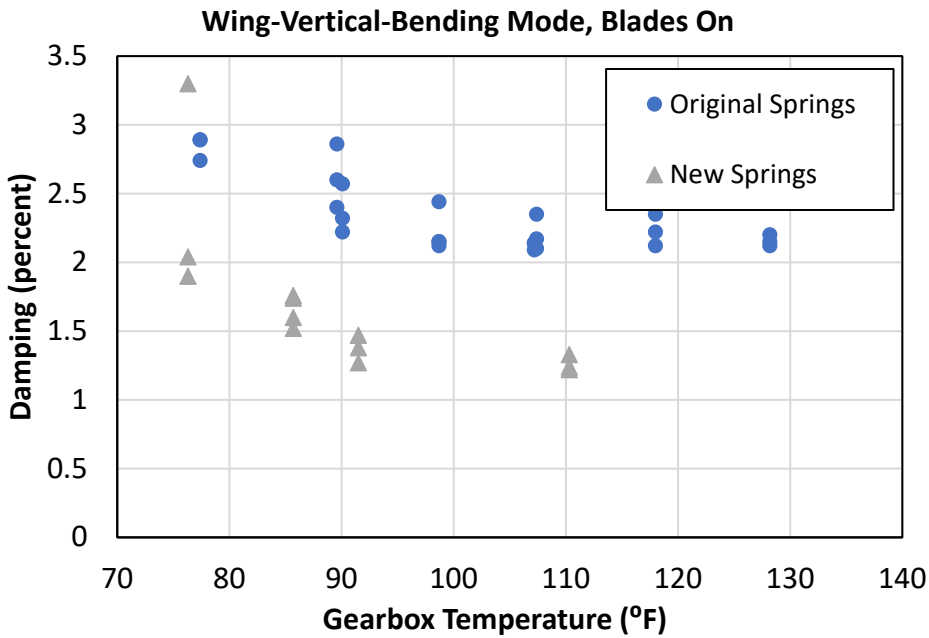


Figure 26. New gimbal springs damping versus gearbox temperature, airplane mode.

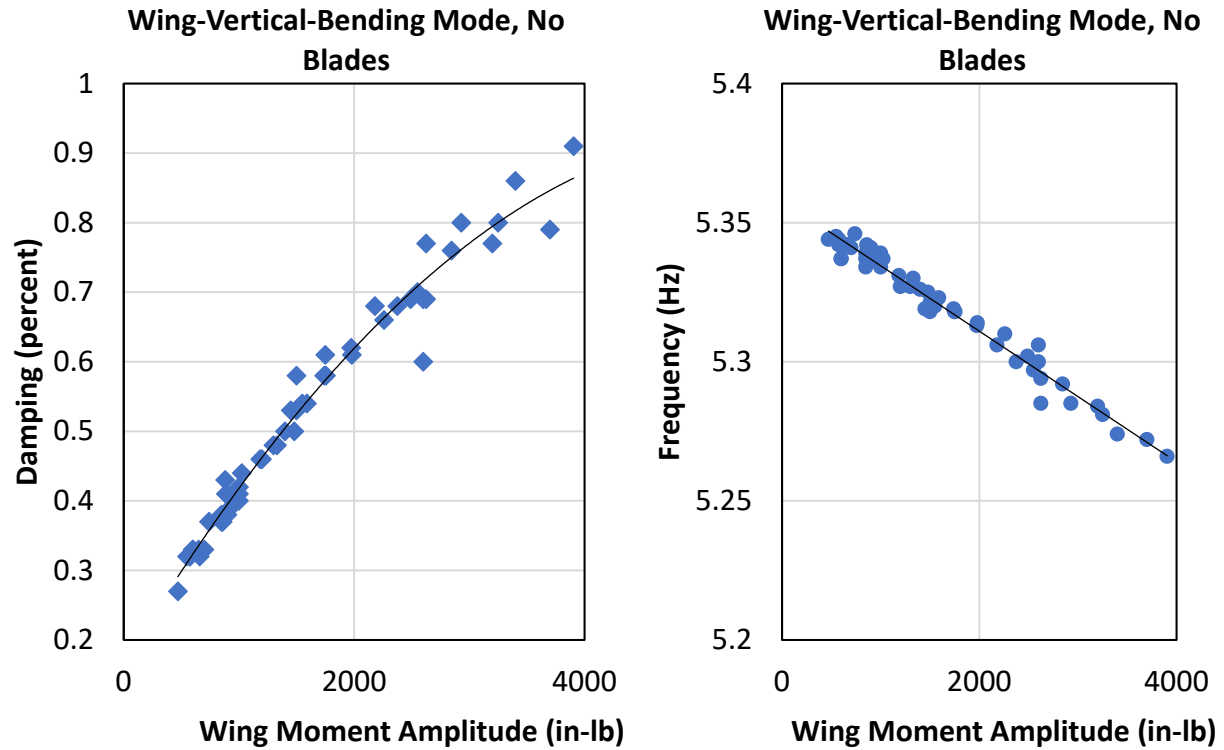


Figure 27. Wing-vertical-bending mode damping and frequency versus wing bending moment amplitude, airplane mode.

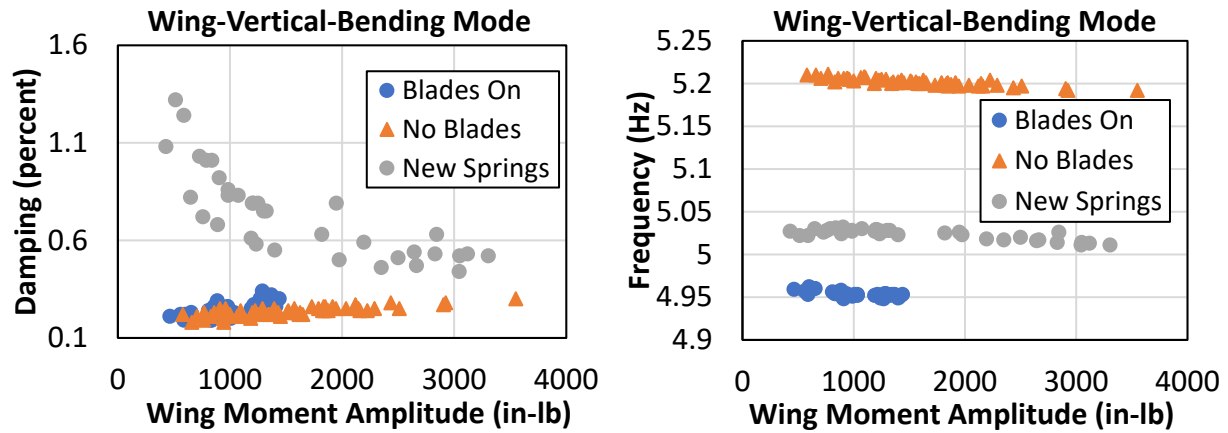


Figure 28. Helicopter mode damping for the original gimbal springs (blades on and no blades) and the new gimbal springs (blades on).

7. Concluding remarks

This document is intended to record the results of the ground vibration testing that has occurred on the TRAST model in relationship to Test 662 in the Transonic Dynamics Tunnel and covers the associated testing with the wing, blade, rotor, and other important structural dynamic parts. This document is intended to be used with the companion TM (Ref. 7) for description of best practices for analytically modeling TRAST as demonstrated in Ref. 8. This document also investigates some of the sources of damping that have occurred during testing with the ultimate intention of minimizing this for any follow-on wind tunnel testing. Several key findings have emerged from the extensive testing and analysis:

1. Pitch and Yaw Dynamics: The pitch and yaw stiffness was measured that control the two fundamental degrees of freedom important for whirl flutter. The pitch stiffness influences the mode shape of the wing-vertical-bending and torsion modes. Whereas the yaw stiffness influences the yaw mode. During the complete TRAST vibration measurement the yaw mode introduces coupling with rotor-blade modes. As the collective increases, the dynamics of the system were influenced by blade pitch, leading to the emergence of coupled pylon-yaw/blade-lag modes.
2. Mode Shape Estimation: Accurate estimation of mode shapes, especially rotations in pitch and yaw, play a crucial role in whirl flutter. These mode shapes were extracted from the measured responses using curve-fit estimators, providing valuable insights into the TRAST's behavior.
3. Blade Effects: The presence of rotor blades had a significant impact on system dynamics, with differences observed in mode frequencies and damping ratios. Blade-induced friction in the gimbal added complexity to the system's response.
4. Gimbal Effects: The TRAST model exhibited a complex relationship between system damping and temperature. The gimbal friction and gimbal spring flexibility had a notable impact on damping which was more temperature dependent than amplitude dependent.
5. Conversion Bearing Effects: The conversion bearing also contributed to damping variations, with an amplitude dependency which was independent of the gearbox temperature.
6. Future Testing Considerations: As a result of the observed temperature dependency, future wind-tunnel testing plans involve allowing the model to reach a stable temperature above 100°F before collecting system damping data. This approach aims to provide more consistent and reliable results.

In conclusion, the TRAST model serves as a valuable testbed for aeroelastic stability investigations, offering insights into the complex interactions between rotorcraft components. This technical memorandum underscores the importance of detailed analysis and testing to enhance our understanding of rotorcraft dynamics, providing essential information for the development of safer and more efficient tiltrotor aircraft.

8. References

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Appendix A. Measured Values During the Free-Free Rotor Testing

Table A.1 Measured frequencies using shaker excitation.

Measured Frequencies (Hz)	Shaker						
Pitch (deg)	0	10	20	30	40	50	60
Flap Sym	16.15	16.66	17.43	18.79	20.36	22.36	24.80
1st Asym Lag	22.89	21.66	18.39	16.62	16.01	15.11	15.09
2nd Asym Lag	25.08	22.57	20.43	18.63	17.19	16.21	15.60
Asym Flap	33.62	36.61	39.65	43.10	46.07	48.12	51.66
Asym Flap	34.43	37.74	41.13	44.54	47.91	51.06	53.60
Lag Sym	59.76	60.12	59.06	56.02	51.46	46.08	40.79
	92.01	86.99	80.27	79.24	79.16	79.86	82.03

Table A.2 Measured frequencies using an instrumented hammer.

Measured Frequencies (Hz)	Hammer						
Pitch (deg)	0	10	20	30	40	50	60
Flap Sym	16.18	16.70	17.41	16.69	20.33	22.29	24.28
1st Asym Lag	23.70	21.44	18.63	18.12	15.36	14.92	14.56
2nd Asym Lag	24.33	21.44	19.57	18.76	16.92	16.09	15.53
Asym Flap	33.36	35.86	39.08	41.54	43.67	46.98	49.55
Asym Flap	34.15	37.34	40.86	41.54	47.48	50.63	53.15
Lag Sym	59.67	59.82	58.30	55.28	51.07	46.86	40.54

Appendix B. Full TRAST GVT Coordinate System & Instrumentation

TRAST has several built-in sensors including the wing strain gauges and pylon accelerometers, additionally hub accelerometers were added to the hub for the GVT (Table B.1). The coordinate system varies depending on the accelerometer. The wing coordinate system is based on the orientation of the wing spar relative to the global reference frame (Figure B.1). It is defined by specifying the coordinates of the spar root in the global coordinate system ($\mathbf{P}_0^G$), and three angles which describe the wing sweep (η_z), dihedral (η_x) and angle-of-attack (α_w). Coordinates and vectors in the wing coordinate system will be designated with a superscript 'W', such as $\mathbf{P}^W$. The origin of the wing coordinate system is the point identified as the wing root such that the centerline of the wing spar is also aligned with the strain gauge measurement centerline (centerline of the box beam).

The pylon coordinate system is used to describe locations within the tiltrotor pylon. This differs from the wing coordinate system principally due to the fact that the tiltrotor pylon can rotate about the conversion axis. Coordinates and vectors in the pylon coordinate system will be designated with a superscript 'P', such as $\mathbf{P}^P$. The origin of the pylon coordinate system is the intersection between the rotor-shaft axis and the conversion axis (thus, they are assumed to intersect). The pylon coordinate system Z-axis is defined to be the rotor-shaft axis (positive forward), and the X-axis is defined to be roughly parallel to the wing Z-axis (in the wing X-Z plane) in airplane mode. Conversion angle (α_c) about a known conversion axis in the wing-coordinate system ($\mathbf{q}^W$) is defined such that $\alpha_c = 0$ in airplane mode and increases as the pylon converts to helicopter mode. Details are further described in Ref. 7.

Finally the hub coordinates are referencing the hub coordinate system with the accelerometer at the shaft centerline oriented in the pylon coordinate system. However, the other hub accelerometers are opposite the blades in roughly a blade coordinate system (X, drag, Y, span, Z, thrust direction).

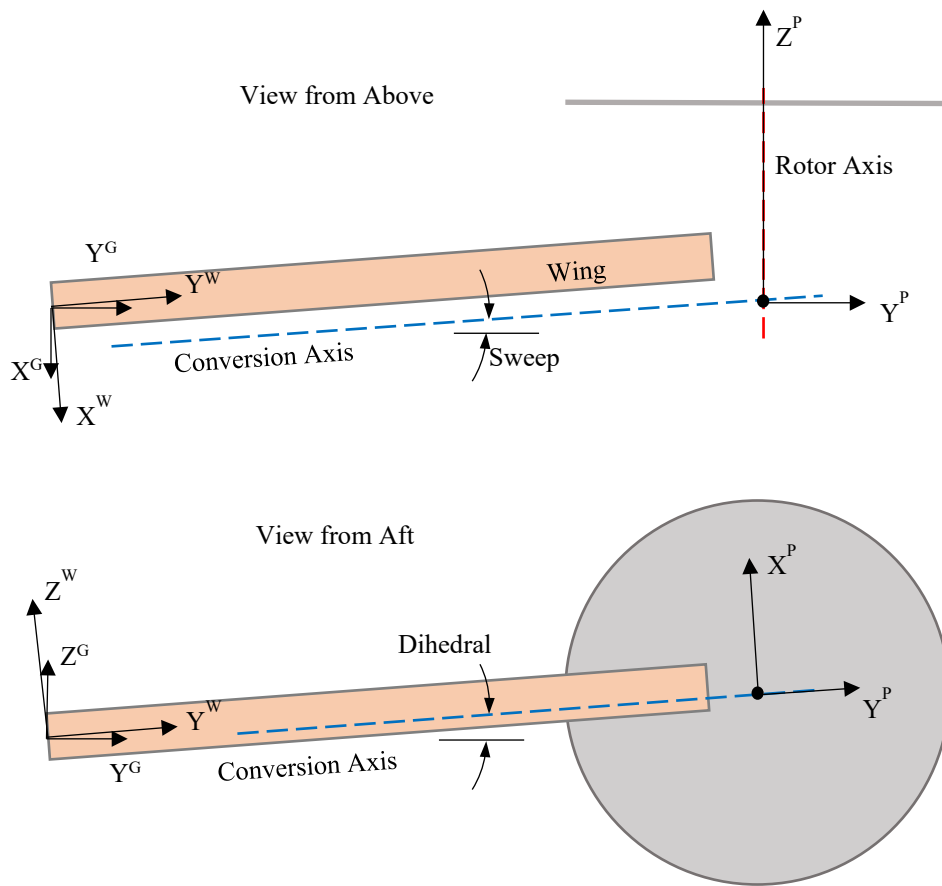


Figure B.1 Coordinate definitions used to describe the TRAST (depicted here with the TRAST in airplane mode).

Table B.1 Instrumentation used on TRAST.

Description	X (in)	Y (in)	Z (in)	Accel Direction
Hub Accelerometers				
Top of Shaft	-1.0	0.0	3.82	X ^P , Y ^P , Z ^P
Between Blade 1 and 2	-1.52	0.88	-0.65	X, Y, Z
Between Blade 2 and 3	1.75	0.0	-0.65	X, Y, Z
Between Blade 3 and 1	-1.52	-0.88	-0.65	X, Y, Z
Strain Gages				
Wing Vertical and In-Plane Bending	0	9.01	0	X ^w , Z ^w
Wing Torsion	0	10.01	0	Y ^w
Wing Vertical and In-Plane Bending	0	23.07	0	X ^w , Z ^w
Wing Torsion	0	24.07	0	Y ^w
Wing Vertical and In-Plane Bending	0	32.83	0	X ^w , Z ^w
Wing Torsion	0	33.83	0	Y ^w
Pylon Accelerometers				
Forward Bulkhead, P1	2.785	3.275	8.669	Z ^P
Forward Bulkhead, P2	-1.300	-4.450	8.669	Z ^P
Forward Bulkhead, P3	-7.570	0.000	9.089	X ^P
Forward Bulkhead, P4	-8.250	0.680	9.089	Y ^P
Forward Bulkhead, P5	-8.250	0.000	9.769	Z ^P
Aft Bulkhead, P6	-3.570	3.038	-7.506	X ^P
Aft Bulkhead, P7	-4.250	3.718	-7.506	Y ^P
Aft Bulkhead, P8	-3.570	-2.538	-7.506	X ^P