NOTICE OF CHANGE

NOT MEASUREMENT SENSITIVE

MIL-HDBK-1783B
NOTICE 1
31 May 2002

DEPARTMENT OF DEFENSE
HANDBOOK

ENGINE STRUCTURAL INTEGRITY PROGRAM (ENSIP)

TO ALL HOLDERS OF MIL-HDBK-1783B:

1. THE FOLLOWING PAGES OF MIL-HDBK-1783B HAVE BEEN REVISED AND SUPERSEDE THE PAGES LISTED:

NEW PAGE	DATE	SUPERSEDED PAGE	DATE
141	15 February 2002	141	REPRINTED WITHOUT CHANGE
142	31 May 2002	142	15 February 2002
142a	31 May 2002	NEW PAGE	
142b	31 May 2002	NEW PAGE .	

2. RETAIN THIS NOTICE AND INSERT BEFORE TABLE OF CONTENTS.

3. Holders of MIL-HDBK-1783B will verify that page changes indicated above have been entered. This notice page will be retained as a check sheet. This issuance, together with appended pages, is a separate publication. Each notice is to be retained by stocking points until the handbook is completely revised or canceled.

Custodians:
Army – AV
Navy – AS
Air Force – 11

Preparing activity:
Air Force – 11
(Project 15GP-0010)

AMSC N/A

AREA 15GP

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different natural frequencies because of FOD, DOD, manufacturing variation, or other occurrences, then a single blade or a set of blades may act as a vibration absorber for the entire assembly. The result is exceptionally-high amplitudes for those blades, which leads to blades that accumulate fatigue damage faster than would normally be expected.

In higher groups, when there is mistuning, the concepts of groups eventually breaks down, as the mistuning causes enough disorder in the blade deformation shapes that a mode cannot be defined as belonging to one group or another. For instance, some blades may have a predominantly bending deformation, while others may have a predominantly torsional deformation. In higher modes, the traditional meaning of mode localization breaks down, and instead modal lines and stress patterns look disorganized.

With the existence of shrouds, greater coupling between blades generally exists, with combined stiffness and energy dissipation (dissipation via contact friction). While the stiffness coupling generally enhances blade coupling and alleviates localization, the non-linear effects of the energy dissipation have been shown by at least one researcher to have the potential to induce the rogue blade effect, in theory.

d. Aerodynamic mistuning

Aerodynamic mistuning is the variation of aerodynamic loading, from one blade passage to the next. While structural mistuning is generally represented as variations of the mass and stiffness matrices in a finite element model, aerodynamic mistuning manifests itself as a variation of the forcing function, or blade loads, from the symmetric case. Bladed disk assembly models are generally linear, with the exception of shroud, root, and root dampers. As a result, a doubling of the applied load will cause a doubling of the system response. As a result, aerodynamic mistuning does not in itself have the equivalent potentially-dramatic impact on blade response as structural mistuning does. However, unlike structural mistuning, it will always have an effect, and that effect is proportional to its magnitude. In addition, the location and motion of shock waves which travel along blades can dramatically change blade response. At higher modes, the projection of blade pressures onto the blade modes (which yields blade-modal forces) is dramatically sensitive to both blade modal mistuning as well as distribution of aerodynamic pressure. Clearly, the mistuned analysis is a larger problem than the tuned analysis. Consequently, recent work has focused on ways to reduce the size of the model computationally while the essential characteristics of the forced response phenomenon are retained. While a great deal of effort has gone into deterministically obtaining mode shapes and natural frequencies from prescribed mistunings, the mode shapes of the bladed disk cannot be confused with the operational deformation shape and amplitude. For example, exciting a simple cantilevered Euler-Bernoulli beam at its first natural frequency with a pressure distribution of its second mode will yield zero response. Also, as a result of the close spacing of modes (or repeated frequencies), and mode localization, the greatest potential response may occur slightly off-resonance(s), where the sum effect of being near multiple resonances is greater than the response at a single resonance. Therefore, pure modal responses are not sufficient to determine operational speeds at which the maximum responses will occur, or what that maximum response is. Simulations should include multiple modal responses. Both the modal frequencies and mode shapes should be known well. When rogue blade analyses is performed, simulations should include multiple system modes for a wide-enough set of cases to obtain a statistically-significant peak response distribution over what are defined as allowable mistunings in the bladed disk assembly operation specifications.

A number of potential reduced-order techniques exist. However, they are not well-tested by third parties. They should be used with caution. They must be validated for each problem to which they are applied until their capabilities are more fully understood.

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REQUIREMENT LESSONS LEARNED (A.4.13.3.2)

None.

A.5.13.3.2 Component vibrations.

Verification of model validity, modal characteristics, vibration amplitudes, steady stresses, and all other aspects of the HCF problem should be performed at each step of the design and verification process. An integrated approach where each stage of the design/verification process builds upon the previous one should be utilized. Verification should include numerical verification (sensitivities to key parameters), and data generated in component bench testing, rig testing, engine testing, and, ultimately, operational use. Established methods to compare experimental and analytical results should be employed where possible. Probabilistic design margins and predictions should be validated with bench, rig, and engine test experience in addition to statistical comparisons to operating fleet databases. Assurance is to be provided by verifying that the probability levels for each contributing random variable used to compute probabilistic design margins or probability of failure are within the experimental data range for that variable.

VERIFICATION RATIONALE (A.5.13.3.2)

An integrated approach to verification insures maximum benefit is gained from each effort expended in the course of the design/verification process. Execution of the task at hand with an understanding of what subsequent phases will require maximizes usefulness of information acquired. This ultimately maximizes knowledge acquired during development and reduces overall development and life cycle costs.

VERIFICATION GUIDANCE (A.5.13.3.2)

A methodical systems engineering approach should be taken to understand fully the design and test parameters that should be undertaken to identify and resolve HCF issues within gas turbine engines. Those design and test parameters, as well as a checklist for test protocol item compliance, are presented as follow:

a) **Design system**: The holistic test and evaluation approach recommended herein begins with the contractor's design system. The manufacturer's design system defines the tools, margins, criteria, and material data used for the design of a gas turbine engine. Given that a definition of robustness is insensitivity to variation, the test protocol first recommends that numerical assessments, including probabilistic predictions, be made to bound the range of variation that will potentially be present in a component. This requires that relevant influence parameters be understood. Some influence parameters may be geometric variations, variations in boundary conditions, local environment and body forces (e.g., RPM), etc. Assessments such as these can be performed by "brute-force" or through use of techniques like eigensensitivity analysis. The latter has the advantage of being useful in the identification of model regions where modal frequencies may be especially sensitive to geometric variations. These results can be used to create specific models for parts that are off-nominal if geometric differences are known (by CMM, for example), or to correct/understand discrepancies between experimental results and nominal model results.

   The manufacturer should insure structural models used for these studies are sufficiently representative of the actual structure. One way to address this is to perform a mesh density investigation to make sure that computed frequencies for normal modes do not change as a function of model discretization. Utilization of solid elements (isoparametric) with parabolic shape functions is recommended.

SUPERSEDES PAGE 142 OF MIL-HDBK-1783B.

	TEST PROTOCOL ITEM	COMPLIANCE?
I.	Design per Standard Work	
II.	Construct FEM	
	Solid Elements	
	Parabolic Displacements Functions	
III.	Perform Normal Modes Analysis	
	Mesh Density Assessment	
IV.	Sensitivity Assessment	
	Crystal Orientation	
	Geometric Variations (Eigensensitivity)	
	Boundary Conditions	
V.	Define Optimum Sensor Locations	
	Mode Measurement Capability	
	Modeshape Identification	
	Sensitivity to Sensor Misplacement	
VI.	Validate FEM	
	Frequency Comparison	
	Strain Ratio or Relative Displacment Comparison	
	Modal Assurance Criteria (MAC) or similar	
VII.	Compute Normal Modes at Speed	
VIII.	Define Limits for All Component Locations	
IX.	Design Experiment to Maximize Exposure to Influence Parameters	
X.	Test Rig and/or Engine	
	Process All Dynamic Data	
	Transform to Frequency Domain	
	Identify Modes Using Frequency and MAC	
	Apply Limits/Use FEM or FEM Derived Look-up Table	
	Database Results	
	Establish Statistical Variations from Database	
XI.	Assess Robustness	
XII.	Fix as-needed using Eigensensitivity to Move Problem Modes	

Partial ≈
Yes ✓
No ✗

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