Journal of Sound and Vibration (2001) **248**(1), 151–165
doi:10.1006/jsvi.2001.3758, available online at http://www.idealibrary.com on IDEAL®

MODAL TEST EXPERIENCES WITH A JET ENGINE FAN MODEL

J. J. HOLLKAMP AND R. W. GORDON

Air Force Research Laboratory, Wright-Patterson AFB, OH 45433-7006, U.S.A.

(*Received* 4 *August* 2000, *and in final form* 20 *April* 2001)

High cycle fatigue in jet engine blades is caused by excessive vibration. Understanding the dynamic response of the bladed disk system is important in determining vibration levels. Modal testing is a useful tool in understanding the dynamic behavior of structures. However, modal tests are not conducted on bladed disks because of the difficulties involved. One problem is that the overall dynamic behavior is sensitive to small perturbations. Another problem is that multiple inputs and high-resolution techniques are required to separate modes that are nearly repeated. Two studies of engine blade response were recently completed in which bench modal tests were successfully performed on simplified fan models. The modal test procedures for the first study were successful in extracting the modal parameters. But the tests in the second study were more demanding. Ultimately, an approach was devised that accurately extracted the modal parameters. This paper describes the challenges and the evolution of the test procedures.

1. INTRODUCTION

High cycle fatigue (HCF) of blades is a concern in the design of high-performance military jet engines. The concern has stimulated research in many areas including better material characterization, flow control, damping, and force response prediction. Recently, two research projects were completed that investigated integrally bladed disks with non-uniform blade damping [1] and the use of piezoelectric actuators as coupling devices [2]. In both projects, the work involved instrumented modal tests of jet engine fan models.

Formal modal tests on bladed disks have not been conducted in the past. A few investigators, e.g., reference [3], have characterized the forced response patterns of blade-disk assemblies. But these patterns can only be good approximations of mode shapes under the right conditions. One reason for the lack of true modal tests is that periodic structures are notoriously sensitive to mistuning. Small perturbations can drastically alter the character of the response, rendering the modal test results useless. Periodic structures also have multiple closely spaced frequencies, requiring multiple input testing and intricate modal identification algorithms.

The focus of this paper is to highlight the challenges encountered in the modal testing of a jet engine fan model used to investigate the effects of piezoelectric coupling on dynamic response [2]. Modal tests were successfully performed on a similar fan model in a previous study of non-uniform blade damping [1]. However, the same techniques were inadequate in the piezoelectric coupling study. The test article in the latter study was tuned, so that the uncoupled natural frequencies of the eight blades ranged from 783·4 to 783·9 Hz for a higher order modal group. Those very closely spaced frequencies resulted in an extremely difficult task in estimating the mode shapes. This paper will provide details, procedures and selected results from the piezoelectric coupling study. The paper will also show how the test procedures evolved from those used in the non-uniform damping study in order to produce accurate estimates.

1.1 BACKGROUND

Ideally, a bladed disk is a periodic or repeated system with the substructures, the blades, having identical natural frequencies. In reality, there will be slight variations in the blades. These variations mistune the individual blade natural frequencies and affect the system as a whole. Many researchers have studied the sensitivity of these periodic structures to mistuning [4–7]. In summary, they have found that the dynamic behavior of the bladed disk depends on the degree of mistuning and coupling that exists among the substructures.

The overall structure will have mode shapes that occur in families. For example, in a system with N blades, N global mode shapes will exist where each blade vibrates in what appears locally as the blade's first bending mode. The amplitude and phasing of each blade varies for each mode shape in the family. There are also modal families for the second bending mode, the first torsion mode, and so on. The modal frequencies within a family are closely spaced.

The modal behavior within a family depends on the relative amounts of mistuning and coupling among the blades. Mistuning is the variation in the uncoupled, individual, blade natural frequencies. The mistuning varies for different modal families. Coupling between the blades is due to disk flexibility and aerodynamic effects. Many researchers [2, 4–7] have modelled coupling as linear springs between adjacent blades. Each blade is represented as a lumped, spring–mass system. The individual blade stiffnesses may vary to represent mistuning, but usually all the coupling springs have the same stiffness value. This value may vary to change the overall coupling. Although this coupling model is unable to represent all the possible interactions between the blades, the essence of the coupling phenomenon is captured. For simplicity, this concept of coupling will be adopted here.

Mistuning and coupling have opposite effects on the mode shapes within a modal family. At one extreme, small amounts of mistuning or strong coupling will produce mode shapes that appear as regular patterns and involve all the blades. These mode shapes are global and are termed "extended". The forced response amplitudes are predictable and the vibration energy will be spread among all the blades. At the other extreme, large amounts of mistuning or very weak coupling produce mode shapes that have response in only a single blade. In this case, the blades are said to be completely uncoupled. Again, the forced response amplitudes are predictable. The vibration energy will be spread among all the blades, as each blade will be independently excited.

Between the two extremes, the mode shapes tend to become localized. The response becomes irregular with amplitude concentrated in a few blades. Individual blades within these groups may experience peak stresses that are significantly higher than in a perfectly tuned or completely uncoupled system. This magnification phenomenon is random. In order to prevent failure, the phenomenon requires that all blades be designed to withstand the highest stresses that only a few blades, in a few engines, may actually experience.

The response characteristics of bladed disks present unique challenges to the modal test engineer. The tight grouping of modal frequencies necessitates multiple input tests. The sensitivity to small changes in the tuning requires non-contacting excitation. Furthermore, we are most interested in identifying the mode shapes, yet these are the parameters most difficult to estimate particularly in the case of repeated or nearly repeated frequencies.

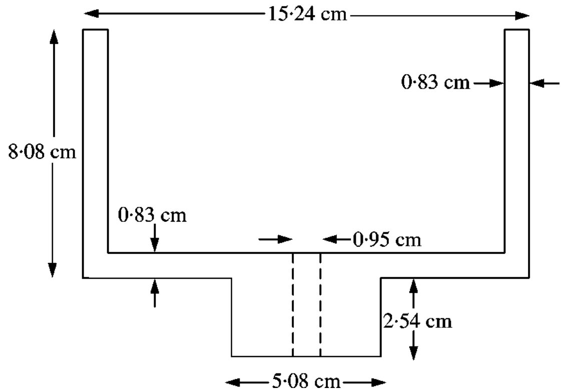

Figure 1. The hub cross-section for the model fan used in the piezoelectric coupling investigation.

2. THE FAN MODELS

Current trends in the design of modern jet engine fans call for integrally bladed disks with low blade aspect ratios. Two fan models were designed and fabricated for study incorporating these trends. The fan models were used in the two experimental investigations referenced previously [1, 2]. The models were designed to be relatively simple and inexpensive to analyze, fabricate and test. A design with eight blades was chosen. To keep the models to a manageable size, it was decided to scale down the dimensions while capturing, generically, the dynamic characteristics of real fans. The overall diameter was chosen to be 45·7 cm—approximately half the diameter of a modern fighter engine first stage fan. The hub was cylindrical with an outside diameter of 15·24 cm. The blades were flat plates 15·24 cm long by 11·43 cm wide by 0·16 cm thick. The blades were attached at a 45° angle to the fan's axis of rotation. Low alloy steel was selected as the model fan material. Although first stage fans of modern engines are fabricated from titanium alloy, steel blades will result in similar frequencies for a given geometry. The blades were attached to the hub by soldering them into shallow slots. A spacer on the rear of the hub allowed the fan to be securely bolted to a bracket or table while allowing the blades to vibrate without mechanical interference. The difference between the two models was hub wall and web thickness. The model in the non-uniform damping investigations had a wall and web thickness of 1·27 cm. The model used in the coupling investigation had a wall and web thickness of 0·83 cm in. A cross-section of this hub is shown in Figure 1. A photograph of this model is shown in Figure 2.

Modal testing in the previous studies [1, 2] was limited to selected modal families. The piezoelectric coupling study focused solely on the first chordwise bending or "two-stripe" modal family. The two-stripe family is a higher order modal family that has been identified as a candidate for passive damping application to reduce HCF [8]. The two-stripe shape on a single blade is shown in Figure 3. The mode is termed the two-stripe mode because it has two longitudinal node lines or nodal "stripes".

The model fan used in the piezoelectric coupling study had piezoelectric strain actuators (3·81 cm × 3·81 cm × 0·025 cm) bonded to the front and back of each blade. The actuators could be interconnected to mimic the simple lumped model popular in the literature [2, 4–7]. A diagram of the electrical connections is given in Figure 4. The size of the piezoelectric actuators is exaggerated in the figure. The negative electrode of each actuator is grounded to the fan. The positive electrodes are connected as shown in the figure. This

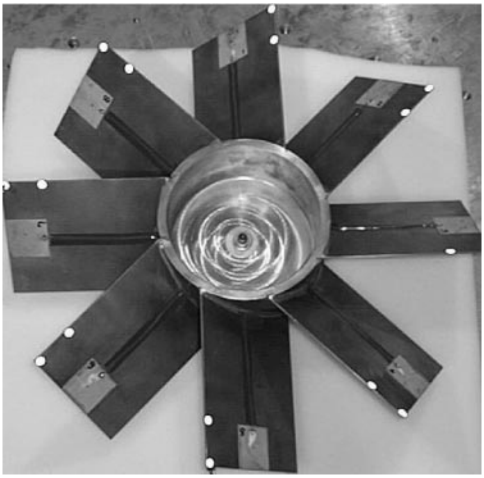

Figure 2. The model fan used in the piezoelectric coupling investigation.

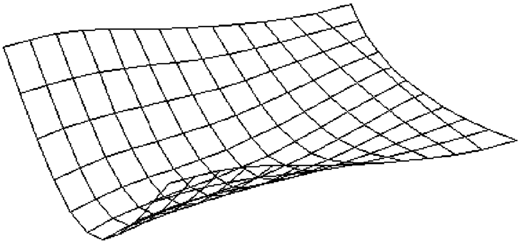

Figure 3. A wireframe of the local two-stripe mode shape (the blade root is the right-hand curved edge).

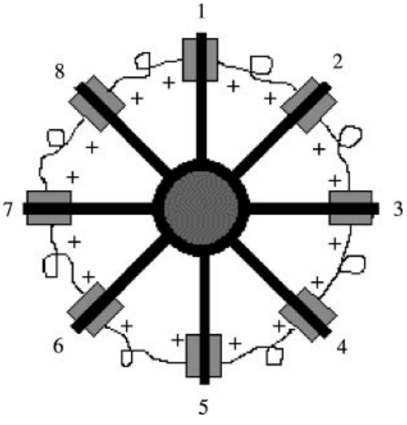

Figure 4. A diagram of the electrical connections between piezoelectric actuators.

electrical configuration makes the actuators act as springs between adjacent blades, increasing blade-to-blade coupling. As we will see later, the additional coupling dramatically alters the frequency spacing and mode shapes of the system and thus impacts the modal identification task. More details on this configuration can be found in reference [2].

The model fans were inherently mistuned due to small variations in the blades. The natural frequencies of the individual blades (with all sensors, actuators, and wiring in place) were measured to quantify the mistuning. The frequency of each blade was measured, one at a time, by temporarily adding small tip masses (approximately 20 g each) to the other blades to "detune" them and localize the mode shape to the blade of interest. In the piezoelectric coupling study, one of the models was tested in a nearly "tuned" configuration. The tuned configuration was achieved by altering the blade natural frequencies through the addition of small permanent masses to each blade. The resulting spread in the individual, uncoupled, blade frequencies was reduced from 10 Hz to a mere 0·5 Hz. The spread was only 0·06% of the new average frequency of 783·7 Hz. This configuration will be referred to, hereafter, as the "tuned" fan.

3. THE MODAL TEST SETUP

The modal test of the tuned fan in the piezoelectric coupling study was very difficult. The eight system modes in the two-stripe family occurred in a frequency band of only 1·1 Hz with the piezocoupling disabled. Modal test procedures used to characterize the model fan in the non-uniform damping study proved inadequate in the testing of the tuned fan. The procedures used in both tests are summarized in the following paragraphs. The inadequacies are pointed out and the new procedures used on the tuned fan are detailed. The emphasis is on the evolution of procedures as the modal test article's response behavior demanded new and more precise testing.

The response of a bladed disk is sensitive to perturbations and the modal excitation system must be chosen with this in mind. In the non-uniform damping study, small piezoelectric actuators (0·95 cm $\times$ 0·95 cm $\times$ 0·025 cm) were bonded to a root corner of each blade. A single amplifier was used to drive one actuator at a time. The electrical boundary condition on the actuator being driven by the amplifier was different from those of the other seven actuators. The difference in boundary conditions caused the frequency of each blade to drop when the amplifier was connected to its actuator. The frequency change was estimated to be slightly less than 0·1 Hz. This model had a relatively large mistuning, a 9·3 Hz spread in the natural frequencies for the two-stripe family. The frequency perturbations were considered tolerable in this case.

The tuned fan in the piezoelectric coupling study had blade frequencies with a spread of only 0·5 Hz. Frequency perturbations due to excitation were not tolerable. An acoustic excitation was used in this case. The input to the acoustic driver was a band-limited fast sine sweep (chirp). The signal swept from 770 to 805 Hz in 0·25 s, so that only the two-stripe family was excited. After the chirp, the signal amplitude was zeroed. Since the acoustic exciter did not physically contact the blade, it could be roved from blade to blade without perturbing the system. The identification procedure used in this study did not require identical excitation for each blade; thus, the exciter did not have to be precisely placed as it roved from blade to blade. The task of the identification procedure was also greatly reduced by band limiting the response to a single family.

A sequential sampling scheme was used to record the response data in the non-uniform damping study. An excitation actuator was driven using a repeating chirp signal. A single laser vibrometer was electronically steered to a point on each blade and that blade's response was recorded. The responses were not measured simultaneously. But since they were due to a periodic input, the response was periodic and the data records could be synchronized digitally after the test. That is, eight single-input–single-output measurements could be synchronized to build a single-input–multiple-output (SIMO) record for each excitation location. The procedure was repeated, providing excitation to each of the eight blades. This produced eight SIMO records. Small synchronization errors occurred in the sampled data and resulted in phase errors in the estimated mode shapes. The system was lightly damped, so in-phase or normal modes were expected. However, the phase errors caused the estimated mode shapes to appear as complex modes.

In the piezoelectric coupling study, it was decided to instrument each blade, so that simultaneous response measurement was possible. Small piezoelectric actuators ($2{\cdot}54$ cm $\times$ $1{\cdot}91$ cm $\times$ $0{\cdot}025$ cm) were used as sensors. These were permanently bonded to each blade. Synchronization errors encountered in the previous test were eliminated. The acoustic exciter was roved to provide eight input cases, but a SIMO time record was recorded directly for each input case.

The model fan in the non-uniform damping study was securely fastened to a rigid fixture using a bolt through the center hole. Similarly, the model fan in the piezoelectric study was securely fastened to a massive vibration isolation table using a bolt through the center hole. The hub thickness on the piezoelectric coupling fan was different than that of the non-uniform damping fan. Unfortunately, this difference lowered the frequency of a quasi-rigid body hub torsion mode to 770 Hz for the piezoelectric coupling model fan. The frequency of this mode was in close proximity to the two-stripe family. Early tests were conducted ignorant of this mode's presence. Modal tests using eight input cases were unsuccessful in accurately identifying the modes of interest. Instead of the eight modes expected, nine modes were identified, but accuracy indicators judged the results to be questionable. A finite element model of the system showed that the additional "phantom" mode could be the hub torsion mode if the interface between the hub spacer and the table became less than perfectly stiff. An auxiliary test was conducted with accelerometers attached tangentially to the hub. The test confirmed the existence of the hub torsion mode. It was realized that the frequency of this mode would drop further with a more flexible interface. A soft rubber pad between the fan spacer and the table moved the frequency out of the range of interest. No more identification problems surfaced as a result of the hub torsion mode.

4. RESULTS

Four configurations of the model fan were tested in the piezoelectric coupling study. The cases included the original mistuned configuration with and without coupling and the tuned configuration with and without coupling. The tuned configuration test required the modification of the test procedures and only the two cases of this configuration will be discussed in this paper.

The Eigensystem Realization Algorithm (ERA) [9] was chosen as the identification procedure. This algorithm was chosen for many reasons. First, ERA is a time-domain method. Time-domain methods generally have better frequency resolution than frequency-domain methods. Second, ERA can resolve modes with repeated frequencies up to the number of input cases. Finally, ERA can use either impulse response or free decay data. Thus, the free decay response of the system after the chirp signal is zeroed can be used without a calibrated measurement of the excitation. ERA was also the method used in the non-uniform damping study [1].

Free decay responses of the tuned model for the two configurations, uncoupled and coupled, are shown in Figures 5 and 6 respectively. The response is due to the acoustic excitation of blade number 1 in both cases. The coupled response appears to be more interesting than the uncoupled case. The modulation of the coupled case shows beating of

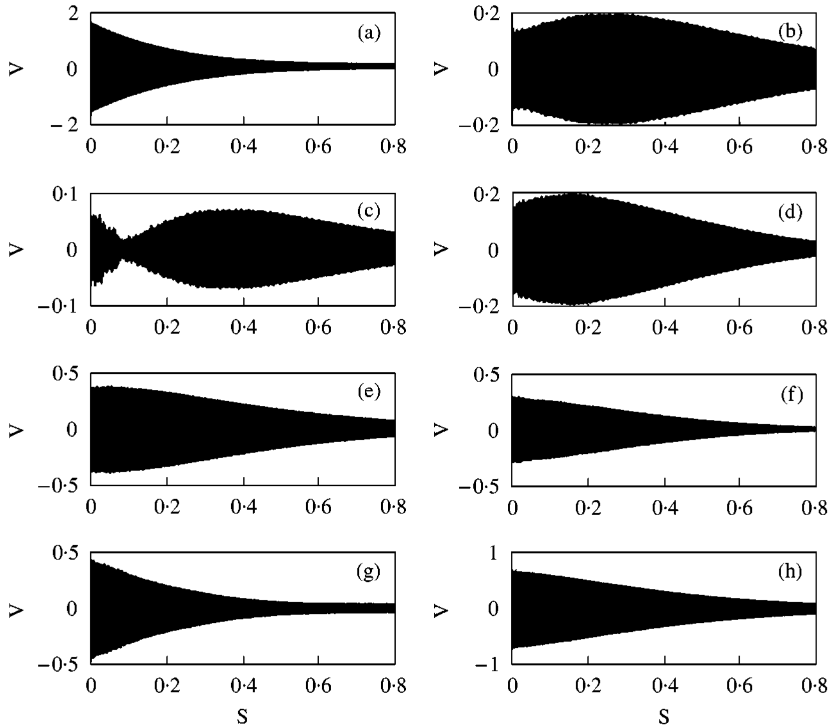

Figure 5. Free decay response due to excitation at blade 1 for the tuned, uncoupled fan: (a)–(h) show the response for blades 1–8 respectively.

the coupled, closely spaced modes very clearly. The uncoupled case also has some modulation but at a much lower frequency. The low-frequency modulation indicates a closer spacing of the modal frequencies and therefore a more challenging identification problem. Modal tests of the uncoupled configuration will be discussed first.

The first step in the identification using ERA is the formation of a data matrix using the eight input cases. The data matrix can be sized a number of ways. The number of columns determines the maximum number of modes that ERA can identify. The minimum number of columns is two times the number of expected modes. Here eight modes are expected, so the minimum number of required columns is 16. But more columns are necessary to remove identification bias caused by noise in the data, so 80 columns were used. The number of rows determines the amount of time response data used. The non-uniform damping study used 0·064 s of data [10]. A more difficult identification problem was expected here, so the length of data was doubled. This resulted in 4000 total rows of data. Once the data matrix is formed, it is factored into its singular value decomposition

$$H(0) = U\Sigma V^{\mathrm{T}}, \tag{1}$$

where

$$\Sigma = \mathrm{diag}(\sigma_1, \sigma_2, \ldots, \sigma_N) \tag{2}$$

and where $H(0)$ is the data matrix. The decomposition of the data matrix includes two orthonormal matrices, U and V, and a set of singular values, σ_i. The singular values are very helpful in assessing the quality of the data, determining the number of modes present,

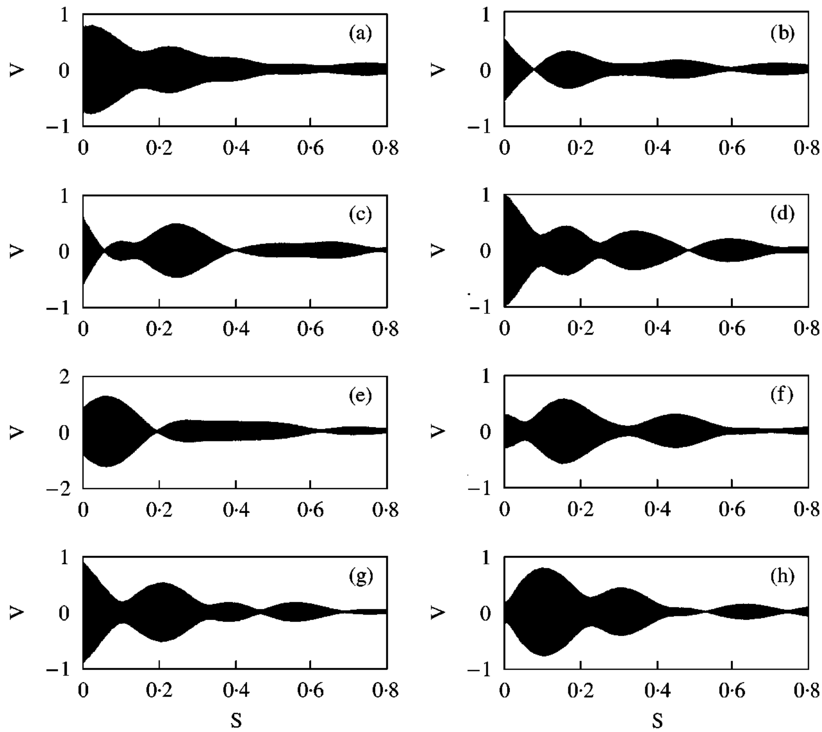

Figure 6. Free decay response due to excitation at blade 1 for the tuned, coupled fan: (a)–(h) show the response for blades 1–8 respectively.

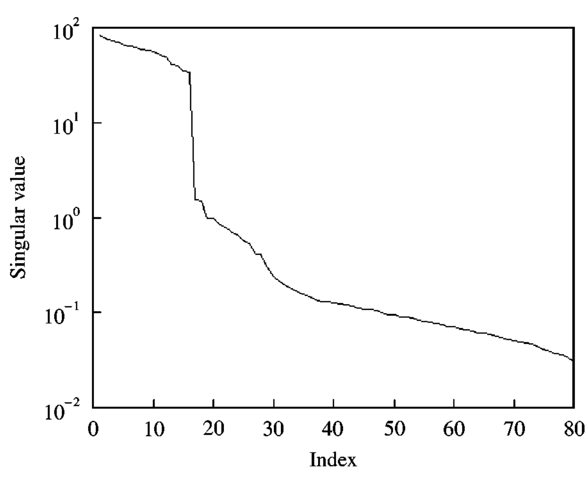

Figure 7. Singular values of the data matrix formed by ERA for the tuned, uncoupled case.

determining the level of noise and eliminating some of the effects of noise. The singular values for the tuned, uncoupled case are plotted on a log scale in Figure 7. The plot indicates that there are principally 16 singular values, corresponding to the eight modes that were expected. The plot also indicates that the noise level is relatively low by the steep drop in the magnitude of the singular values after number 16.

Theoretically, the noise can be removed from the data by retaining only a signal subspace. The signal subspace is composed of the principal singular values and the

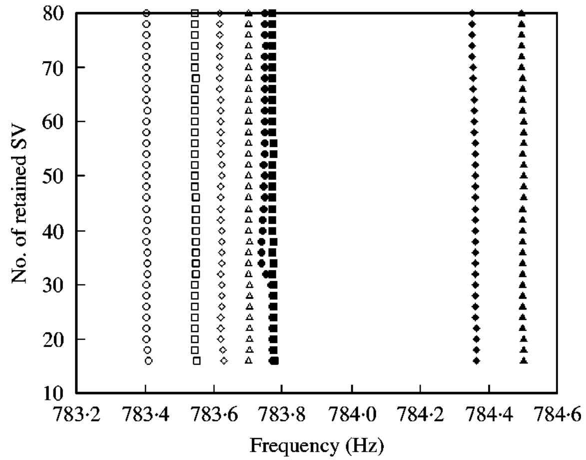

Figure 8. The modal frequencies of the two-stripe family as the number of retained singular values varies. Modes 1, 2, 3, 4, 5, 6, 7, and 8 are indicated by ○, □, ◇, △, ●, ■, ◆, and ▲ respectively.

corresponding columns of U and V. The remaining singular values and corresponding columns of U and V form the noise subspace which is truncated in the ERA solution. The modal parameters are then extracted from various formulations of the signal subspace combined with a time-shifted version of the original data matrix. Although the singular value decomposition can theoretically separate the noise from the data, in practice the so-called noise subspace retains some information. Typically, the number of retained singular values is varied in the solution to determine the proper singular value cutoff. Varying the cutoff produces multiple ERA solutions. Modal parameters and accuracy indicators from the solutions are viewed to judge the overall quality of the identification, and to help select a singular value cutoff. This establishes the "best" solution.

The number of retained singular values was varied from 16 to 80 in the identification of the tuned, uncoupled case. The identified frequencies for the two-stripe modes are shown in Figure 8 as a function of the number of retained singular values. Eight modes are identified once 16 singular values are retained and the spread of frequencies is only 1·1 Hz. More than 30 singular values need to be retained to separate the frequencies of modes 5 and 6. Notice that no frequencies appear other than the eight two-stripe modes; in fact, these are the only identified frequencies between 750 and 820 Hz. Basically this plot tells the user that all eight modes were excited, there are no other modes in this span, the effects of noise are small, and the modes can be resolved in a temporal sense. The extended modal amplitude coherence (EMAC) for these solutions was plotted. The EMAC is a measure of the temporal consistency of the identified results [11]. An EMAC of 100% is perfect correlation and usually anything above 95% is considered excellent. The EMAC for all eight modes in all the solutions exceeded 95%. The EMAC values confirm that the temporal nature of the modes has been captured by the identification results.

The modal phase collinearity (MPC) is typically used as a measure of the spatial consistency of the identification results [11]. The MPC measures the deviation from normal mode behavior for an individual modal shape vector. Normal modes will have components in phase or 180° out of phase from each other. Complex modes have multiphase components. The tuned fan is lightly damped, so the mode shapes are expected to be normal. Thus, the complexity of the modes can be used to discern the spatial quality of the identification results. The MPC is used to quantify the complexity. The MPC is a more

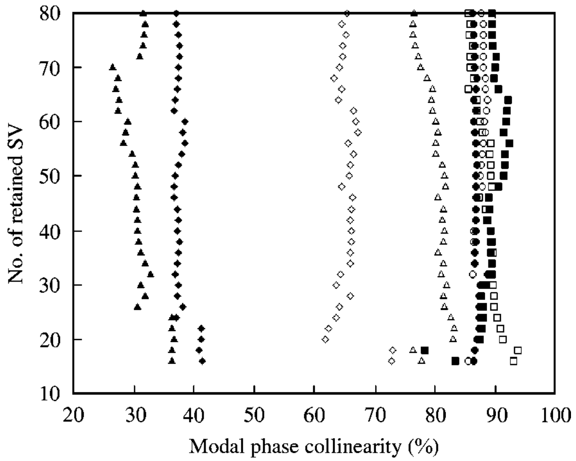

Figure 9. The modal phase collinearity of the two-stripe modes as the number of retained singular values varies. Modes 1, 2, 3, 4, 5, 6, 7, and 8 are indicated by ○, □, ◇, △, ●, ■, ◆, and ▲ respectively.

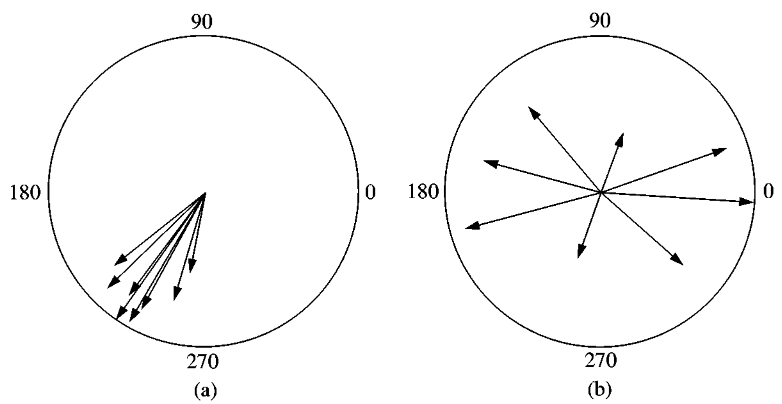

Figure 10. Identified mode shapes using ERA for the tuned, uncoupled case plotted on a phase diagram: (a) mode 5, (b) mode 8.

discriminating measure than the EMAC. A MPC of 100% indicates perfect spatial correlation for an assumed normal mode, but anything above 80% is considered acceptable.

The MPC of the identified mode shapes as a function of the number of retained singular values is shown in Figure 9. The MPC for modes 3, 7 and 8, are unacceptably low. The solution using 40 retained singular values was chosen as the overall solution. This solution will be referred to as the standard ERA solution. For demonstration purposes, the mode shape vectors for modes 5 and 8 for this solution are plotted on phase diagrams in Figure 10. Mode 5 appears to be approximately normal (monophase) while mode 8 is obviously too complex. This indicates an inaccurate identification of mode 8.

Approximate real mode shapes can be calculated based on the amplitude and relative phasing of the modal vectors. Note that the calculated real shapes for modes 3, 7, and 8 will not be good approximations. The approximate real modes are plotted in Figure 11. The blades are represented clockwise starting with blade 1 in the 12 o'clock position. The lengths of the radial lines symbolize the relative vibration amplitudes. Solid lines denote vibration in-phase while the dotted lines denote vibration 180° out of phase. Note that the

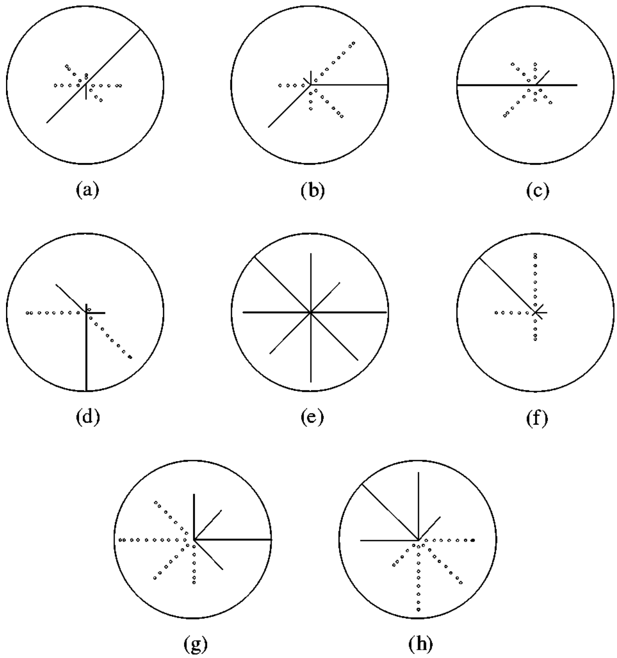

Figure 11. Approximate real mode shapes for tuned, uncoupled case from the standard ERA solution: (a)–(h) show modes 1–8 respectively.

Table 1

Identified modes

Mode	Uncoupled fan standard results: Frequency (Hz)	Uncoupled fan standard results: MPC (%)	Uncoupled fan composite results: Frequency (Hz)	Uncoupled fan composite results: MPC (%)	Coupled fan: Frequency (Hz)	Coupled fan: MPC (%)
1	783·4	87	783·4	87	776·9	99
2	783·5	89	783·5	89	778·4	100
3	783·6	66	783·7	97	778·7	99
4	783·7	82	783·7	82	780·2	99
5	783·7	87	783·7	87	780·5	100
6	783·8	89	783·9	95	782·6	100
7	784·4	37	784·4	98	782·9	98
8	784·5	31	784·5	93	783·7	100

approximations for modes 7 and 8 resemble what are often called single-nodal diameter modes. That is, both modes have a single diametral node line. The frequencies and MPC values are listed in Table 1.

The parameters on the ERA solution were varied in the hope of improving the phase accuracy of modes 3, 7, and 8. The variations included changing the number of rows and columns in the data matrix and reducing the sample rate by skipping data points. None of the variations were able to produce a better single overall solution. A different ERA formulation using data correlations was also tried. This formulation is known as ERA/DC [12]. In ERA/DC, the original free response time data is used to estimate cross- and

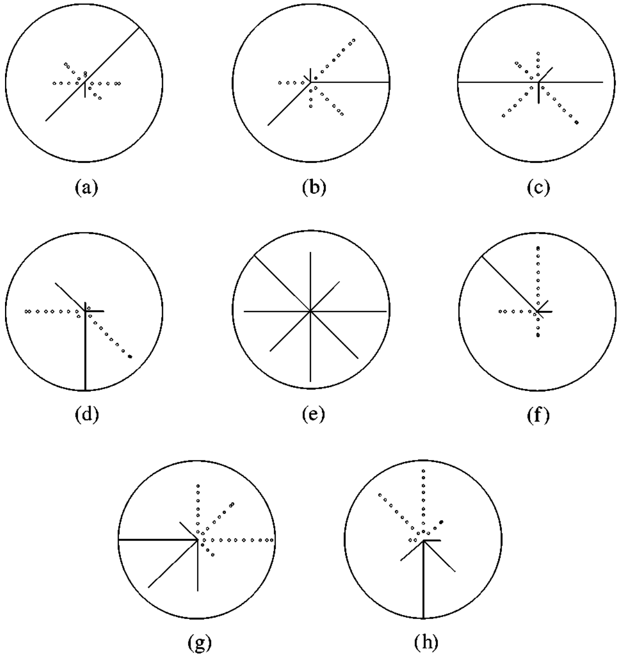

Figure 12. Approximate real mode shapes for tuned, uncoupled case from a composite of solutions: (a)–(h) show modes 1–8 respectively.

auto-correlations. These data correlations are then used by ERA instead of the original data. The use of data correlations has two major impacts on our problem. First, more of the original data can be used and condensed into the data correlations. Thus, the algorithm will, in effect, see more of the slow modulation of the original data. The other major impact is that noise bias can be reduced through the averaging process in the correlation calculation.

The ERA/DC algorithm was exercised with a variety of solution parameters. No single overall solution was obtained that was better than the standard ERA solution. But in some solutions, individual mode shapes improved. A "composite" solution was formed with some modes from the standard ERA solution and other modes from ERA/DC solutions. The approximate real mode shapes for this composite solution are shown in Figure 12. The frequencies and MPC values are listed in Table 1. Modes 1, 2, 4, and 5 are from the standard ERA solution. Modes 3 and 6 are from one ERA/DC solution and modes 7 and 8 from another solution. The only substantial difference between the improved results and the modes shown in Figure 11 is the improved MPC values for modes 3, 7, and 8. For comparison purposes, the improved mode shape vector for mode 8 is plotted on a phase diagram in Figure 13. The modal vector is now mostly in-phase or 180° out of phase producing an MPC of 93%. Other differences between the standard ERA solution and the composite include an increase in the estimated frequencies of modes 3 and 6, each by 0·1 Hz. Note that modes 7 and 8 are still single-nodal diameter modes, but the orientations of the nodal lines have changed. Repeated modal pairs like these have arbitrary orientations. When constructing a composite model, arbitrary modal pairs should come from a single identification solution to insure orthogonality.

In contrast to the tuned uncoupled case, the modal identification of the tuned model with coupling was relatively easy using standard ERA. The eight two-stripe modes are readily

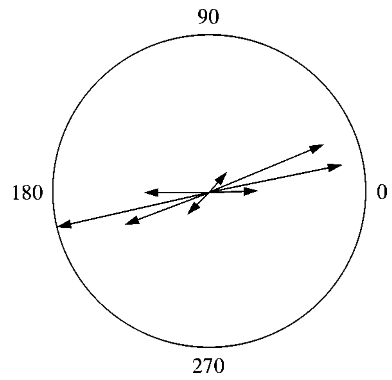

Figure 13. The identified shape for mode 8 for the tuned, uncoupled case from ERA/DC case plotted on a phase diagram.

resolved both temporally and spatially. The ERA data matrix again had 80 columns and 4000 rows. When the number of retained singular values is varied between 16 and 80, the eight two-stripe modes are identified with EMAC and MPC values exceeding 97% for all the solutions. The solution when 40 singular values are retained was chosen as the overall solution. The approximate real mode shapes for this solution are shown in Figure 14. Table 1 lists the frequencies and MPC values for this solution. The MPC values indicate near normal behavior. Note that the frequencies for the coupled case are lower than those for the uncoupled case. The piezoelectric actuators were electrically open in the uncoupled case. This is equivalent to a spring between each blade and ground. The actuators mimicked springs between adjacent blades in the coupled case, which is a less stiff configuration.

The primary reason for the improved identification results of the tuned coupled case compared to the tuned uncoupled case is the frequency separation of the modes. The additional coupling provided by the piezoelectric actuators causes a frequency separation of 6·8 Hz for the eight modes. The coupling also causes the occurrence of some interesting modal behavior. The mode shapes are now very regular and approach the mode shapes of a perfectly tuned system. But as discussed in reference [2], the piezoelectric actuators provided weak coupling at best. They were unable to produce regular mode shapes when the individual blades were mistuned.

5. SUMMARY

Modal tests on bladed disks are challenging. Bladed disks are repeated structures. Repeated structures have families of modes with closely spaced frequencies. The dynamic response becomes a function of the tuning of the substructures. Small changes drastically alter the character of the response. Modal tests conducted on these types of structures must use multiple inputs and minimize perturbations to the structure. Modal tests conducted on a fan model used to study non-uniform damping were successful. However, the same procedures were inadequate on a second model used to study augmentation of coupling using piezoelectric actuators. The individual blades on this fan were tuned to be much closer in frequency and the demands on the modal test procedure were even stricter.

The entire modal test procedure had to be changed to meet the demands. The small perturbations caused by a roving exciter had to be eliminated. Potential synchronization errors were also eliminated with a revamped data acquisition scheme. Even the trusted

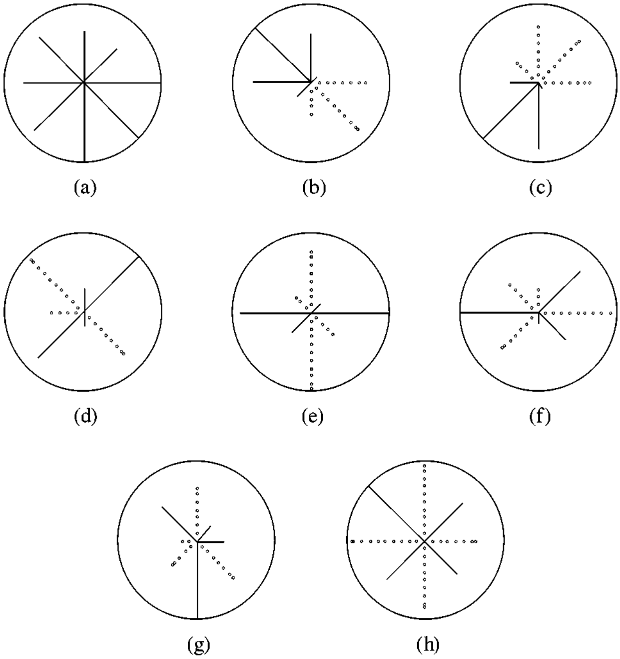

Figure 14. Approximate real mode shapes for tuned, coupled case: (a)–(h) show modes 1–8 respectively.

standard Eigensystem Realization Algorithm had to be altered to resolve the closely spaced modes. The modal test was difficult but in the end, the new procedures were successful. The same procedures where then applied to the same structure only now with additional coupling added by the piezoelectric devices. The additional coupling caused a spread in the modal frequencies and regular mode shape patterns to occur. The results in this case were relatively easy to extract with the new procedures.

The modal test techniques described in this paper were ultimately successful when applied to an idealized 8-bladed fan model on the bench. However, performing a modal test on a real jet engine fan will be much more difficult. Real fans have many more blades and more complex geometry. Ultimately, modal parameters are needed for a fan while it is rotating in air flow. The rotational acceleration changes the blade frequencies and makes instrumentation more difficult. The air flow adds damping and blade-to-blade coupling. These complicating factors make the prospect of identifying the modal parameters of a real fan a practically impossible task. Therefore, approximations must be made using bench tests and other dynamic measurements.

REFERENCES

1. R. W. GORDON and J. J. HOLLKAMP 1998 *The* 34*th AIAA/ASME/SAE/ASEE Joint Propulsion Conference and Exhibit*, *AIAA*-98-3747. An experimental investigation of non-uniform damping in bladed disk assemblies.
2. R. W. GORDON and J. J. HOLLKAMP 2000 *The* 41*st AIAA/ASME/ASCE/AHS/ACS Structures, Structural Dynamics and Materials Conference*, *AIAA*-2000-1700-*CP*. An experimental investigation of piezoelectric coupling in jet engine fan blades.
3. J. JUDGE, C. PIERRE and O. MEHMED 2000 *Proceedings of ASME TURBOEXPO* 2000, *ASME Paper* 2000-*GT*-358. Experimental investigation of mode localization and forced response amplitude magnification for a mistuned bladed disk.

4. S.-T. WEI and C. PIERRE 1988 *Journal of Vibration, Acoustics, Stress, and Reliability in Design* **110**, 429–438. Localization phenomena in mistuned assemblies with cyclic symmetry. Part I: free vibrations.
5. S.-T. WEI and C. PIERRE 1988 *Journal of Vibration, Acoustics, Stress, and Reliability in Design* **110**, 439–449. Localization phenomena in mistuned assemblies with cyclic symmetry. Part II: forced vibrations.
6. G. OTTARSON and C. PIERRE 1995 *The* 36*th AIAA/ASME/ASCE/AHS/ACS Structures, Structural Dynamics and Materials Conference*, *AIAA*-95-1494-*CP*. On the effects of interblade coupling on the statistics of maximum forced response amplitudes in mistuned bladed disks.
7. J. J. HOLLKAMP, C. L. HUSTEDDE and R. W. GORDON 1997 *The* 38*th AIAA/ASME/ASCE/AHS/ACS Structures, Structural Dynamics and Materials Conference*, *AIAA*-97-1395-*CP*. Effects of damping on a blade-disk assembly.
8. R. KIELB, et al. 1998 *The* 34*th AIAA/ASME/SAE/ASEE Joint Propulsion Conference and Exhibit*, *AIAA*-98-3863. Advanced damping systems for fan and compressor blisks.
9. J.-N. JUANG and R. S. PAPPA 1985 *Journal of Guidance, Control, and Dynamics* **8**, 620–627. An eigensystem realization algorithm for modal parameter identification and model reduction.
10. J. J. HOLLKAMP and R. W. GORDON 1999 *Proceedings of the* 17*th International Modal Analysis Conference*, 826–832. Modal testing of a bladed disk.
11. R. S. PAPPA, K. B. ELLIOT and A. SCHENK 1992 *AIAA Dynamics Specialists Conference*, *AIAA*-92-2136-*CP*. A consistent-mode indicator for the eigensystem realization algorithm.
12. J-N. JUANG, J. E. COOPER and J. R. WRIGHT 1988 *Control-Theory and Advanced Technology* **4**, 5–14. An eigensystem realization algorithm using data correlations (ERA/DC) for modal parameter identification.