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Accelerated Fatigue Measurements Using a Vibration Assembly Which Loads Two Specimens Simultaneously

Fatigue testing is notoriously slow, requiring many hours to obtain 1 data point on an S–N curve. Such tests can be accelerated somewhat using vibration-based techniques, which load on the order of kHz, such that tests which would traditionally have taken multiple days can now be completed in a few hours. This work presents a novel assembly that can further accelerate fatigue testing campaigns by loading multiple specimens simultaneously. The assembly has two resonant modes of interest. The lower frequency mode puts a slightly higher strain on one specimen, and the higher frequency mode puts a slightly higher strain on the other specimen. By alternating between the two modes, fatigue loads are rapidly applied until one specimen fails, during which the unfailed specimen will have experienced the same number of cycles, and therefore has less time remaining until its own failure. The test is paused while the failed specimen is removed and replaced by a fresh specimen. Testing continues in this fashion, alternating between the two resonant modes, until a sufficient number of data points have been maintained to map out an S–N curve. Because the assembly loads two specimens simultaneously, the system is up to twice as efficient compared to testing all specimens individually. [DOI: 10.1115/1.4068699]

Keywords: high-cycle fatigue (HCF), vibration-based fatigue testing, digital image correlation (DIC), high-throughput measurements, failure criteria, mechanical properties of materials, stress analysis

1 Introduction

Aircraft components often experience high-frequency vibrations during flight, which can result in high-cycle fatigue [1]. For example, turbine engine blades can accumulate on the order of 10^7 bending cycles over their design life [2]. Since fatigue strength occurs below ultimate and yield strength, materials can experience unexpected failure during operation. To accurately predict the fatigue life of a material, repeated fatigue testing is required to develop an S–N curve for the turbine material. However, high-cycle fatigue experiments are notoriously slow, requiring many weeks to generate S–N curves. Traditional fatigue testing techniques such as servo-hydraulic tests operate in lower frequency ranges, near 40 Hz [3]. At this frequency, a single sample takes 70 h to accumulate 10^7 cycles, corresponding to one data point on an S–N curve. Additionally, a uniaxial fatigue test as performed in a servo-hydraulic setup is not characteristic of the vibrational bending fatigue experienced in a turbine, due to the low frequency and uniaxial loading. Rotating beam tests can

operate at higher frequencies than servo-hydraulic tests, at 1000 rpm (167 Hz), but still require many hours per data point [4]. Fatigue modeling is also a stochastic process that is highly sensitive to the unique microstructure [5] and surface finish [6] of individual test specimens, and thus experiments must be repeated under the same operating conditions to ensure proper statistical treatment. By reducing testing time in populating an S–N curve, materials can be characterized much more quickly and/or a larger sample size can be used to account for material variability. Additionally, the shortened times allow for efficient testing of a variety of surface finishes.

To improve high-cycle fatigue testing times, one notable vibration-based technique was pioneered by George et al. [7]. This technique uses a square plate specimen cantilevered to a mounting block on an electrodynamic shaker. The specimen is actuated out-of-plane to excite a resonant mode that occurs near 1600 Hz, resulting in the same number of cycles 40x faster than an axial load frame [7]. This cantilevered vibration technique has been applied to common turbine engine materials such as Al-6061-T6 [8], Ti-6Al-4V [9], and Ni Alloy 718 [10]. In each case, the vibration-based method produced S–N curves from about a dozen data points each in only tens of hours versus the hundreds of hours required for axial testing.

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To reduce material waste during these tests, Bruns et al. [11] modified George's technique by replacing the uniform square plate with a modified carrier plate for which material was removed from the center of the free edge and a rectangular specimen was bolted in place. The carrier plate can be reused for multiple tests, while the inserted specimen is fatigued to failure and replaced. However, one limitation of the Bruns–Zearley design was that the insert specimens occasionally cracked at the tab interface where strain was not monitored. To resolve this issue and improve the repeatability of these insert specimen tests, Scott-Emuakpor et al. [12] replaced the rectangular specimen with an hourglass-shaped insert. The narrowed gauge region ensures that failure will occur in the center of the specimen, where it can be easily monitored by instruments. These specimens have since been implemented to test the fatigue properties of Ti–6Al–4V [9].

The work in this article further improves upon the carrier-insert design to improve throughput during testing. The base carrier plate was modified to accommodate two-insert specimens that are fatigued simultaneously [13]. Two different resonant modes are used, one to target each specimen. The test alternates between these two modes, fatiguing one specimen to failure and partially fatiguing the second before switching modes to focus on the second specimen. When a specimen fails, it is replaced with a new specimen and testing resumes. By testing in this way, twice as many fatigue experiments can be performed in the same amount of time. This reduction in testing time can be applied to both industry testing and research on experimental materials, decreasing overall testing costs.

2 Materials and Methods

Unlike the single-insert carrier plate, the two-insert carrier features a pair of closely related modes that achieve subtly different strain fields in the samples. A finite element model of these mode shapes and the associated strain fields are shown in Fig. 1. Notably, the third mode of the two-insert carrier, at 896.4 Hz, puts the outer specimen under a larger bending strain than the inner specimen. Conversely, the fourth mode, at 994.3 Hz, puts the inner specimen under the larger strain. A finite element model for the carrier predicts the strain ratio of the low-strain sample to the high-strain sample to be 94.2% for the third mode and 90.3% for the fourth mode. The existence of these two closely related modes enables simple test plans that alternate between these two modes to step through sequentially lower constant strain measurements.

A generalized test plan that utilizes the twin fatigue modes of the two-insert carrier is shown in Fig. 2. The first step of the test utilizes the third mode of the carrier to achieve a strain amplitude of ϵ_1 in the outer sample and ϵ_2 in the inner sample. The test runs until the outer sample fails at cycle count N_1 . At this point, the failed outer sample is replaced, and the test is restarted at the fourth mode, which places a greater strain on the inner sample. The strain amplitude is chosen such that the inner sample maintains a strain amplitude of ϵ_2 , as it

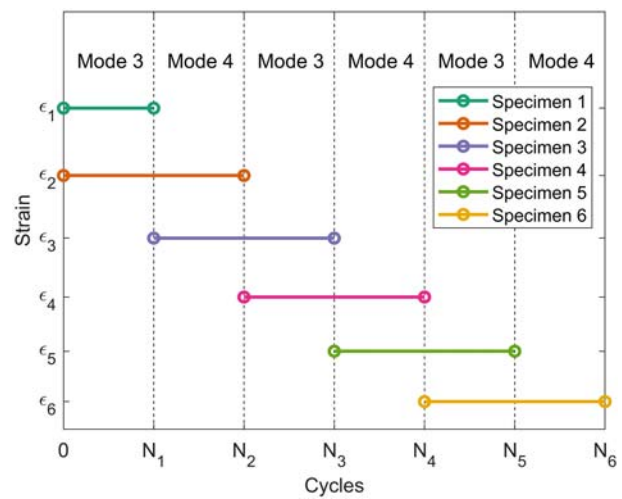


Fig. 2 Generalized test plan for the two-insert carrier assembly

did before the first sample failed. However, now the outer sample has a lower strain amplitude of ϵ_3 . The test is run until the inner sample fails at cycle count N_2 . The inner sample is then replaced, and the entire system is excited at mode 3 once again. This process of failing specimens and alternating modes repeats until the strain amplitude is too low to induce fatigue damage in a reasonable span of time.

The efficiency of the two-insert carrier can be readily deduced from Fig. 2 [14]. After N_6 load cycles applied by the shaker, samples 1, 3, and 5 accumulated a combined total of N_5 cycles, whereas samples 2, 4, and 6 accumulated a combined total of N_6 cycles. In general, a test plan that fatigues k samples achieves a cycle efficiency of

$$N_{\text{eff}} = \frac{N_k + N_{k-1}}{N_k} = 1 + \frac{N_{k-1}}{N_k} \quad (1)$$

In other words, for every load cycle applied by the shaker system, N_{eff} useful fatigue cycles are generated. Thus, the two-insert carrier tends to be more efficient as the number of failed samples increases, with the efficiency approaching 2 in a limiting sense.

2.1 Experimental Setup. Six carrier plates and their accompanying samples were cut from a 0.120 in. thick sheet of 6061-T6 aluminum. One of the carrier plates was used for a traditional modal analysis, while the remaining plates were used in fatigue tests. All samples were prepared for fatigue by hand polishing with successively higher grits of silicon-carbide sandpaper, up to 600 grits. The samples were then fastened into the carrier plates. The screws that join the carrier and samples were torqued to 85 in-lbs. The carrier plates were then cantilevered to a Data Physics V617

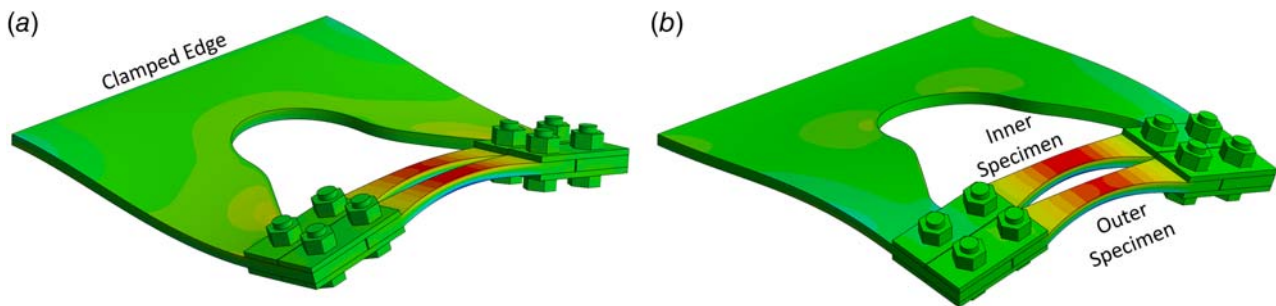


Fig. 1 Fatigue-inducing mode shapes and uniaxial strain contours for the two-insert plate. (a) Third resonant mode at 941.2 Hz. The outer specimen has the higher strain. (b) Fourth resonant mode at 1046 Hz. The inner specimen has the higher strain.

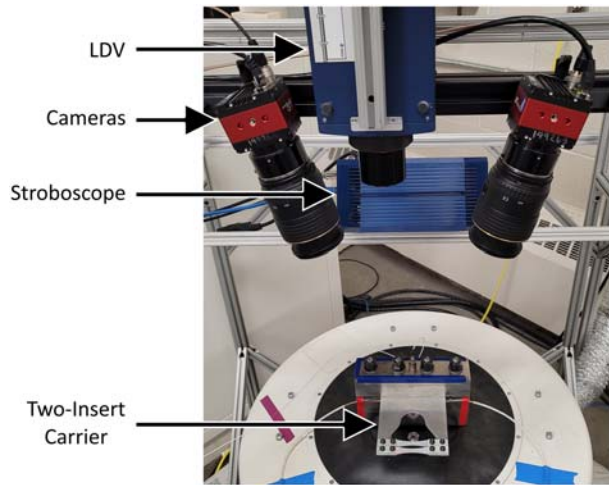


Fig. 3 Experimental test setup showing the LDV, cameras, stroboscope, and the two-insert carrier plate

electrodynamic shaker via a clamping block made from 304 stainless steel.

A Polytec laser Doppler vibrometer (LDV) is used to measure the velocity of the samples (OFV-505 sensor head and OFV-5000 controller). The built-in VD09 velocity decoder is used to decode the Doppler signal into a real-time velocity output, which is used as the control signal for the resonance-dwell test. A velocity range of 10 m/s with a frequency bandwidth of 1.5 MHz was chosen for the velocity output. The laser was positioned at the edge of the outer specimen. Retro-reflective tape was used to ensure a good signal-to-noise ratio from the LDV and prevent signal dropouts due to the random nature of back-scattered light. The LDV velocity signal also acted as the phase reference for camera synchronization.

Two Allied Vision Prosilica GT6600 cameras were positioned above the carrier and samples for DIC as shown in Fig. 3. A high-contrast pattern was applied to the samples using VHT spray paint. During fatigue tests, images were taken alternately between maximum and minimum displacement to ensure that crack propagation was always captured [15]. Displacements are extracted from stereo image pairs using the vic-3d software package from Correlated Solutions. A subset size of 29 and a step size of 7 are used in the correlation. Although they happen to also be the default values upon first installing vic-3d, they were chosen intentionally after also exploring other neighboring values. A cracked-beam regression model [16] is used to assess crack propagation from images captured during the fatigue test.

To enable real-time strain control during the fatigue test, specimen strain and velocity are correlated via a linear relation. To determine the coefficients of this relation, strain is measured during a series of resonance dwells at varying velocity levels. At each velocity level, 100 images are captured: 50 at maximum displacement and 50 at minimum displacement. The crack-free regression model presented in Ref. [16] is used to estimate the peak strain in the specimen for each of the images. The strain amplitude is then computed based on the difference in strains between the images captured at maximum displacement and minimum displacement.

To validate the strain-velocity relation, the plate used for modal analysis additionally had strain gauges adhered to the samples. This is shown in Fig. 4. Enamel-coated copper wires are routed from the strain gauges to bonded terminals at the clamped edge of the carrier. This is done to minimize mass and stiffness effects on the carrier stemming from the strain gauge hookup. Vishay 2310B signal conditioners are used to measure the change in resistance of the gauges. Shunt calibration is performed to measure the total system gain of the Wheatstone bridge contained in the signal conditioners, resulting in minimal measurement uncertainty from the signal conditioner itself (<0.27% typical). Consequently, the majority of the

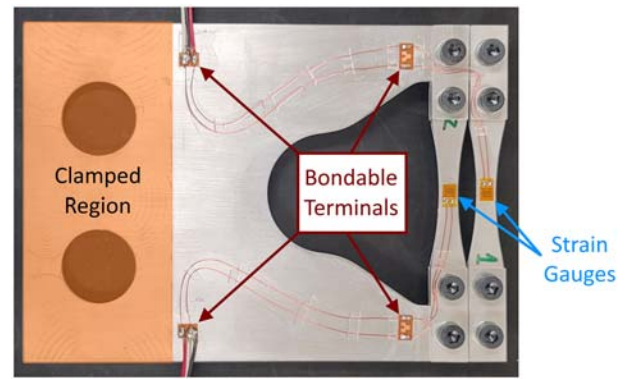


Fig. 4 Carrier plate with strain gauges on the specimens. Copper wires (34 American Wire Gauge) run from the gauges to bondable terminals at the clamped edge of the carrier plate.

uncertainty in subsequent strain gauge measurements stems from the manufacturer's rated uncertainty of the gauge factor ($\pm 0.5\%$).

3 Results

3.1 Modal Response. The frequency response of the carrier plate, as measured by the LDV and the strain gauge on the outer specimen, is shown in Fig. 5. Five modes are detected in the range of 50–1500 Hz, three of which are bending modes and the two remaining are twisting modes. The third and fourth modes are located at 869.8 Hz and 962.2 Hz, respectively, and are indicated by the dashed vertical lines in Fig. 5. These natural frequencies are somewhat lower than those predicted by the finite element model (896.4 and 994.3 Hz, respectively), which suggests that the physical assembly is slightly more compliant than the model.

Both fatigue modes are lightly damped, with damping ratios estimated at 0.077% and 0.100% for the third and fourth modes, respectively. Consequently, the modes are relatively easy to excite to fatigue-level strains. The strain sensitivity was measured to be on the order of 250 $\mu\epsilon/g$ of applied excitation. Ordinarily, such lightly damped modes would make resonance dwells hard to implement. Small fluctuations in frequency can result in large amplitude excursions due to the sharpness of the peak. However, both fatigue modes tended to exhibit some nonlinear hardening at higher excitation levels, which flattens the peak of the mode, thereby reducing the amplitude sensitivity.

Due to the differences in natural frequencies between the physical carrier plate and the finite element model, there was some concern that the mode shapes might be adversely affected. The measured mode shapes for the carrier plate and samples are compared to the finite element model in Fig. 6. For both modes, the measured displacement fields show generally good agreement with the finite element analysis (FEA) model, despite the difference in natural frequency between the model and the experiment.

3.2 Strain Velocity Calibration. A crack-free regression model is used to measure the strain in the samples [16]. A typical example of the residuals of the regression model is shown in Fig. 7. For a sample bending out-of-plane by 0.5 mm, the standard deviation of the residuals is on the order of 300 nm; lower at the center of the sample where the cameras have better focus and higher at the edges. The coefficient of determination (R^2) of the regression exceeds 0.99999 in all cases. In other words, over 99.999% of all variance in the measurement is explained by the regression model.

The strain-velocity calibration curves for the third and fourth modes are shown in Fig. 8. In all cases, the strain gauges show good agreement with their DIC counterparts. At very low-strain

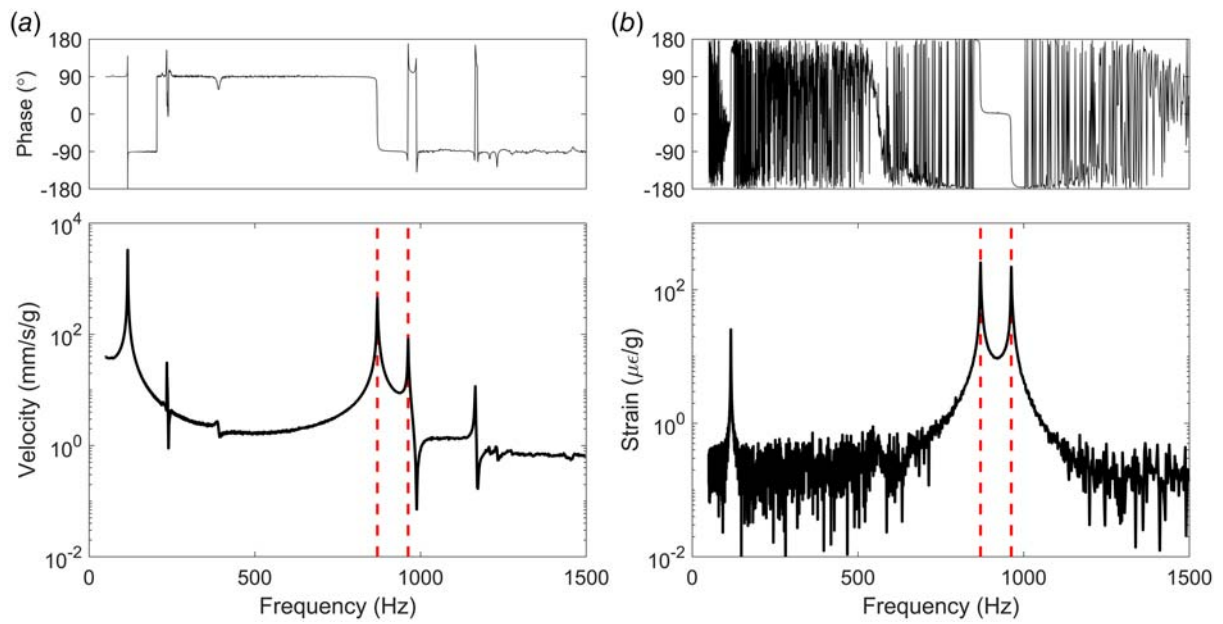


Fig. 5 Frequency response of the two-insert carrier plate. The fatigue-inducing modes are indicated by the dashed vertical lines. (a) The velocity response of the carrier and samples. (b) The strain response of the samples.

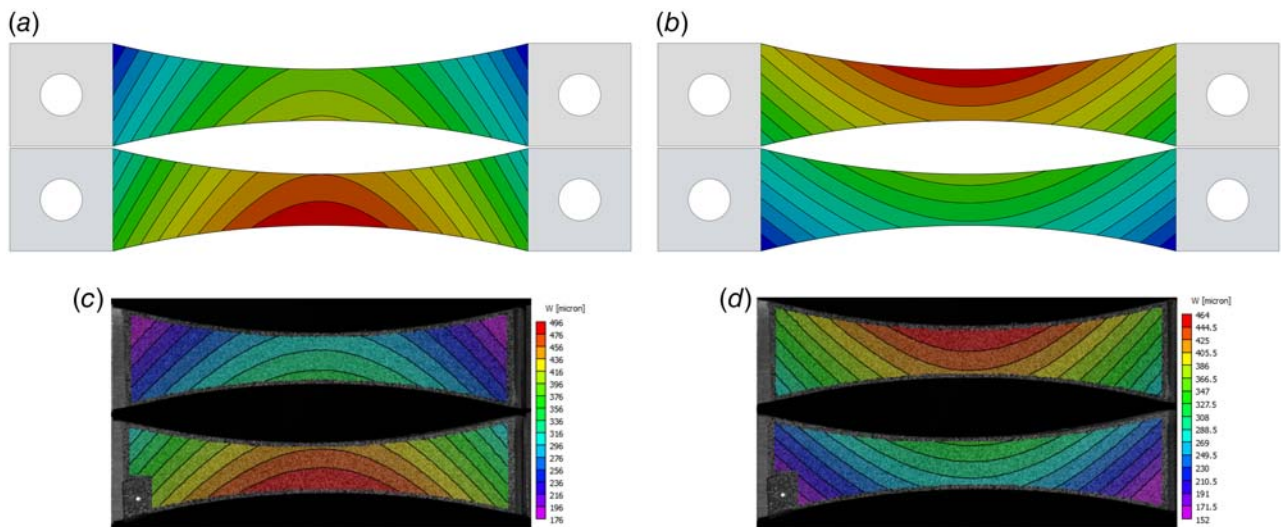


Fig. 6 Out-of-plane displacement comparison between the finite element model and DIC measurements: (a) mode 3 finite element contour, (b) mode 4 finite element contour, (c) mode 3 DIC measurement, and (d) mode 4 DIC measurement

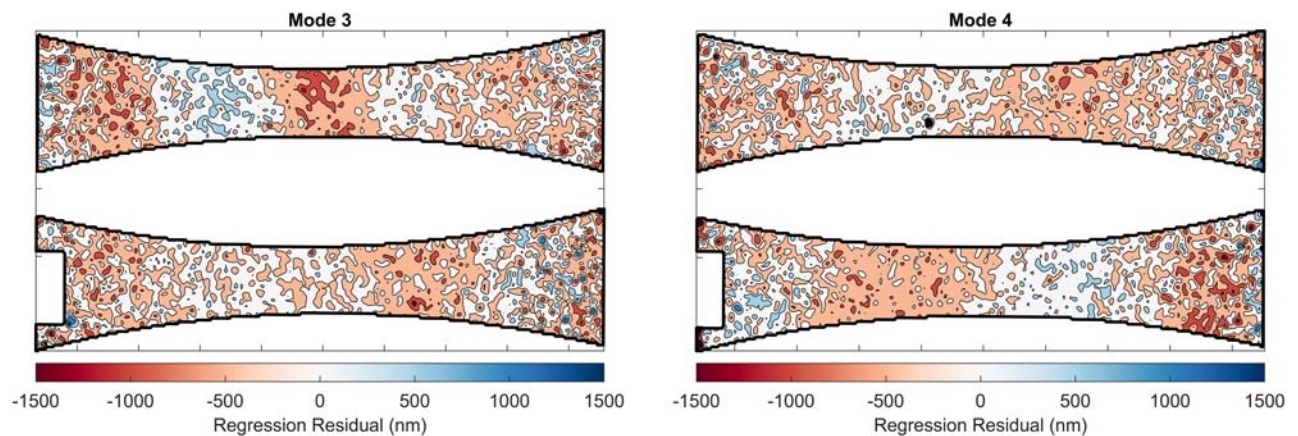


Fig. 7 Out-of-plane displacement residuals from the applied regression model

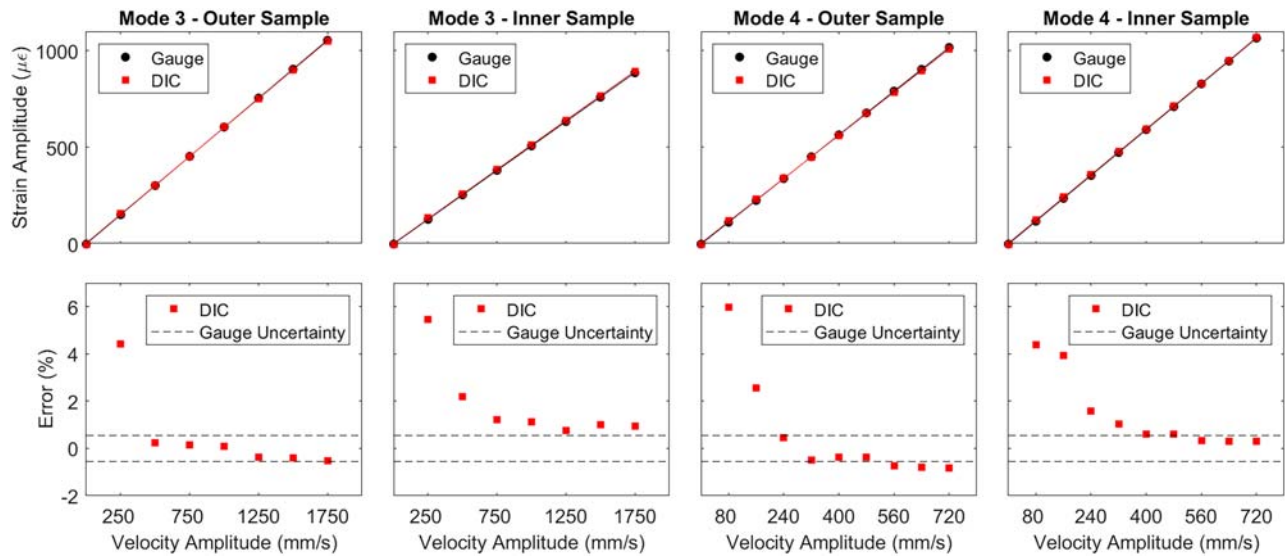


Fig. 8 Strain–velocity calibration curves for the third and fourth modes, and outer and inner samples of the two-insert carrier plate. The upper row shows the strain–velocity calibration curves as measured by the strain gauge and DIC. The lower row corresponds to the error of the DIC measurement relative to the strain gauge.

levels, DIC measurements are observed to slightly overpredict the strain, which agrees with past studies utilizing solid rectangular plates [17]. As the strain increases, the DIC measurements quickly stabilize to a constant percentage error relative to the strain gauges. DIC tended to slightly underpredict the strain on the outer sample, whereas it tended to overestimate the strain on the inner sample. However, in all cases, DIC estimates of the strain were within 1% of the gauges at amplitudes above $600 \mu\epsilon$.

For reference purposes, the 95% confidence band for the strain measurements is also included in Fig. 8. This confidence band is a result of the manufacturer’s rated uncertainty for the gauge factor and the uncertainty of the total system gain of the signal conditioner after shunt calibration. DIC estimates of the strain tend to fall within or very near the confidence interval of the gauges in all four calibration curves. It should also be noted that the reported strain gauge uncertainty does not include the effects of temperature sensitivity, transverse sensitivity, or gauge misalignment. Thus, in reality, the confidence interval for the strain gauge measurements is expected to be slightly larger than what is reported here.

The strain sensitivity of the third mode was measured to be $0.602 \mu\epsilon/\text{mm/s}$ for the outer specimen and $0.506 \mu\epsilon/\text{mm/s}$ for the inner specimen, resulting in a strain ratio of 84.0%. For the fourth

mode, the strain sensitivity was measured at $1.413 \mu\epsilon/\text{mm/s}$ for the outer sample and $1.479 \mu\epsilon/\text{mm/s}$ for the inner sample, resulting in a strain ratio of 95.5%. In comparison, the FEA model reported strain ratios of 94.2% and 90.3% for the third and fourth modes, respectively. Thus, the strain ratio of the fourth mode showed reasonable agreement with the finite element model, whereas the third mode was much lower than expected. A ramification of the difference in strain ratio between the model and the experiment is that less load steps are achievable before the specimen strains are too low to accumulate meaningful damage.

3.3 Fatigue Results. In total, 13 fatigue tests were run on five carrier plates, however, only eight samples were fully fatigued. The details of these tests are given in Table 1. To summarize, the first carrier plate was used in four fatigue tests, with the first test utilizing the third mode. Three tests were completed with this carrier, fatiguing specimens A–C, before the carrier plate itself failed during the fourth test. The second carrier performed similarly to the first but instead started with mode 4, completing three fatigue tests on specimens F–H, but failing during the fourth. The third carrier managed to start in mode 4 but at a smaller amplitude, completing two fatigue tests on samples L and K but failing during the third. Finally, the last

Table 1 Fatigue test details for five carrier plates across 13 tests

Carrier no.	Test stage	Outer specimen	Inner specimen	Fatigue mode	Outer specimen strain amplitude ($\mu\epsilon$)	Inner specimen strain amplitude ($\mu\epsilon$)	Failure site
1	1	A	B	3	3002	2667	Specimen A
	2	C	B	4	2560	2692	Specimen B
	3	C	D	3	2581	2290	Specimen C
	4	E	D	4	2263	2304	Carrier plate
2	1	G	F	4	2813	3010	Specimen F
	2	G	H	3	2822	2531	Specimen G
	3	I	H	4	2378	2512	Specimen H
	4	I	J	3	2382	–	Carrier plate
3	1	L	K	4	2591	2904	Specimen K
	2	L	M	3	2627	2352	Specimen L
	3	N	M	4	–	2382	Carrier plate
4	1	O	P	3	2452	2294	Carrier plate
5	1	R	Q	4	2042	2391	Carrier plate

Note: Specimens J and N were unmonitored dummy specimens, expected to have strain amplitudes too low to accumulate significant fatigue damage.

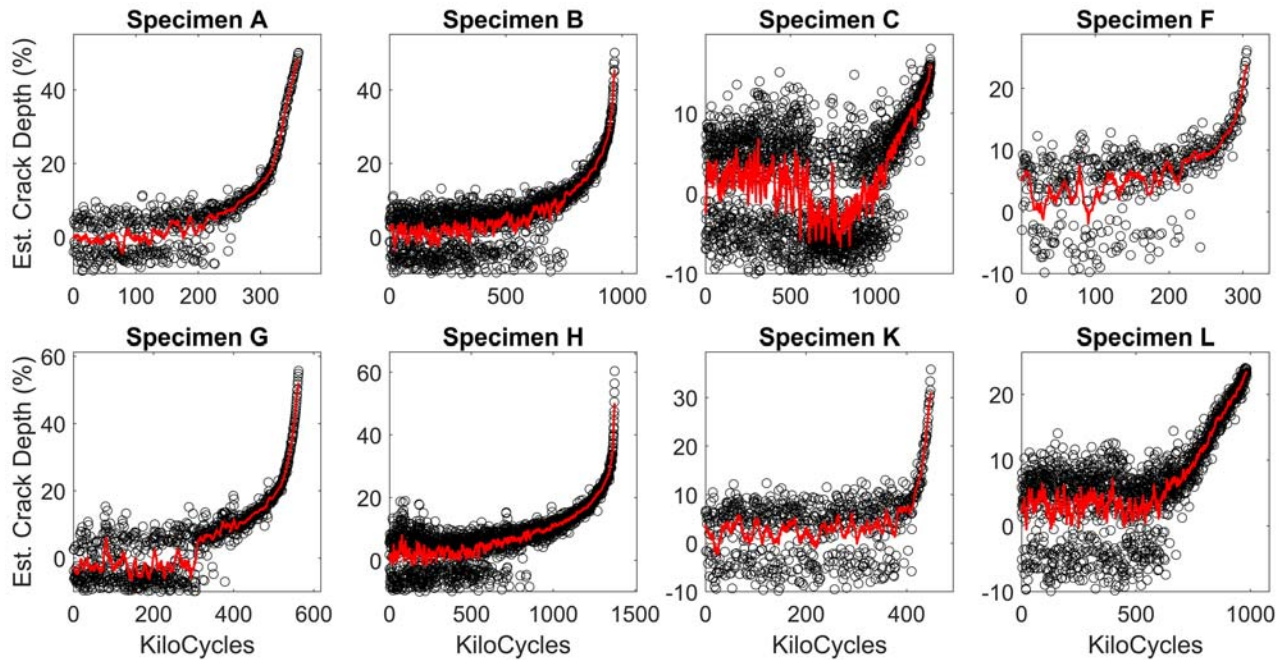


Fig. 9 Estimated crack depth as a function of cycle count for the eight fatigued samples. The black circles are the measured values, whereas the lines are a 15-point moving average.

two carrier plates were used in low-amplitude tests, but ultimately both carriers failed before their respective tests could be completed. In all cases, the carrier plate failed during long-duration, low-amplitude fatigue tests.

For the eight fatigued samples, the estimated crack depth over their fatigue lives is shown in Fig. 9 using the methods described in Ref. [16]. These data provide a means for assessing fatigue life in a consistent manner, and independently of the natural frequency of the carrier plate. The convention adopted here is that the fatigue life corresponds to the accumulated cycles when a moving average of the crack depth exceeds 20%. Under this definition, specimens A and F were the shortest-lived, both lasting slightly over 300,000 cycles. Specimens C and H were the longest-lived, both lasting over 1,300,000 cycles. Notably, there was some variance in crack propagation rates after nucleation. For example, specimen K had a very rapid propagation step, as indicated by the relatively steep

slope near the end of the test, whereas samples B and H had comparatively slow propagation rates, as indicated by relatively mild slopes compared to the scale of the horizontal axis.

The Wöhler diagram for the eight damaged samples is shown in Fig. 10. The figure also includes measurements reproduced from Ref. [11] that utilize the single-insert carrier plate for the same aluminum alloy. Notably, all eight fatigue measurements utilizing the two-insert carrier fall within the 95% confidence interval of the prior work.

4 Discussion

Figure 10 provides good evidence that the two-insert carrier can produce fatigue measurements consistent with the established literature. The measurements for both present and past work plot linearly on a logarithmic scale. This is expected by Basquin's equation [18], which states that fatigue life in the elastic regime is described by

$$\sigma_a = \sigma_f (N_f)^b \quad (\text{MPa}) \quad (2)$$

where σ_a is the stress amplitude, σ_f is the fatigue strength coefficient, N_f is the number of cycles to failure, and b is the fatigue strength exponent. Regression against the present data produces fatigue strength parameters of $\sigma_f = 938 \pm 155$ MPa and $b = -0.119 \pm 0.012$. In comparison, the measurements made in Ref. [11] produce fatigue strength parameters of $\sigma_f = 561 \pm 313$ MPa and $b = -0.0824 \pm 0.0408$. Thus, the 95% confidence intervals overlap between measurements. Although the two-insert carrier plate produces reasonable fatigue data, there are two major deficiencies in the two-insert carrier design: accelerated wear and strain amplitude fluctuations.

4.1 Rapid Wear of the Carrier Plate. In all five carrier plates, severe fretting was observed in two regions: at the clamped edge and on the circumference of the washer pressure cone where the sample is attached to the carrier. These are the two regions in which cracks were found on all five carrier plates. Specifically, carriers 2 and 4, which failed during mode 3 fatigue tests, had cracks

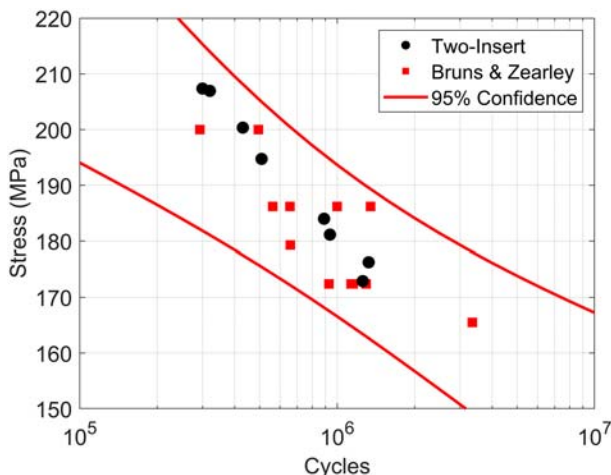


Fig. 10 S-N curve for 6061-T6 aluminum in uniaxial bending. Circles are from the present study. Squares and their accompanying confidence intervals are from Ref. [11].

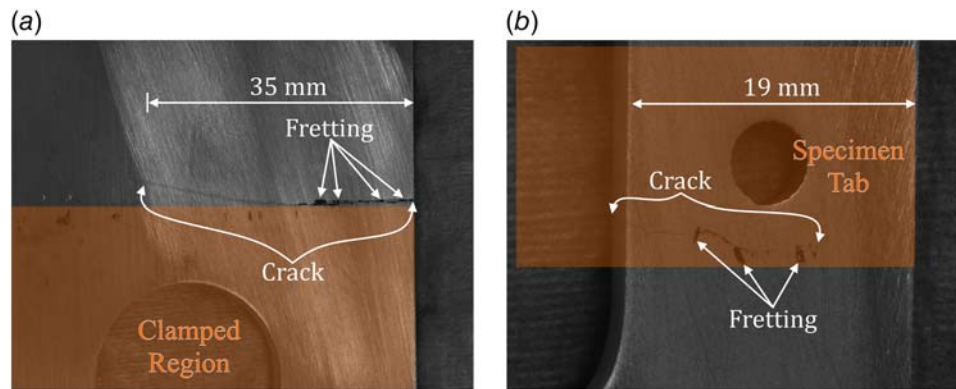


Fig. 11 Cracks on two different carrier plates: (a) crack at the clamped edge and (b) crack under the tabs connecting the specimen to the carrier plate

located at the clamped edge, as shown in Fig. 11(a). Carriers 1, 3, and 5, which failed during mode 4 fatigue tests, instead had cracks located on the washer pressure cone, as shown in Fig. 11(b). Thus, both fatigue modes seem to be exceptionally damaging to the carrier plate in different ways.

To further investigate, the finite element model was updated with mesh refinement near the locations where cracks were observed on the physical carriers. The von Mises stress contours for the two fatigue modes of the carrier plate are shown in Fig. 12. In mode 3, a stress singularity is found at the corners of the clamped edge. Similarly, in mode 4, a stress singularity is observed near the connection point between the specimens and the carrier. Spatially, both stress singularities agree with where cracks were observed to form in the physical carriers.

The stress in the carrier plate cannot be meaningfully evaluated at the locations of the singularities. To obtain an estimate of the stress at these locations, the stress is instead evaluated on a semicircular contour (of radius 1 mm) around the singularities. The stress is then averaged over the contour. Unlike the stress at the singularities, this estimate of the stress converges as the mesh is refined. Using this approach, the finite element model reports a stress ratio between the specimen and the carrier of 1.58 for the third mode and 0.79 for the fourth mode. Based on these results, it is not surprising that the fourth mode tends to rapidly fatigue the carrier plate. Indeed, based on this estimate of the stress ratio, the carrier plate should always be expected to fail before the specimens when the fourth mode is utilized. In reality, this was clearly not

the case as carriers 1, 2, and 3 fatigued multiple samples before failing. Thus, it is likely that the stress ratio is slightly closer to 1 than the finite element model would suggest. In contrast, it is somewhat surprising that the third mode also rapidly damages the carrier, as the stress in the specimens should be substantially higher than at the clamped edge. The presence of cracks at the clamped edge suggests that fretting is the primary mechanism behind the nucleation of cracks in this region. In sum, these two regions of the carrier plate are problematic, and special care should be taken to minimize the stresses in these regions in future work.

One might reduce fretting using larger fastening tabs or washers (thereby distributing contact stresses over a larger area) or by adjusting the plate dimensions to put less stress at the clamped edge. When determining plate dimensions, there is a clear tradeoff between ensuring that the resonant frequency is sufficiently isolated from other neighboring modes and ensuring that the strain is sufficiently concentrated in the specimens. A determined user might reasonably skew more heavily toward prioritizing the strain isolation, but this is out of scope of this specific article.

Equation (1) provided a quick metric of how much more efficiently this system applies fatigue cycles relative to testing all specimens sequentially at the same frequency. Because the assembly loads two specimens simultaneously, the upper limit of the efficiency is 2. However, due to the rapid wear on the plate, the efficiency never reaches 2, but remains greater than 1. In other ongoing work, we have explored variant assemblies with up to six simultaneous specimens [14,16] and have achieved efficiencies

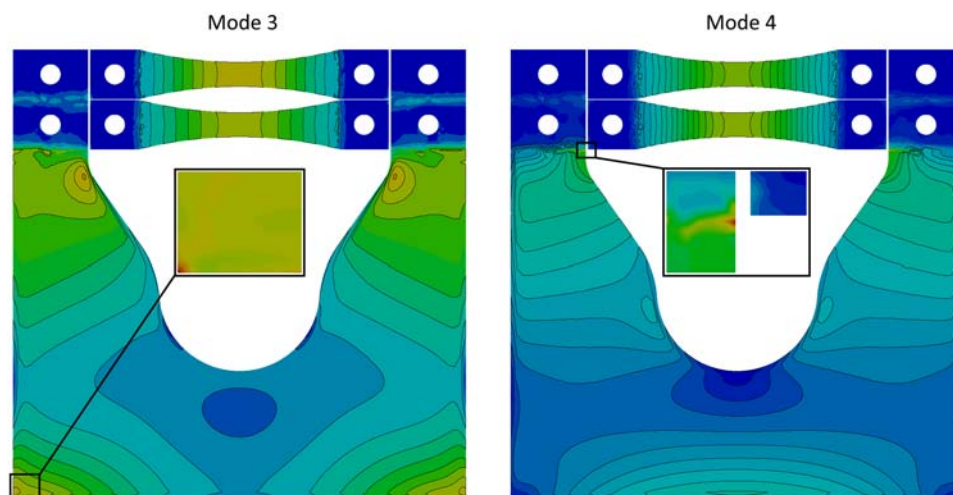


Fig. 12 Finite element von Mises stress contours for modes 3 and 4 of the two-insert carrier plate. Mode 3 has a large stress concentration at the corner of the clamped edge. Mode 4 has a large stress concentration near the connection between the carrier and the inner specimen.

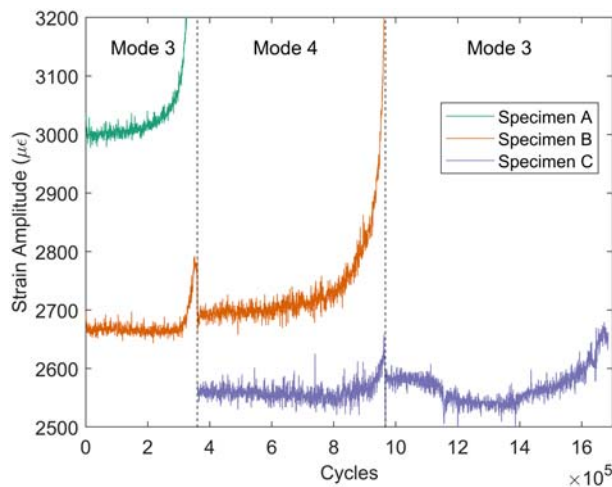


Fig. 13 Strain amplitude over the fatigue lives of specimens A, B, and C

on the order of 4–5. However, we feel that this two-insert assembly is still worth publishing because: (1) with two specimens, we could still monitor each specimen with strain gauges. As the number of specimens increased, we became wholly reliant on DIC. (2) By alternating between two modes, we were able to ensure relatively consistent strain on each specimen (although with DIC, we were still able to monitor the strains such that we at least knew what they were throughout each specimen's history, whether or not the amplitude remained fixed).

4.2 Strain Amplitude Fluctuations. It was observed that when a crack nucleates and propagates in one of the samples, the mode shape changes in a substantial enough manner to affect the strain in the other sample. Figure 13 shows the strain amplitudes in specimens A, B, and C over their fatigue lives. It is noted that between 300,000 and 360,000 cycles, the strain amplitude in specimen B increases by over 4%, corresponding to the rapid propagation of the crack in specimen A. Similar trends are observed when specimen B fails; the strain in specimen C increases temporarily until specimen B is replaced.

Short-duration overloads in the strain can have unpredictable effects on the fatigue life of the sample. In the simplest case, the overloads merely decrease the fatigue life according to Miner's rule [19]. However, there is also ample evidence that some alloys respond positively to short-duration overloads, resulting in a longer fatigue life [20]. Thus, strain overloads should be minimized to remove overload effects as a confounding variable from fatigue life measurements.

This reinforces the importance of sufficiently monitoring the strain on each specimen throughout testing. Because strain gauges have fatigue lives of their own, past research has relied on calibration curves that relate between the velocity as measured by a laser vibrometer (which is noncontacting and can remain active over the full duration of testing) and the strain as measured by a strain gauge (which fails prematurely) [11]. Without direct strain measurements, one might incorrectly conclude that the strain amplitude remains constant for fixed velocity. But because our test setup measures strain using digital image correlation (which is also noncontacting and can monitor strain for the full duration), we can directly monitor these changes in strain and account for them when determining fatigue properties [17].

To account for this discrepancy, we recommend splitting the data into two datasets: those by specimens that failed in their first loading step and those by specimens that survived multiple load steps. If both S–N curves are within the confidence bands of the other

one, the two populations of specimens can be used in combination, but with perhaps higher uncertainty.

5 Conclusion

In summary, a two-insert carrier approach was investigated for accelerating resonance-dwell fatigue measurements. In total, eight samples were fatigued using the two-insert carrier plate. From these tests, the carrier was found to be highly susceptible to fatigue itself and often failed before the samples. The tendency of the two-insert carrier to fail at the clamped edge and near the specimen attachment point highlights the importance of the stress ratio criterion established in prior work [21] for solid rectangular plates. Additionally, the tendency of specimen strains to change as cracks propagate suggests that future carrier plate designs should have resonant mode shapes that are tolerant to damaged specimens.

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Conflict of Interest

There are no conflicts of interest.

Data Availability Statement

The datasets generated and supporting the findings of this article are obtainable from the corresponding author upon reasonable request.

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