Crash Testing of Helicopter Airframe Fittings

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ABSTRACT

As part of the Rotary Wing Structures Technology Demonstration (RWSTD) program, a surrogate RAH-66 seat attachment fitting was dynamically tested to assess its response to transient, crash impact loads. The dynamic response of this composite material fitting was compared to the performance of an identical fitting subjected to quasi-static loads of similar magnitude. Static and dynamic tests were conducted of both smaller "bench level" and larger "full-scale" test articles. At the bench level, the seat fitting was supported in a steel fixture, and in the "full-scale" tests, the fitting was integrated into a surrogate RAH-66 forward fuselage. Based upon the lessons learned, an improved method to design, analyze, and test similar composite material fittings is proposed.

INTRODUCTION

The RAH-66 Comanche, like most modern rotorcraft, was designed with crash protection for the pilot and co-pilot. During a crash, the landing gear would slow the aircraft prior to fuselage ground contact. Once the fuselage contacted ground, the airframe would deform in a predictable and controlled fashion to attenuate energy and reduce airframe decelerations. The airframe structure would also maintain a protective shell around the occupants, and it would prevent the intrusion of objects into the occupied space. Crashworthy seats would keep occupant accelerations within human tolerance, and restraint systems would protect the occupants from flailing injuries.

In 1998, Sikorsky Aircraft Corporation and the U.S. Army Aviation Applied Technology Directorate (AATD) undertook a structure, materials, and manufacturing research project entitled the Rotary Wing Structures Technology Demonstration (RWSTD) program [1]. One of the primary goals was to demonstrate the application of new materials and manufacturing processes that would reduce the cost and weight of rotorcraft structure. During this program, new technologies were demonstrated on a re-engineered RAH-66 Comanche forward fuselage.

Fittings that attach crashworthy design features such as a landing gear or a seat to an airframe provide a vital but sometimes overlooked role in occupant protection. Although this structure does not usually attenuate a large amount of crash energy, it must reliably transmit large transient dynamic loads. In a project conducted in the late 1990's, Bell Helicopter Textron acknowledged the importance of such fittings and designed a new metallic

landing gear attachment fitting more tolerant to dynamic impact loads [2]. Aircraft fittings made of composite material are becoming more commonplace, and special design and analysis guidelines are needed to assure that they will reliably transmit dynamic loads during a crash scenario.

In 2002, Sikorsky Aircraft and AATD embarked upon an analytic and experimental study of surrogate RAH-66 landing gear and seat fittings that were subjected to high transient dynamic loads. It was anticipated that a better understanding of the dynamic behavior of composite parts would lead to more weight efficient, crash resistant designs in the future. The results from the seat fitting study are discussed in the following paragraphs, and these results are thought to be applicable to many other types of composite airframe fittings subjected to transient dynamic loads.

CO-PILOT SEAT FITTING DESCRIPTION

The co-pilot seat fitting, shown in Figure 1, is a single integrated composite part. The fitting was chosen to demonstrate the cost and weight savings possible by replacing 18 separate metal and composite details with a single component. The seat fitting was fabricated using the Resin Transfer Molding (RTM) process where a dry composite preform was placed into a mold, sealed, and injected with resin under pressure. The mold, shown in Figures 2 and 3, is made up of a series of mandrels nested within the upper and lower mold halves. The seat fitting consisted of two square graphite tubes into which the front left and right seat legs fit. The front seat legs attach to the square seat tubes with two 3/8" diameter pins. The square tubes are incorporated into an I-beam, which transfers the vertical seat loads to the fuselage's left and right keel

beams. Z-pin reinforcements were added in specific areas to increase load-carrying capability without adding extra plies or mechanical fasteners.

Figure 1: Co-pilot Seat Sitting

Figure 2: Co-pilot Seat Fitting Upper Mold Tool

Figure 3: Co-pilot Seat Fitting Upper Mold Tool

REPRESENTING THE CRASH ENVIRONMENT

To begin the analytic and experimental study, loading conditions representative of the crash environment were required. During the design of the RAH-66, numerous crash scenarios were simulated using the lumped mass, computer simulation code KRASH [3]. Figure 4 is an illustration of the RAH-66 KRASH model. Pre-existing crash simulations were reviewed to determine appropriate seat fitting loads.
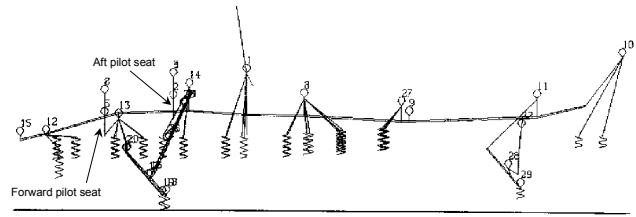

Figure 4: RAH-66 Gear-Extended KRASH Model

The highest design, crash sink speed for the Comanche is 38 fps with an aircraft orientation such that the main gear and tail gear simultaneously impact the ground. The Comanche was designed for a more extensive pitch and roll envelope at a 32 fps impact sink speed with the landing gear extended and 27 fps with the gear retracted. Reference 4 describes the RAH-66 Comanche crash requirements and analysis models in more detail.

Seat loads from the 38 fps, gear-extended case and the 27 fps, level impact, gear-retracted design case are shown in Figure 5. This data indicates that the maximum rate of load application prior to seat stroke is similar for both the forward and the aft pilots. This rate of load application is similar in both the 38 fps and 27 fps impact conditions. As a result, this 400,000 lb per sec rate of load application was chosen for the seat fitting analysis and testing.
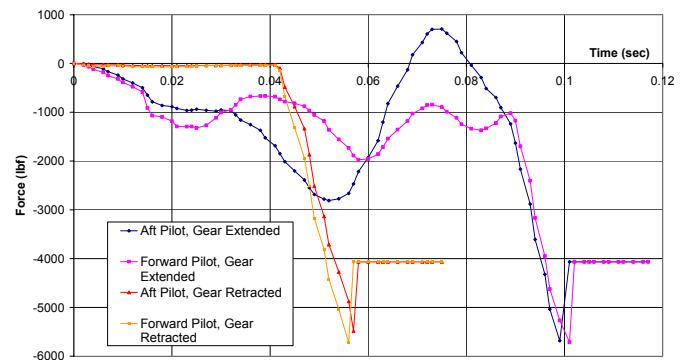

Figure 5: KRASH Seat Loads

The Comanche has load limiting crew seats that are designed to stroke at approximately 14.5 times the occupant weight. This feature limits the loads on the occupants and also limits the loads on the seat fittings. Although the co-pilot seat fitting was designed not to fail under any crash design requirement, it was the intention of this study to predict and demonstrate the actual ultimate failure load of the fitting.

For the demonstration phase of this study, the dynamic tests were conducted at increasing levels of severity until the RWSTD RTM seat fitting eventually failed. Load levels of 4K lb, 8K lb, and 20.7K lb were chosen for the dynamic bench tests. Using the load application rate of 400,000 lb per second, the time durations for the three tests would be approximately .010, .020, and .052 seconds respectively.

During the tests, a methodology was needed to apply the loads to the specimen in the appropriate time duration. A conventional drop tower with impact "cushions" between the dropping mass and the test article were chosen to achieve the desired force time history. Two wedge-shaped aluminum honeycomb cushions were used for each test. Cushions with near linear load-deflection characteristics resulted in an applied load time history that resembled one quarter of a sine wave. Figure 6 illustrates the results of an aluminum honeycomb wedge characterization test.
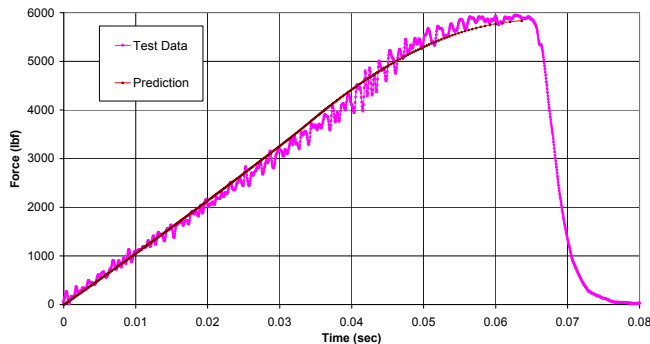

Figure 6: Aluminum Honeycomb Wedge Test Results

Besides generating the required load time history, the wedges attenuate a considerable amount of energy. The drop weight, drop velocity, wedge dimensions, and crush stress had to be selected appropriately to provide the correct load application profile. Spreadsheet software was used to predict the load time history of each test. The basic premise of the software was that all the energy of the drop mass would be attenuated by the aluminum wedges, and the energy absorbed by the seat fitting or backup structure would be negligible.

STATIC TEST OVERVIEW

Prior to conducting the bench level dynamic tests, quasi-static tests were conducted for several purposes. First, the initial static failure load of the seat fitting was established. Second, after experiencing the initial failure in the part, the capability of the specimen to sustain additional load was determined. Strain gage data along with an understanding of the progressive failure damage was then available for predicting the failure scenario during the dynamic bench tests.

The steel test fixture was designed to support the seat fitting is a manner similar to the flight article and to fit within a standard MTS test machine. The installation in the test machine is shown in Figure 7. As can be seen, a bar was used to distribute the applied load from the test machine to the two attachment locations. The seat fitting was attached to left and right steel angle fixtures with six Hi-loc fasteners. The top flange of the seat fitting I-beam continued back to form a beaded panel that attached to another heavy steel fixture via nine Hi-loc fasteners.

The force was applied gradually until the article would no longer sustain any further load. At each seat attachment location, a load cell was used to monitor the applied load. Strain gages, whose locations are shown in Figure 8, were monitored during the test along with the displacement of the platen applying the load.

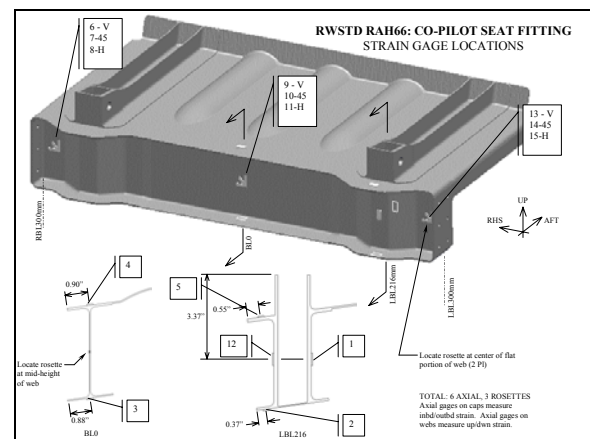Figure 7: Seat Fitting prior to static test

Figure 8 Seat Fitting Strain Gage Locations

Figure 9 shows the failed RTM fitting, Figures 10 and 11 show the shear failure, and Figure 12 shows an example of the bearing failures that occurred at the load introduction points. The ultimate failure of the test article occurred at 18,735 lbs when the outer beam shear webs failed at the attachment to the fixture.
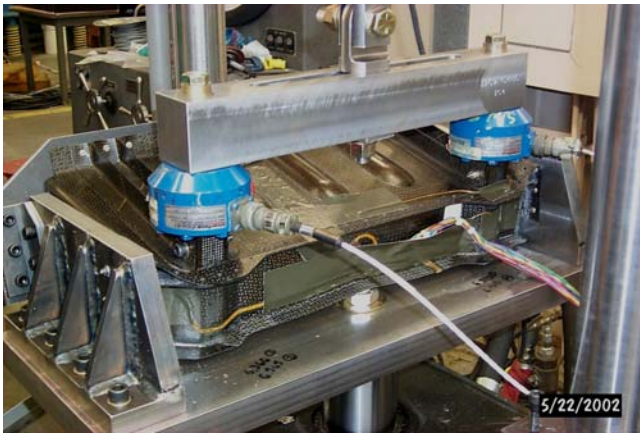
Figure 9: RTM Fitting Post Test.

Figure 10: RTM Fitting Shear Failure

Figure 11: RTM Shear Failure

Based on the static test results, the original analysis of the fitting was reassessed to better understand the failure mechanisms. For this load case, crippling of the upper flange cap between the two load application points, and shearing of the web between the load applications points and the supports were thought to be critical.
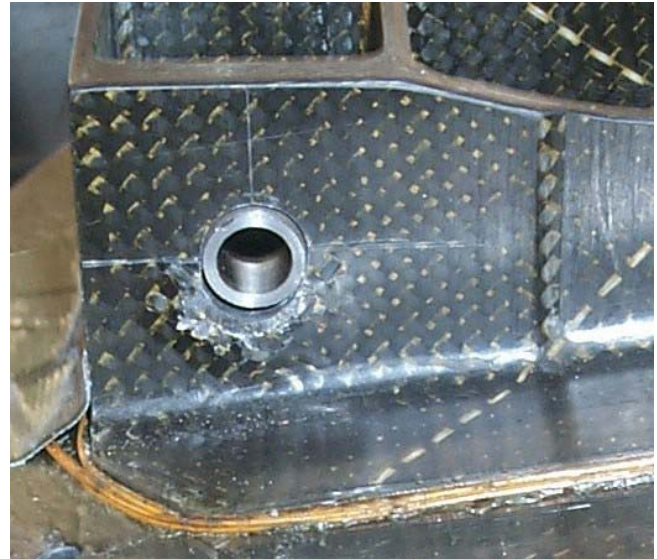

Figure 12: RTM Test Bearing Failure

After reviewing the test data and the specimen, the failure mode appeared to be a shearing off of the web from the wall. Figures 10 and 11 support this interpretation. This failure mode is very similar to that experienced in rail shear specimen tests during the RWSTD program for a prepreg material. Since the RWSTD test values were for a different resin system than used in the RTM seat fittings, the test values were reduced by the ratio of the allowable strains for the two different systems. Also, the inherent strength differences between prepreg and RTM was accounted for in adjusting the failure load.   Both of these adjustments resulted in a revised predicted failure load of 18,046 lbs. When compared to the actual failure load of 18,735 lbs for the RTM fitting, this is a remarkable correlation.

DYNAMIC "BENCH TEST" OVERVIEW

In the summer of 2002, the seat fitting was tested dynamically at the Impact Dynamic Research Facility at NASA-Langley, Hampton, VA. Figure 13 illustrates the test facility that was used to conduct these "dynamic bench tests." Two vertical tubes were used to guide a mass that was dropped from a pre-determined height onto the seat fitting.   The tubes passed partially through the two aluminum honeycomb wedges that were used to tailor the load pulse. See Figure 14.

A schematic of the hardware used to introduce the loads to the specimen is shown in Figure 15. The load application hardware was allowed to telescope inside the guide tubes so the tubes would not restrain the specimen from deflecting vertically.

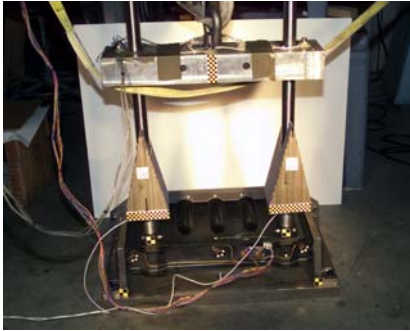Figure 13: Seat Fitting Test Facility

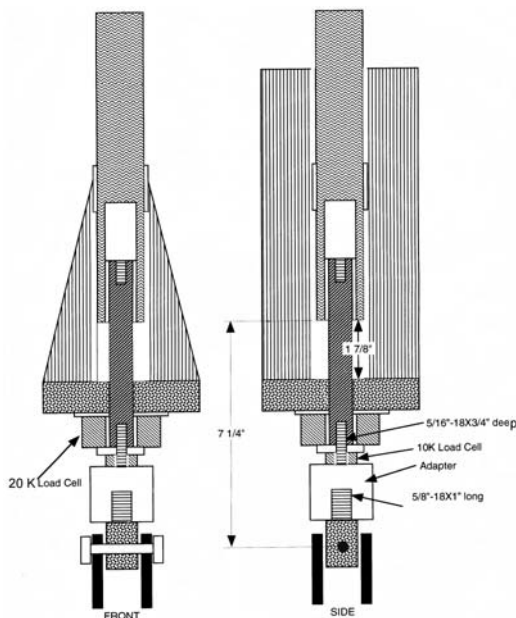Figure 14: Seat Fitting and 46 lb Drop Mass

Figure 15: Load Introduction Hardware

Instrumentation in the dynamic tests included strain gages, load cells, and accelerometers. The strain gages were applied to the seat specimen in a similar pattern to those in the static test. Two sets of redundant load cells measured the loads applied to each end of the seat specimen. Three accelerometers were mounted on the drop mass.

In the dynamic tests, it was anticipated that using a traditional method, such as an LVTD, to measure the deflection of the load applications points was going to be difficult. Instead, a video photogrammetry method to measure the displacements was proposed.

All tests were recorded with high-speed video using a single Phantom 5 camera at rates of either 1000 or 1920 frames per second. The tests recorded at 1000 frames per second had a resolution of 1024 by 1024 pixels and were 24-bit color images. The tests recorded at 1920 pictures per second were 24-bit color images of resolution 1024 by 512 pixels. The lenses used by the Phantom 5 were Nikon F mount and generally had a focal length greater than 50 mm to minimize distortions. (A wide lens was later required in the full-scale seat-fitting test because of restricted camera placement.)

The camera was placed to gather two-dimensional displacement data using traditional photogrammetry techniques. Yellow and black checkerboard targets with known dimensions were placed on the object whose motion was followed. If the motion of the object stayed parallel to the plane of the camera sensor, simple optical scaling laws applied. The sequence of uncompressed video images was used to obtain displacement time-history data at important locations on the seat fitting.

The computer program, Commotion Pro 4.1 [5], was used to conduct the motion tracking of selected points such as the seat attachment bolts and the checkerboard intersections. The output of Commotion was in-plane x and y positions of the marked targets in terms of their pixel locations from (0,0) to (1024, 1024). To convert these locations into engineering units, known distances between the checkerboard squares were compared to the pixel distances determined in the motion tracking.

Several dynamic tests were conducted in increasing level of severity from 4K lb, to 8K lb, and to 20.7K lb. In each test, aluminum honeycomb wedges of the same dimensions were used. However, the drop heights were varied from 92 to 113.1 inches, and the drop weight was varied from 46 to 628 lb.

During the testing, the accuracy of the load cell measurements was questioned. The redundant accelerometer measurements did not always predict the same load. Even though the telescoping design of the guide tubes allowed the specimen to move vertically, it is believed that the guide tubes and the deforming specimen put the load cells in bending and corrupted their measurements. Nevertheless, the accelerometers on the drop mass were consistent and were used to predict the

applied load during the dynamic tests.   Figure 16 shows the applied load on one end of the seat fitting in blue along with the predicted load in pink.
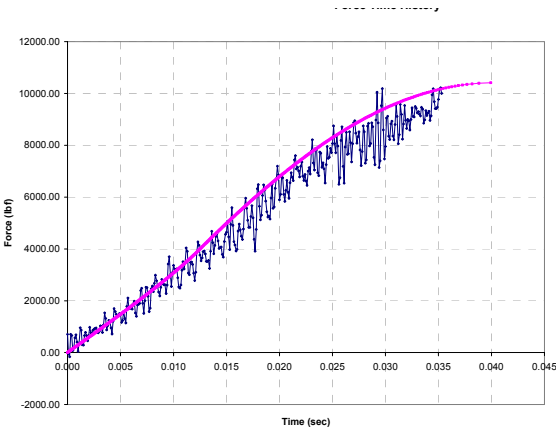

Figure 16: Load Measurements and Predictions from the 20.7K lb Dynamic Test.

DYNAMIC BENCH TEST RESULTS

The strains measured during the 4K and 8K dynamic tests were compared to the strains measured at corresponding load levels from the static test. When the peak strains measured in the 4K and 8K impact tests are compared to the strains measured for the corresponding static loads, similar trends were observed.   This comparison is summarized in Table 1.

Test	Load (lb)	Static Micro Strain	Impact Micro Strain	Difference
4K Drop	4138	1030	1192	15.7%
8K Drop	8224	2021	2330	15.2%

Table 1: Comparison of the Drop Test Strains to the Static Test Values

Based on this information, it was concluded that the impact failure load of the bench article could be up to 15% higher than the static test failure load, and the proposed drop weight for the 20.7K lb. test would be adequate to ensure failure.

In the 20.7 K impact test, the seat fitting failed at 20,440 lbs. which is an increase of 9.1% over the static test. In Figure 17, the static test failure strain on rosette ROSA-45 was 4085 $\times 10^{-6}$ in/in, while in the impact test this same gage registered 5228 $\times 10^{-6}$ in/in. This is an increase of 27% in the failure strain at this location.
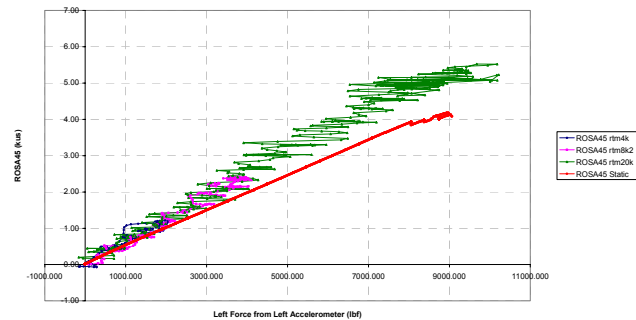

Figure 17: Comparison of the Static and Impact Failure Strains for ROCA-45

FULL-SCALE TEST OVERVIEW

An identical seat fitting was installed in the RAH-66 RWSTD forward fuselage assembly and was tested at the Aviation Applied Technology Directorate, Ft. Eustis, VA. Figure 18 illustrates the facility. The aft portion of the forward fuselage assembly was attached to the facility backstop. The assembly was also supported vertically at the main landing gear drag beam attachments.

Figure 18: AATD Drop Test Facility

Similar to the bench tests, static tests were conducted before the dynamic tests. The static test arrangement is shown in Figure 19. A load cell and an actuator were used to apply the load to the seat fitting. The first test was a simple "push" on the seat fitting to the limit load of 3000 lb, and the second was a "pull" of 5000 lb. These tests provided the baseline strain levels to use for the comparison with the impact testing.
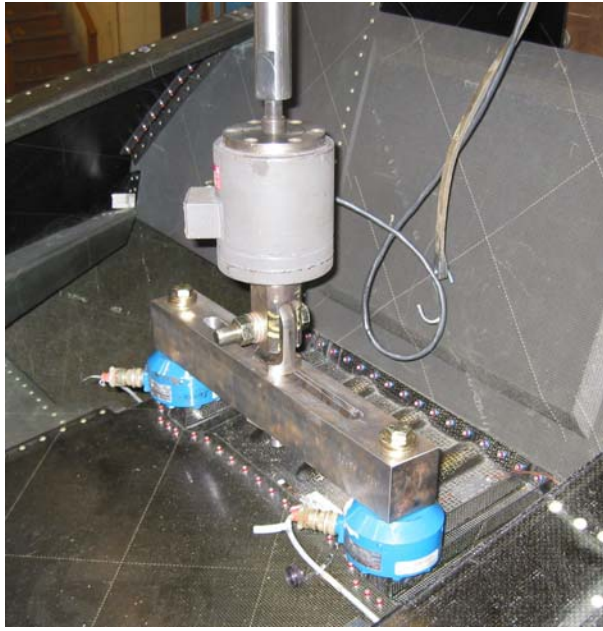
Figure 19: Seat Fitting Load Setup

Following the static test, the first two dynamic tests (4K and 8K lb) were conducted with the same drop heights and drop weights as in the dynamic bench tests. The same guide rails, drop masses, and load application hardware were used in this full-scale test. Aluminum honeycomb wedges of the same dimensions and crush strength were also used.

Based upon the information from the 4K and 8K full-scale tests, the anticipated failure load was predicted to be higher than in the dynamic bench test. (This will be further explained in the following section.) To be assured that the part would fail on the last dynamic test, the drop weight was increased from 628 lb used in the most severe dynamic bench test to 707 lb.

FULL-SCALE TEST RESULTS

In order to predict the impact failure load, information from several sources were compared. First the strain data collected during the full-scale static test and the 4K and 8K full-scale dynamic tests were compared. See Figure 20. The trend of the impact strain being some 15% higher than the static strain, as indicated in the bench testing and Table 1, continued. Additionally, the response of the fitting under load was not significantly different than the bench testing, even when the end fixity of the full-scale article was accounted for in the predictions.

Based upon this logic, it was determined that the failure load would be some 10-15% higher than the bench static failures. Since the bench static failure load was 18,735 lb, the failure load was predicted to be between approximately 20,440 to 21,400 lb. This is consistent with the bench dynamic failure of 20,440 lb. The actual force generated during the drop test was 25,694 lb. This magnitude of force

is 20% above the most optimistic estimate and 37% higher then the bench static failure load.
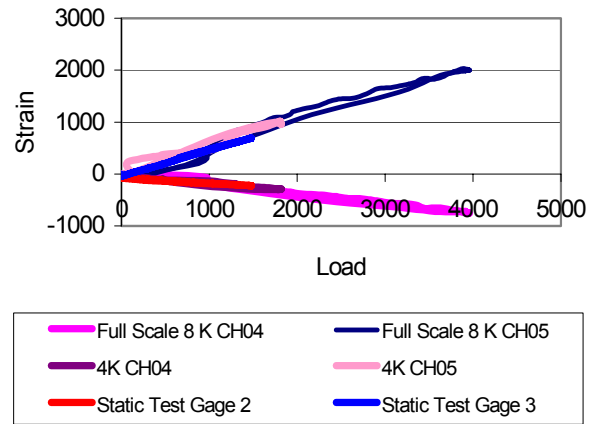

Figure 20: Strain Comparisons between Full Scale Static Test and 4K and 8K Drop Test

In addition, the only failures discovered during the inspection after the impact test were bearing failures (Figure 21) at the load introduction points as was seen in all of the previous tests to failure. The rail shear failures experienced during the bench testing were not repeated at 37% more load.
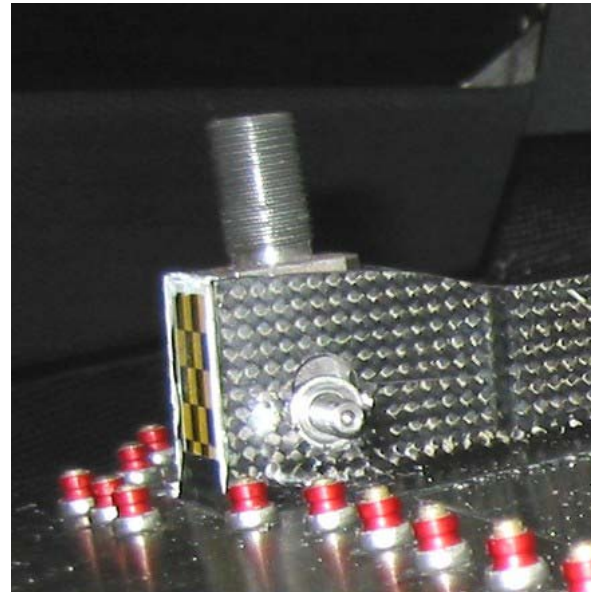

Figure 21: Bearing Failures at Load Introduction Point

CONCLUSIONS

To better summarize the structural aspects of entire seat fitting testing, Table 2 compares the failure load and failure modes for the entire seat fitting testing performed.

Test	Failure or Test Load	Impact Test % Increase over Static Test	Failure Mode
Bench Static RTM seat Fitting	18735 lb	N/A	Bearing at Load Intro. Rail Shear at Ends
Bench Impact RTM Seat Fitting	20440 lb	9.1%	Bearing at Load Intro. Rail Shear at Ends
Full Scale Impact Test RTM Seat Fitting	25694 lb	37.1%	Bearing at Load Intro

Table 2: Summary of Test Loads and Failures

From this table, three conclusions can be made. The first is that using the bench testing results to predict the full-scale failure load for the seat fitting would have been overly conservative. Even with stiffness and test strain values on both the full scale and the bench dynamic testing, the prediction of the full-scale failure load was off by dramatic amounts. The most optimistic prediction fell 20% below a load the full-scale article resisted without failure.

The second conclusion is that even with the differences between the static and dynamic bench testing, significant weight can be saved when the increased load carrying capability from impact testing is accounted for in the analysis and design. Projecting the failure load trend seen in Figure 17, the conclusion can be drawn that the use of loads developed from impact simulations as a quasi-static load along with a static ultimate strain allowable is conservative for the prediction of the final failure load by at least 9%. Since large portions of the fuselage structures are designed using "crash" loadings, the weight of a composite fuselage could be reduced by up to 9%. On utility class helicopters, this could mean as much as a 50 lbs weight savings.

Finally, although it was not a big part of this RWSTD project, the application of large-deformation finite element analysis may be very helpful in the design of similar composite fittings in the future. The flexibility of the fuselage structure and the apparent increase in strength could be accounted for in the dynamic simulations. Applying a large-deformation finite element methodology could result in considerable weight savings for crash design conditions. Based on the testing performed in this effort, an additional 15% may be possible. With this level of weight savings potentially available, the additional cost of this analysis to make predictions and the cost to confirm the predictions by limited testing could be easily justified.

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