Improving Modal Parameter Predictions for Jointed Airframe Panels. Part I: Experiments.

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This paper describes a combined experimental and numerical investigation aimed at identifying the physical mechanisms and modeling errors that are the culprits for the lack of correlation between experimental measurements and analytical predictions of the modal parameters of helicopter airframes. The identification of the major contributors to this lack of correlation is an indispensable first step to actual modification and improvement of modeling processes and tools. The study focuses on simple configurations: panels with lap and butt joints fastened with HI-LITE fasteners or rivets. Impact hammer tests were used to measure their natural frequencies, mode shapes, and damping ratios. The results were compared with numerical predictions based on current industry modeling practices. Next, shaker tests were used to investigate the nonlinear behavior of the fastened connections. The most significant discrepancy between measurements and predictions were due to modeling idealizations of the joints. Nonlinearities were also important, though to a lesser extent.

Introduction

Reliable prediction of the structural dynamic behavior of airframes to avoid resonance has been a long term problem for the helicopter industry. Although efforts such as the NASA DAMVIBS program (Ref. 1) have been reported in the literature and have improved industry modeling practices, improvements have been limited to lower frequencies for a variety of reasons (Refs. 2,3).

In contrast to fixed wing aircraft, rotorcraft tend to have more secondary structure or non-structural items attached to the fuselage, which have been found to significantly alter the dynamic response. Recently, this tendency has been further aggravated by the use of a central frame in place of the conventional semi-monocoque construction. This increases the number of secondary structural panels in the exterior enclosure, modifying again the dynamic behavior. Another distinction which complicates the structural dynamic behavior is the major influence of engine, transmission and rotor blade vibrations and aerodynamics on the fuselage structural response, requiring a better knowledge of high frequencies than is required for fixed wing aircrafts (Refs. 4,5). Whereas a fixed wing vehicle analysis may typically need no more than the first dozen eigenmodes, helicopter models must often be reliable for fifty or more modes to ensure non-resonance and a jet smooth ride (Ref. 6). As a result, predicting sufficiently reliable modal parameters is still difficult and expensive. The lack of reliable predictions available in a timely manner has adversely affected the structural design process, and dynamics related problems are still a source of many redesign tasks after the airframe enters service.

The development of static and dynamic airframe models for finite element analysis has traditionally been executed manually with the assistance of automatic mesh generators, preprocessors, and drawing data exchange software. However, these tools do not directly address idealization errors which are inevitable as design data and physical behavior of interest are reduced into sets of algebraic equations that can be solved by a computer. In part, idealization errors are aggravated by the desire to reduce total airframe analysis cost by reusing as much of the static model as possible for the dynamic analysis (Refs. 7,8). Although this can save approximately 4% of the total airframe design effort (Ref. 9), modeling errors are created. For example, connections between parts are conservatively modeled as rigid for ultimate, static strength analysis, but connection flexibility and nonlinear behavior under operational loadings can be critical for an accurate prediction of the dynamic behavior. Even though better software support for the dynamic analysis of helicopter airframe components can reduce the cost of producing models, the implementation of idealizations still implies numerous modeling errors.

Some progress has been made towards incorporating improved dynamic analysis models into the design process. For example, the DAMVIBS program has reviewed and developed state-of-the-art finite element model practices for integral loads and vibrations analysis of airframes. Two issues related to the modeling of reinforcing members such as stringers have been identified and shown to be important contributors to airframe dynamics. The first concerns the treatment of those members which are discontinuous across manufacturing splices. The effect of such joints is considered in compression only, and results in effective axial continuity of unconnected stringers. The second concerns stringer's shear area. Although airframes usually contain many stringers, the cross-sectional areas of the stringers are not considered as contributing to the total effective shear area of a cross-section because of the usual assumption that the skin carries all the shear loads. Analytical studies have shown that shear loads carried by the stringers may not be negligible, as previously assumed.

Other research has focused on the adjustment of finite element models

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to more closely match experimental results. For instance, the PAREDYM package was jointly developed by industry and universities (Ref. 1): given the results of a finite element analysis of the fuselage and the experimental data, this software indicates which regions of the finite element model should be modified, in terms of mass and/or stiffness characteristics, to produce better agreement of the finite element predictions with test data. The regions are not substructures in the finite element sense, but a conglomeration of elements so as to reduce the total number of parameters for which the sensitivity analysis is performed. After inspecting the sensitivity of the results to the parameters of the various regions of the model, the analyst decides which design variable should be changed. If the regions are too coarse to allow proper understanding or adjustment, they can be subdivided further.

It should be stressed that both of the above activities operate on a finite element model that is based on an idealization of the actual structure. Furthermore, more fundamental errors in idealizations such as insufficient modeling of connection details and exclusion of dynamically significant secondary structures can not be automatically detected or corrected (Refs. 10,11). As a result, the experience gained in adjusting the finite element model for one aircraft design will not necessarily be applicable to the next design. Additionally, even if the adjusted dynamic model is considered reliable, an improved model of the connection regions including relevant joint flexibility will be needed to generate more reliable predictions.

This paper will describe a combination of experimental and numerical investigations aimed at identifying the physical mechanisms and modeling errors that are the culprits for the lack of correlation between experimental measurements and analytical predictions for panels with fastened connections. The identification of the major contributors to this lack of correlation is an indispensable first step to actual modification and improvement of modeling processes and tools. The first objective of the paper is to investigate the effects of several factors such as dimensional reduction and simplification, idealization errors of the connection details, nonlinearities, or the presence of friction and damping. The second is to quantify the relative magnitudes of the errors associated with these various modeling practices and simplifications. Lap and butt joints, two typical connection details common to helicopter designs, will be used in this study.

Description of the test set-up

An important problem in helicopter fuselage concepts is the design of connections between similar structural members subjected to bending. Such joint may be a connection between the outer panel and reinforcing members of the inner side of the airframe structure, or the connection between bulkheads or keels and skins in semi-monocoque fuselages. Fig. 1 shows two typical connection configurations commonly used in fuselage designs: lap joints and butt joints. The fasteners should be designed to carry a load greater than the design loads in the thin panels because there is considerable uncertainty about fit characteristics. The flanges to be fastened to the thin panels should be wide enough to provide proper strength

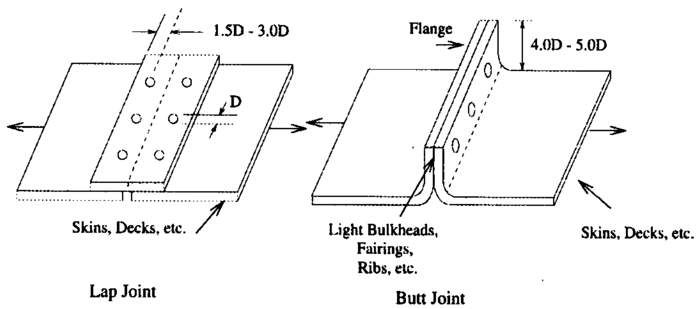

Fig. 1. The typical connection design configurations.

to the connection and adequate fastening space. A wide flange also prevents wrinkling of the thin sheets caused by concentrated loads at the fasteners (Refs. 12).

A lap joint typically transfers membrane loads whereas a butt joint carries transverse loads and/or stiffens a panel or membrane. Symmetric configurations are preferred over non-symmetric connection geometries that tend to warp under loading. Current helicopter airframes are made of thin sheet metal or composite shells, and must be designed so as to be primarily subjected to in-plane stresses. On the other hand, the stresses in the lap and butt joints will tend to cause relatively large bending deflections. HI-LITE fasteners and rivets are commonly used by rotorcraft airframe manufacturers for these various types of joints. This paper will focus on these two benchmark configurations and compare their experimentally determined dynamic behavior with finite element predictions.

Test samples

All test samples were made of aluminum (Al 5052) material with the following nominal properties: Young's modulus $E = 1.015 \times 10^7$ *psi*, Poisson's ratio $\nu = 0.33$, and density $\rho = 0.097$ *lbs/in*^{3}. The sheets had a nominal thickness $t = 0.09$ *in.*

Two types of samples were tested: simple rectangular panels, and panels with fastened connections. The first set of samples, used for calibration of the experimental set-up, are rectangular panels of length $L = 16$ *in* and width $W = 10$ *in,* clamped along the two short edges, free along the two others, and denoted *CFCF* for later reference. Similar panels with two lumped masses each of 0.236 *lbs* located at midspan of the free edges will also be investigated and are denoted *CFCF +M.*

The second set of samples consists of panels with lap and butt joints located at midspan. The panels are rectangular with a length $L = 16$ *in* and a width $W = 10$ *in,* clamped along the two short edges, free along the other two. Rivets or HI-LITE fasteners of diameter $D = 0.25$ *in* were used for the joints. In the lap joints, lap widths of 6 and 12 D were investigated, whereas for butt joints a height of 4 D was selected, see Fig. 1. In both cases, configurations with various numbers of fasteners were investigated. The various types of test samples will be identified using the following notation: *L/B xD yH/R,* where *L/B* stands for Lap or Butt joint, respectively; *x* indicates the size of the joint, *6D* or *12D* for the lap joints, *4D* or *5D* for the butt joints; and *y* indicates the number of fasteners, HI-LITE (*H*) or rivets (*R*). For instance, *L12D10H* indicates a lap joint with a lap width of 12 $D = 3$ *in,* fastened with 10 HI-LITE fasteners, *i.e.* 5 fasteners on each side of the joint; *B4D2R* indicates a butt joint of height 4 $D = 1$ *in,* fastened with 2 rivets. When manufacturing the specimens, the various parts of the sample were accurately positioned and clamped together, then fasteners or rivet holes were drilled, reamed and countersunk so as to minimize assembly mismatches.

Test fixture

The supporting fixture shown in Fig. 2 was made as stiff and massive as possible to avoid interference with the test results. The test samples are clamped with four steel bars (length 15 *in*, width 4 *in*, and thickness 2 *in*) by means of six 5/8 *in* bolts. A base steel plate (length 40 *in*, width 15 *in*, and thickness 1 *in*) supports the entire assembly. The complete test fixture weighs over 300 *lbs*, as compared with 1.4 *lbs* for the actual specimen.

Testing equipment

A single row or column of the transfer function matrix needs to be measured to identify the modal properties of a structure. In the Moving-Input approach, a row of the transfer matrix is measured. This can be achieved by locating a stationary accelerometer at one point on the struc-

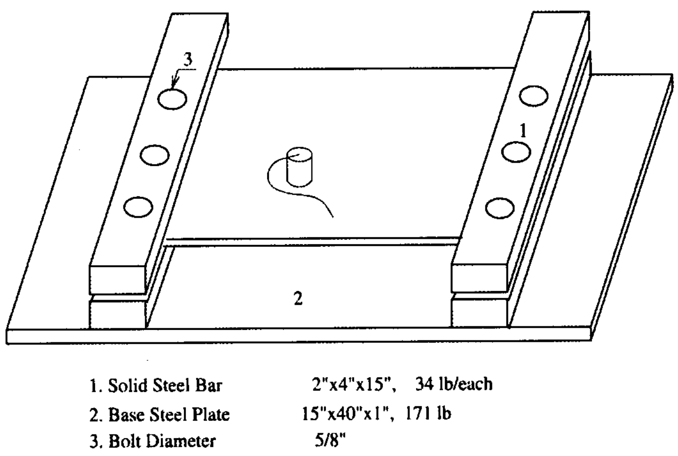

Fig. 2. The Clamped-Free-Clamped-Free (CFCF) boundary condition.

ture which is then excited at various points. This test requires a structure stiff enough to withstand the amount of impact force necessary to adequately excite the whole structure. The calibrated hammer test implements this methodology.

Figure 3 shows a schematic of the equipment used for the impact hammer test set-up. A calibrated impact hammer (PCB 291) with a built-in force transducer is used to excite the structure. The resulting signal is conditioned by a signal amplifier (PCB 480D06/480B02), and fed to a dynamic signal analyzer (HP 3562A). Signals from an accelerometer (PCB 336C34) attached at one location of the specimen with a thin wax film are conditioned and fed to the other input of the signal analyzer. The results of the signal analyzer are transferred to a data acquisition computer (HP9000, 330C) which processes the data. The SMS MODAL 3.0 software package was then used to identify the modal parameters.

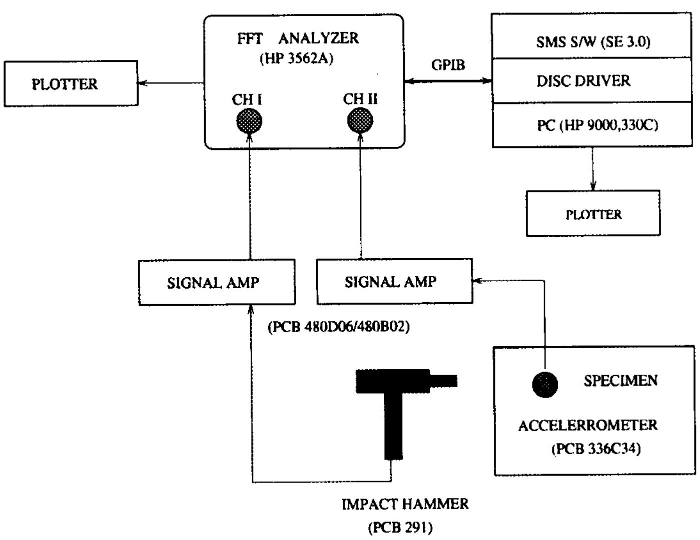

Fig. 3. The schematic diagram of impact hammer test setups.

In the Moving-Output approach, a column of the transfer matrix is measured. The structure is excited at a fixed location and the resulting response is measured at various locations on the structure. The shaker test implements this methodology. Figure 4 shows a schematic of the equipment used for the shaker test. A power amplifier (WR PA7C) provides the input signal for the shaker (WR F4 Z-820W) which also provides both force and acceleration transducers, effectively forming an impedance head. Both signals are fed into the signal analyzer.

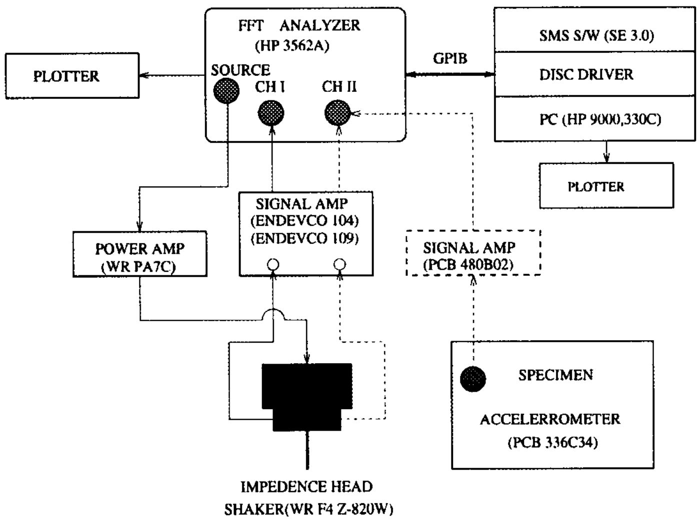

Fig. 4. The schematic diagram of shaker test setups.

The shaker was suspended down from a supporting fixture with three soft springs. The lowest natural frequency of the shaker/soft spring system was far lower than the lowest frequency of the test specimen. A threaded stinger rod was firmly attached to the test sample to transfer the shaker force. A swept sine wave excitation with one frequency component at a time was applied to the structure within the frequency range of interest.

Impact hammer test procedure

For both impact hammer and shaker tests, measurements were made at various locations on the test articles. Figure 5 shows the grid points that were selected in this effort so as to enable the accurate measurement of the five lowest modes of the structure, and to avoid spatial aliasing. In the impact hammer test, each point of the grid is successively excited by the hammer and the structural response is measured by an accelerometer placed at grid point 2.

The procedure used for data collection is summarized as follows:

1. Calibrate individual equipment components such as the force and acceleration transducers and signal amplifiers and then set up input data in the FFT analyzer and software that include a grid point geometry, window selection, frequency resolution parameters, and input and output ranges.
2. Measure the response of the test sample at the appropriate grid point location corresponding to impacts at all grid points.
3. Process the measured data to extract the modal parameters of the structure.

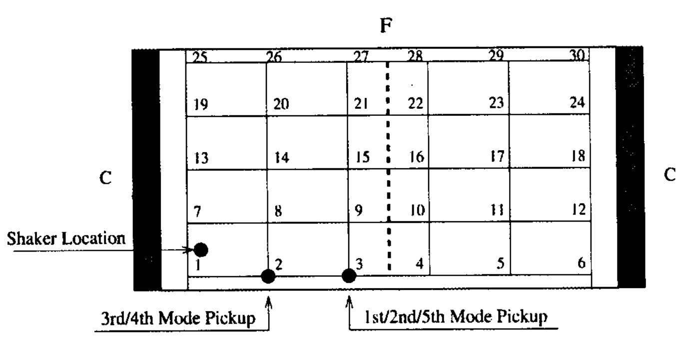

Fig. 5. The test points on the structure.

Shake test procedure

For the shake tests, the shaker location was chosen between grid point 1 and 7, see Fig. 5. This location was determined by trial and error, and found to provide the desired level of vibration amplitude for all the modes of interest. The response of the test article was then measured by an accelerometer successively located at all grid points. When focusing on the nonlinear behavior of the panels, an accelerometer was placed at grid point 3 for the determination of the first, second, and fifth modes, and at grid point 2 for that of the third and fourth modes. The location of the accelerometer was chosen close to the maximum deflection amplitude points for those specific modes. The procedure used for data collection is summarized as follows:

1. Same as step 1 for the impact hammer test.
2. At a given vibration amplitude level, extract the two first modes of the test sample using the swept sine method.
3. Increase the excitation force level and repeat step (2) for several vibration amplitudes until maximum vibration amplitude is reached.
4. Process the measured data to extract the modal parameters of the structure as a function of vibration amplitude.

Test results

The initial step for evaluating the dynamic behavior of lap and butt joints is to calibrate the test set-up by comparing the measured frequencies with the analytical frequencies for simple *CFCF* panels without joints. Next, impact hammer tests will be used to identify the modal parameters of panels with various joint configurations. Finally, the nonlinear behavior of these panels at large vibration amplitudes will be studied using shaker tests.

Calibration tests

The purpose of these initial tests is to calibrate the experimental set-up, verify the repeatability of the test results, and check the linear behavior of the structure, a basic assumption made in the modal parameter extraction algorithm. These calibration tests were performed on simple panel without joints, *i.e.* configurations *CFCF* and *CFCF +M*.

To check the consistency of the measured results, the tests were repeated for various levels of applied impact, various locations of the driving point and acceleration pick-up points, and various magnitudes of the clamping force on the supporting structure. The coefficient of variation of the experimental measurements was below 1%. For all tests conditions, a well defined frequency response function was found for the five lowest modes, implying the linear behavior of the structure.

Test results for the first five natural frequencies for both types of plates were compared with analytical predictions from the ABAQUS finite element package. Figure 6 shows that an excellent correlation was obtained between measured and predicted natural frequencies of the clamped-clamped panels with and without masses, *CFCF +M* and *CFCF*, respectively. In this figure, experimental measurements are plotted along the abscissa and predictions along the ordinate. Perfect correlation is indicated by a point on the 45° line bisecting the graph. Points falling above this line correspond to predictions that overestimate the measured frequencies; underestimations correspond to points below that line. The observed discrepancies are less than 3%, *i.e.* within the expected experimental error range.

The five lowest measured mode shapes were also compared with the analytical predictions. The Modal Assurance Criterion (MAC), a quantitative measure of the correlation between measured and predicted mode shapes, was computed and is shown in Table 1 where the row modes are extracted from numerical results using the ABAQUS finite element package results and the column modes from the modal testing data. The values on the diagonal in the MAC matrices are close to unity, implying a good correlation between measured and computed mode shapes.

Damping levels for the simple plate with and without masses remained below 1% critical for all tested configurations. Thus, the damped natural frequency measured experimentally and the resonant frequency predicted from the finite element analysis can be considered as the same.

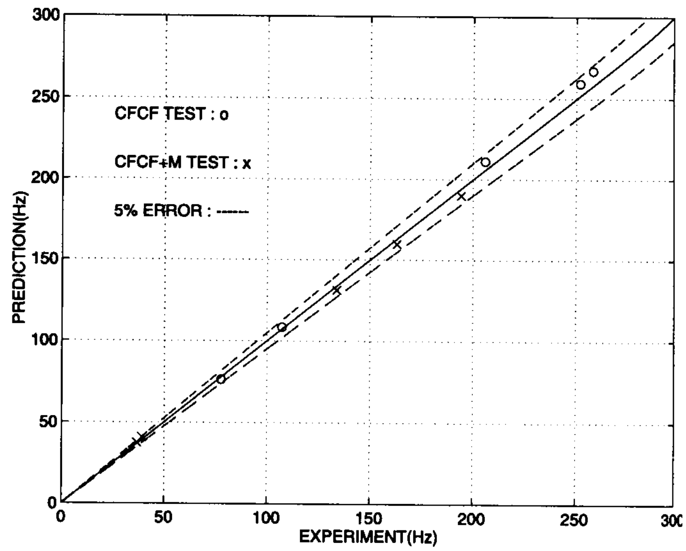

Fig. 6. Correlation between measured and predicted frequencies for the *CFCF* and *CFCF+M* panels.

Table 1. The MAC matrix for the *CFCF* plate.

MODE	1	2	3	4	5
1	.92	.02	.00	.00	.26
2	.00	.98	.00	.00	.00
3	.00	.00	.98	.00	.00
4	.00	.00	.00	.98	.00
5	.11	.00	.00	.00	.99

Impact hammer tests

The purpose of the impact hammer test is to determine basic modal parameters of the various fastened joint test articles. The experimentally measured results will be compared with numerical predictions obtained using representative industry modeling practices. It should be noted that no clear modeling guidelines are available for dealing with fastened joints. Two methods are widely used in industry for modeling joint areas, such as the connections between skin panels, bulkheads, fairings, decks, and doublers (Refs. 8,9). The first method, the *double-thickness* method denoted (DB-TH), merges the two panels forming the actual joint into a single idealized plate of double thickness. This approach ignores any stiffness reduction due to fastener holes, or the fact that two distinct panels are pressed together by the fasteners.

The second method, the *beam reinforcement* method denoted (BM-RF), replaces the joint by a reinforcing beam positioned at the location where the two disjointed panels come together. The sectional stiffness and inertial property of the beam are adjusted to represent those of the splice plate. The reinforcing beam and the two panels share a common line of nodes in the finite element representation. Here again, the stiffness reductions inherent to the lap or butt joint configuration are ignored. It is expected that both approaches should overestimate the frequencies of the fastened panels.

During preliminary testing it was found that very different modal parameters were obtained when the HI-LITE fasteners were *thumb tight* or *fully tightened.* In the thumb tight condition, the torque used to install the HI-LITE fastener is smaller than the break-off torque required to shear off the hex head portion of the HI-LITE collar on installation, whereas the fully tightened condition corresponds to the regular HI-LITE installation with the head portion of the fastener sheared off. For all hammer tests described in this section, the HI-LITE fastener were in the thumb tight condition.

Figures 7 and 8 show the correlation between experiments and predictions for panels with lap joints fastened with HI-LITEs, using the BM-RF and DB-TH methods, respectively. The BM-RF method systematically overestimates the measured frequencies by about 20%, whereas up to 40% discrepancies are observed with the DB-TH approach. The measured damping ratios were less than 2% for all test configurations.

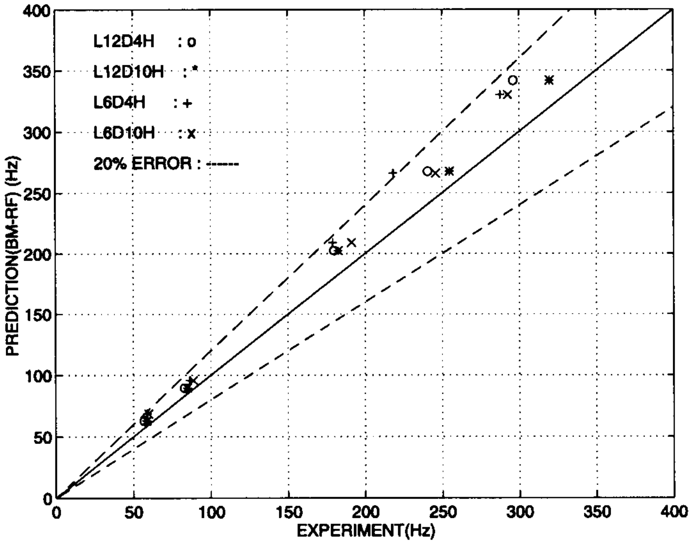

Fig. 7. Correlation between measured and predicted frequencies for lap joint panels fastened with HI-LITEs using the BM-RF method.

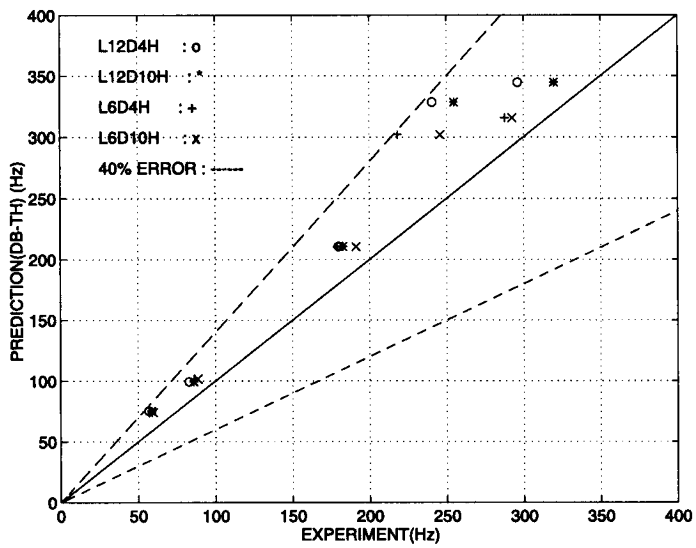

Fig. 8. Correlation between measured and predicted frequencies for lap joint panels fastened with HI-LITEs using the DB-TH method.

Figures 9 and 10 show similar correlations between experiments and predictions for panels with lap joints fastened with rivets, using the BM-RF and DB-TH methods, respectively; both methods overestimate the measured frequencies by about 15% and 30%, respectively. Because rivets are pressed to fit adjoining panels very tightly, they produces a stiffer joint than the corresponding configurations using HI-LITE fasteners. Accordingly, the natural frequencies predicted by the BM-RF and DB-TH methods that both assume an infinitely stiff joint are in better correlation with the measured frequencies than for the configurations involving HI-LITEs.

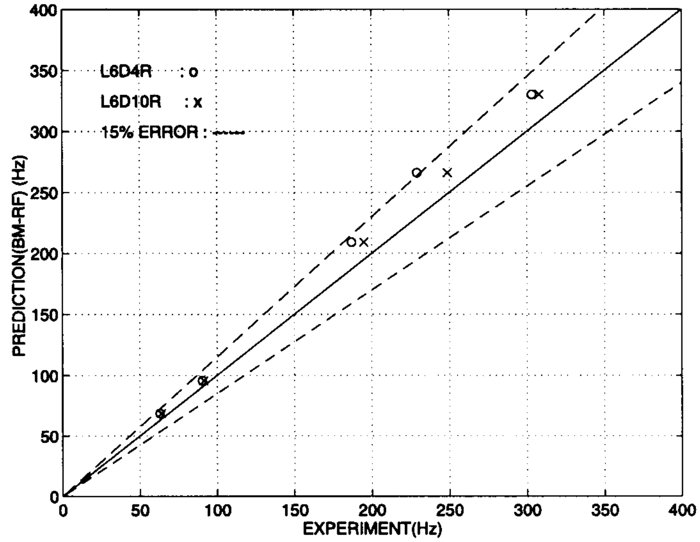

Fig. 9. Correlation between measured and predicted frequencies for lap joint panels fastened with rivets using the BM-RF method.

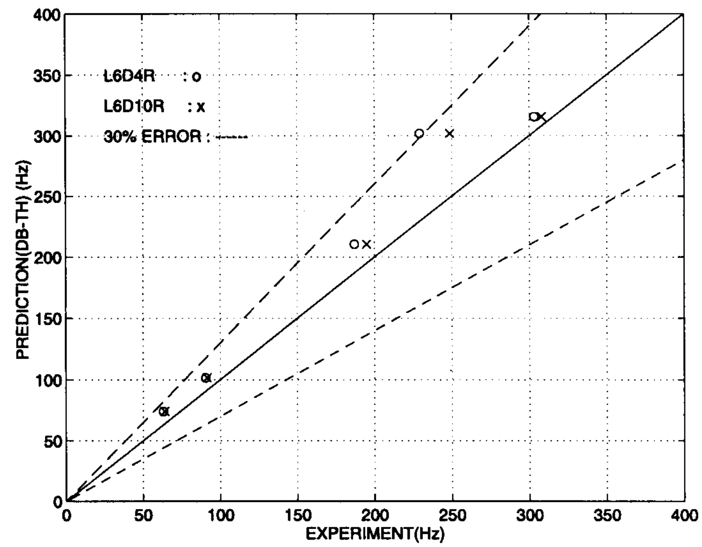

Fig. 10. Correlation between measured and predicted frequencies for lap joint panels fastened with rivets using the DB-TH method.

Finally, Figs. 11 and 12 present the correlation between experiments and predictions for the butt joint panel fastened with HI-LITEs and rivets, respectively. Since the BM-RF approach gives systematically better predictions than the DB-TH approach, only the BM-RF predictions are presented. Discrepancies of up to 20% and 15% are found for the HI-LITE and rivets joints, respectively.

A representative MAC matrix for the $L12D10H$ lap joint panel is given in Table 2. The diagonal entries of the MAC matrix are above 0.95, indicating a good correlation between measured and predicted mode shapes. Table 3 lists the corresponding data for lap joint panels fastened with rivets. Here again a good correlation is observed between experimental and predicted mode shapes except for the fifth mode shape which presents a 20% error.

In summary, the impact hammer tests show a good correlation between the experimentally measured mode shapes and the corresponding predictions based on current industry modeling practices, even though natural frequency correlations were rather poor. In all test configurations, the observed damping levels were less than 2%.

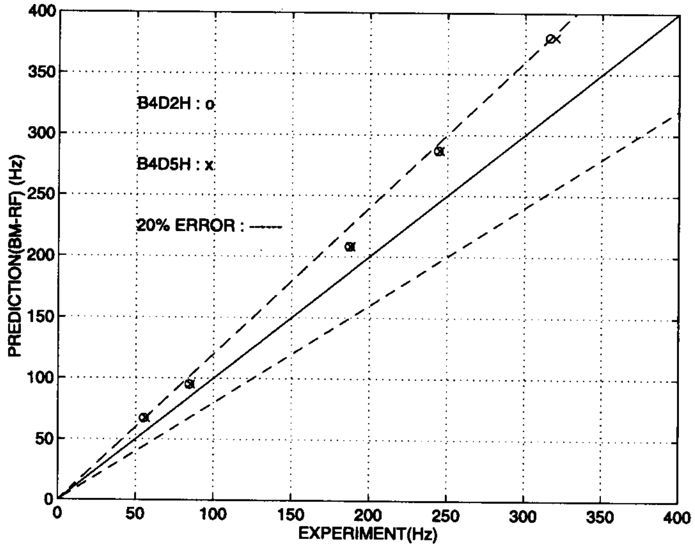

Fig. 11. Correlation between measured and predicted frequencies for butt joint panels fastened with HI-LITEs using the BM-RF method.

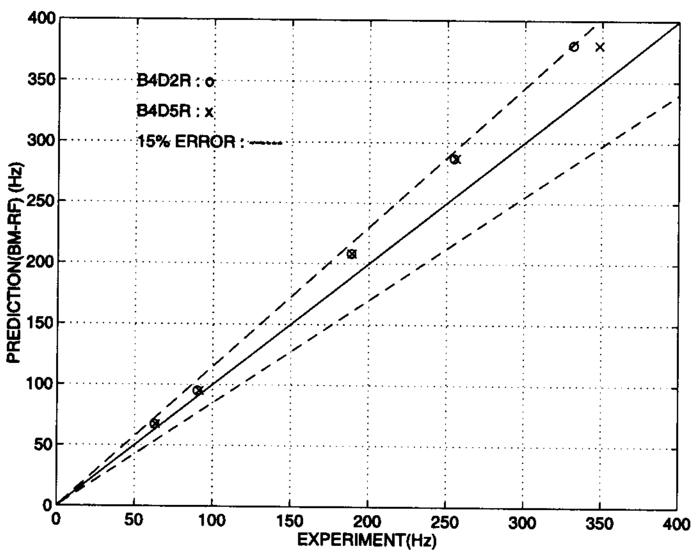

Fig. 12. Correlation between measured and predicted frequencies for butt joint panels fastened with rivets using the BM-RF method.

Table 2. The MAC matrix for the *L12D10H* panel modeled using the DB-TH method.

MODE	1	2	3	4	5
1	.99	.00	.04	.00	.03
2	.02	.98	.00	.00	.01
3	.00	.00	.98	.01	.00
4	.00	.00	.01	.98	.00
5	.05	.00	.00	.00	.98

Table 3. The MAC matrix for the *L6D4R* panel modeled using the BM-RF method.

MODE	1	2	3	4	5
1	.99	.00	.00	.00	.05
2	.00	.99	.00	.00	.00
3	.00	.00	.91	.03	.00
4	.00	.00	.03	.92	.00
5	.06	.05	.00	.01	.80

Shaker tests

In the impact hammer test, a very low level of excitation force is applied to the structure and the response closely approximates the linear response of the structure at infinitesimal amplitudes. However, if a larger excitation force is applied to the jointed structure, the frequency response function will be markedly different (Ref. 13). An important implication of this nonlinear behavior is that the modal parameters of the structure become amplitude dependent. The dynamic response of fastened connections is influenced by the nonlinear behavior associated with friction and/or relative motion between the two parts of the joint caused by clearance at the fastener or rivet holes, improper assembly or insufficient fastener clamping force, and one sided contacts between the components of the joint. In fact, significantly different responses should be expected for the thumb tight and fully tightened conditions. Although regular installation of the HI-LITE fastener corresponds to the fully tightened condition, it should be expected that the fastener clamping force will progressively loosen under in-flight service loadings, thermal effects, and aging effects.

The impact hammer test, because of its very high ratio of peak level to total energy, is not well suited for investigating the nonlinear behavior of structures. Since the total energy imparted to the test sample is distributed over a broad frequency range, the actual energy density is quite small. Excitation by a purely sinusoidal wave is more useful for nonlinear systems because of the precise control of the input spectrum level although it requires a long data acquisition time to attain the response at each frequency (Ref. 14). Furthermore, by concentrating the entire force spectrum at a single frequency, sinusoidal excitation can produce the highest possible response levels for a given force rating of the exciter (Ref. 15). As a result, individual modes can be studied at controlled amplitude levels. This method of testing is highly desirable in the nonlinear situations due to slippage or friction, rattling or loose components (Ref. 16). Accordingly, the nonlinear behavior of lap and butt jointed plates as a function of vibration amplitude will be studied using sine excitations within the frequency range of interest.

The effects of nonlinearities are investigated by measuring natural frequencies as a function of the amplitude of vibration using the swept sine test for the *CFCF* panel as well as panels with various types of joints. Figure 13 shows the frequencies of the first two modes for the *CFCF* and *L6D4H* panels as a function of vibration amplitude. The abscissa shows the measured vibration amplitude normalized by the plate thickness, whereas the ordinate shows the measured natural frequency normalized by the natural frequency obtained with the hammer test for the corresponding panel configuration.

For the *CFCF* panel the fundamental natural frequency rises by about 5% when the amplitude of vibration is about equal to the plate thickness. This behavior is to be expected: as vibration amplitude increases, transverse vibrations involve membrane stresses of increasing magnitude, thereby increasing the effective panel stiffness. This kinematic stiffening stems from the clamped boundary conditions. The frequency variation for the panel with joints is sharply different. At first a significant drop in frequency is observed. For the first mode, 12% and 7% drops are observed for the thumb tight and fully tightened conditions, respectively; whereas 4% and 3% drops are observed for the corresponding frequencies of the second mode. This initial drop in frequency can be explained by arguing that under increasing vibration amplitude, looseness in the joint allows relative motion between the various components of the joint, effectively making the joint increasingly compliant and lowering the natural frequencies. Past a certain vibration amplitude, the measured frequencies start to rise. The fundamental frequency of both thumb tight and fully tightened configurations rise by about 4% from their lowest observed value, and a similar rise is observed for the second mode. This rise can be explained by the kinematic nonlinearity associated with the clamped condition: once the fastener's motion amplitude reaches the maximum value allowed by the hole clearance, membrane stresses will appear resulting in rising effective panel stiffness.

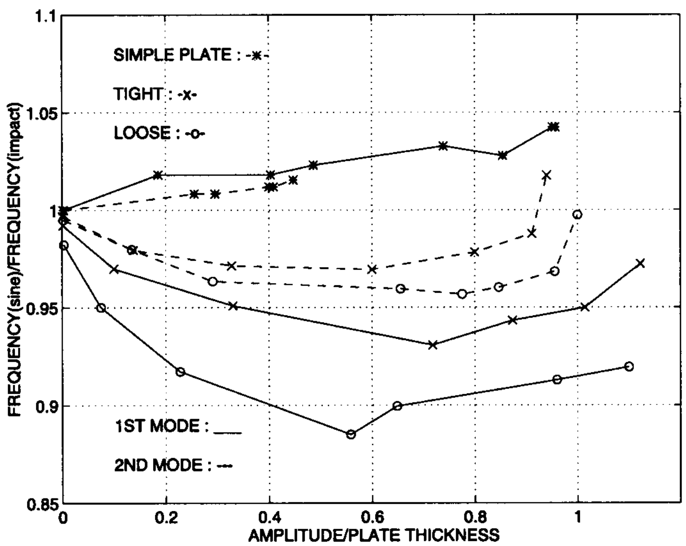

Fig. 13. Normalized first and second natural frequencies of the *L6D4H* panel versus normalized vibration amplitude.

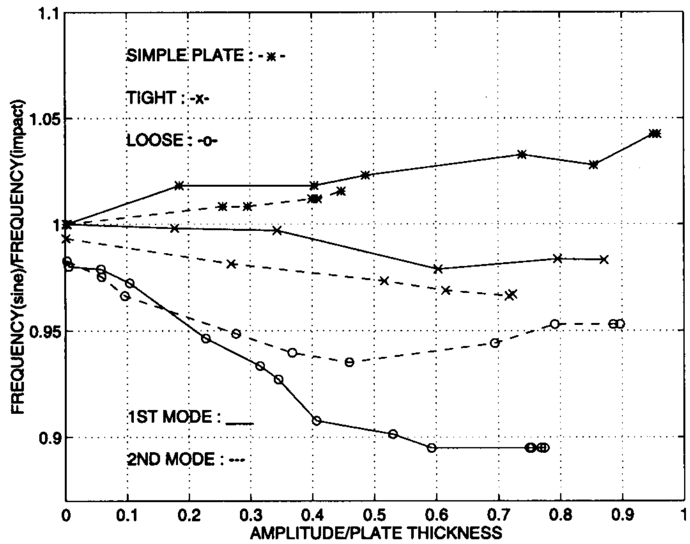

Fig. 14. Normalized first and second natural frequencies of the *L12D4H* panel versus normalized vibration amplitude.

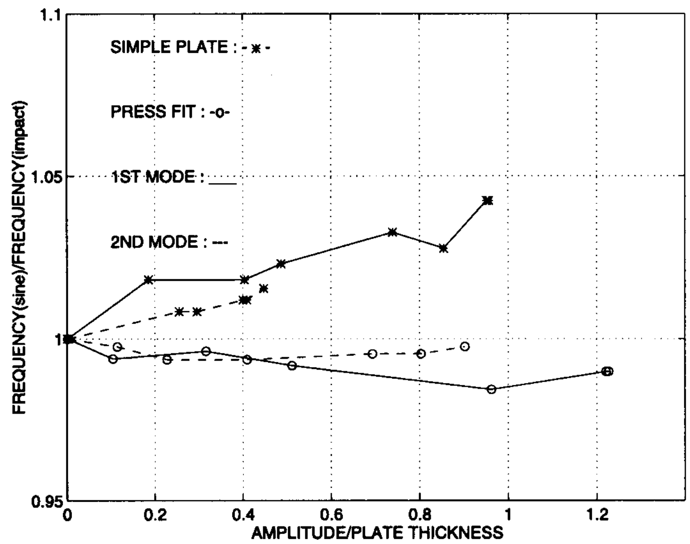

Fig. 15. Normalized first and second natural frequencies of the *L6D4R* panel versus normalized vibration amplitude.

Similar phenomena can be observed for the *L12D4H* panel which frequency response is shown in Fig. 14 as a function of amplitude. Figure 15 shows the corresponding results for the lap joint panel fastened with rivets. Similar trends are observed, though both the drop and subsequent rise in frequency are of a much smaller magnitude than was observed for the panels fastened with HI-LITEs. This is probably due to the fact that the press-fit process used to assemble the riveted joints produces much smaller hole clearances, resulting in stiffer joints.

Finally, the nonlinear behavior of butt joints is shown in Figs.16 and 17 for joints fastened with HI-LITEs and rivets, respectively. Much smaller variations in frequency are observed for these configurations. This is probably due to the smaller amount of relative motion of the components of the joint inherent to the butt joint configuration and the fact that greater joint axial flexibility delays the appearance of membrane stresses.

Conclusions

This paper has described an experimental program focusing on the dynamic behavior of panels with lap and butt joints fastened with HI-LITEs and rivets. The measured natural frequencies and mode shapes were compared with numerical predictions using current industry modeling practices.

At first, the modal characteristics of various panel configurations were measured at very small vibration amplitudes using the impact hammer test methodology. Next, the modal characteristics of the same panel configurations were investigated as a function of vibration amplitude. The following conclusions can be drawn form this study:

1. Comparison of measured and predicted frequencies showed very significant discrepancies, up to 20 and 40% errors when using the reinforcing beam and double thickness methods, respectively.
2. Measured and predicted mode shapes were found to correlate closely.
3. Nonlinear effects were shown to be significant. Up to 12% drops in frequency were observed for increasing vibration amplitudes, followed by a modest subsequent increase in frequency of up to 4% for vibration amplitudes equal to the plate thickness.
4. Very low damping levels, below 2% critical, were observed in all test cases.
5. The large discrepancies between measured and predicted natural can be primarily ascribed to the errors associated with current industry modeling practices.
6. Though significant, nonlinear effects are not as large a contributor to the observed discrepancy.
7. The low critical damping ratio observed in all tests indicates that dissi-

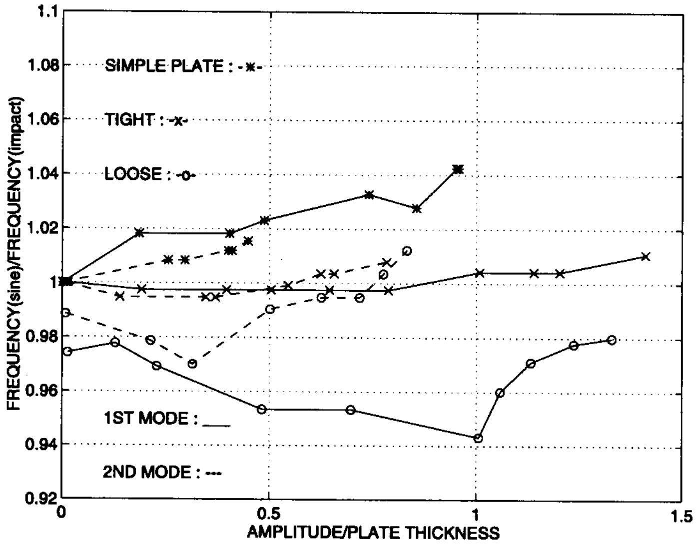

Fig. 16. Normalized first and second natural frequencies of the *B4D2H* panel versus normalized vibration amplitude.

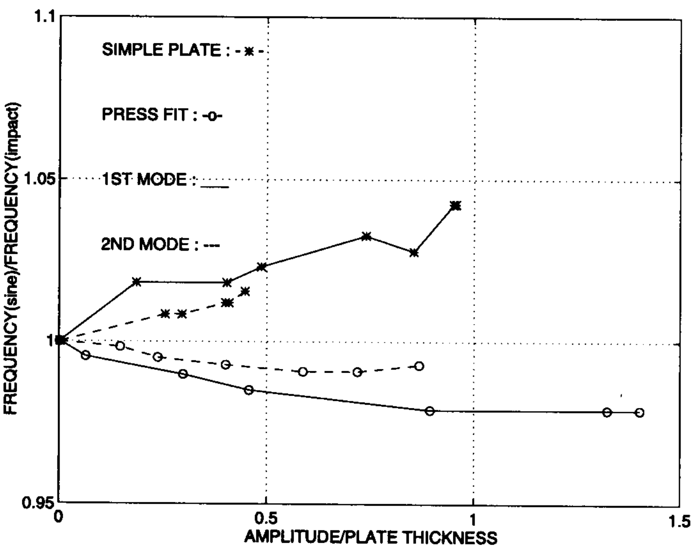

Fig. 17. Normalized first and second natural frequencies of the *B4D2R* panel versus normalized vibration amplitude.

pation mechanisms are not significant contributors to the observed discrepancies in natural frequencies.

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