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Comprehensive Structural Dynamic Analysis of the SSME/AT Fuel Pump First-Stage Turbine Blade

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TABLE OF CONTENTS

LIST OF FIGURES

TECHNICAL MEMORANDUM

COMPREHENSIVE STRUCTURAL DYNAMIC ANALYSIS OF THE SSME/AT FUEL PUMP FIRST-STAGE TURBINE BLADE

1. INTRODUCTION

Pratt & Whitney of West Palm Beach, Florida, is designing and building alternate high-pressure turbopumps for the Space Shuttle main engine (SSME). The fuel pump (see fig. 1) was in the final stages of testing in November 1997 when two airfoil sections of the first-stage turbine blades were liberated during a test, forcing a shutdown. Inspections showed that up to 50 percent of the blades in several units had cracks in the inside hollow core of the leading edge tips of blades (see fig. 2), and that the failure was a result of one of these cracks growing through the entire wall thickness of the blade. Many of the blades had been tested up to 13,000 sec with 21 starts.

Metallographic inspection of the cracked surfaces verified that the cracks were due to high cycle fatigue, which can be an indicator of substantial dynamic stress. Examination of the existing Campbell diagram showed no resonant crossovers with primary sources of excitation, so a more detailed

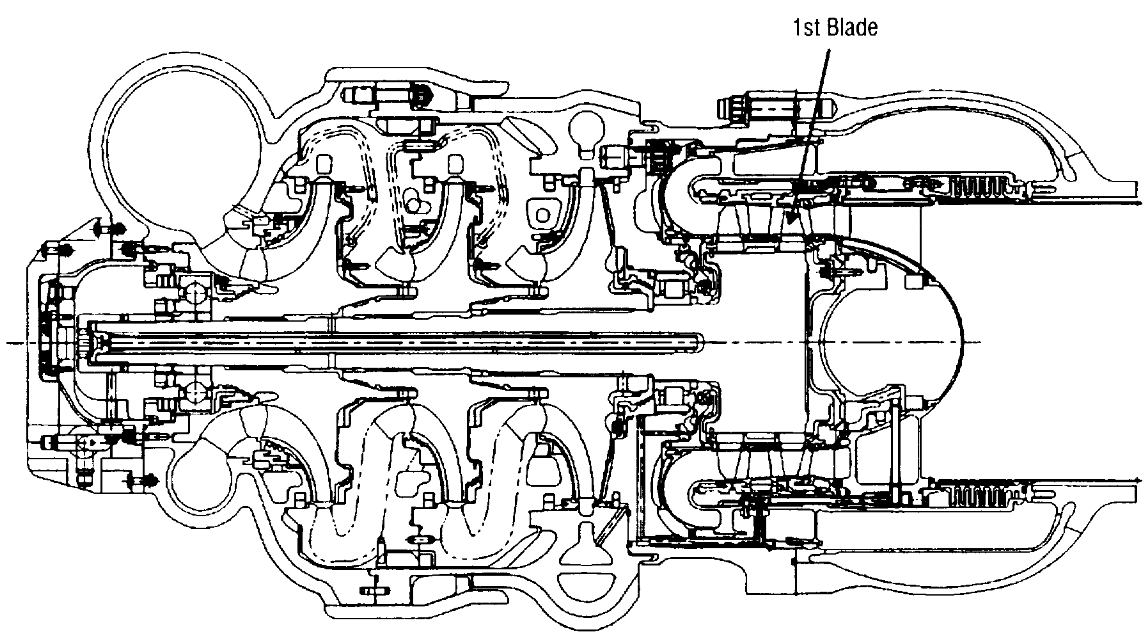

Figure 1. SSME/AT fuel pump.

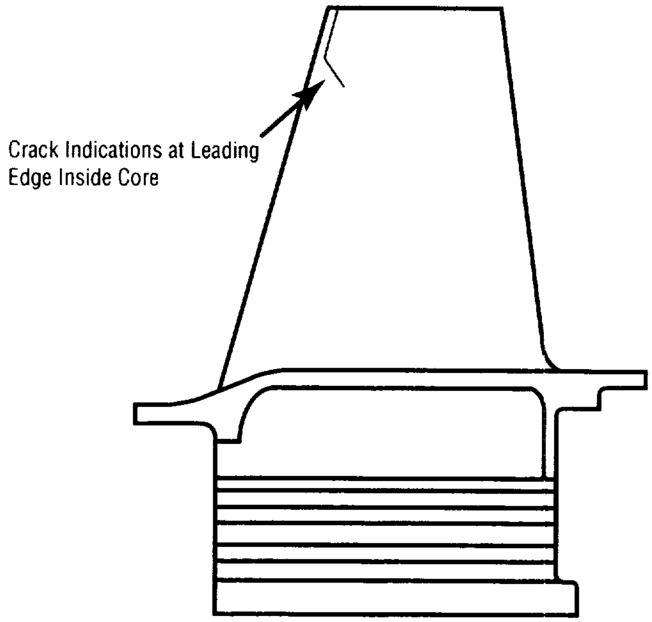

Figure 2. Crack in first-stage turbine blade.

examination of the character of the excitations and the stress distribution of the modes was required. To fully identify the source of the problem, evaluate proposed fixes, and make believable recommendations for problem resolution, virtually all of the analytical approaches available to the structural dynamics engineer had to be applied. These included modal displacement and stress analysis, frequency response analysis to tip loading from the first-stage blade outer gas seals (BOGS), fourier analysis of the excitation, and finally, a series of computer-intensive transient response analyses for different engine speeds and shock spectra analysis of the time history of the response. This problem, therefore, serves as an excellent example of the comprehensive dynamic analysis required for the resolution of high cycle fatigue problems due to structural vibration in highly stressed turbomachinery.

2. STRUCTURAL MODEL AND MODAL ANALYSIS

Pratt & Whitney had contracted Adapco, Inc. to build high-fidelity finite element models of the turbine blades for the alternate turbopump (AT) program. Because detailed in-house examination of the mode shapes and additional analysis was necessary, conversion of the Adapco ANSYS model to NASTRAN, which is the finite element code presently in use by the Structural Dynamics Branch at Marshall Space Flight Center (MSFC), was required. The Adapco mesh was therefore written to a geometry file and a FORTRAN program was written to read this geometry and write out GRID cards for use in creating the NASTRAN deck. The material properties were applied by splitting the blade into five sections, which have different properties, due to varying operating temperatures and using the appropriate material properties, for each section. Boundary conditions were applied by using springs along all the nodes on the fir tree contact location to generate a spring rate equivalent to the one used by Adapco, which was obtained by adjusting the spring rate until the first analytical blade natural frequency agreed with modal test. Spin stiffening effects were also examined, but were found to be negligible. The final NASTRAN finite element model is shown in figure 3.

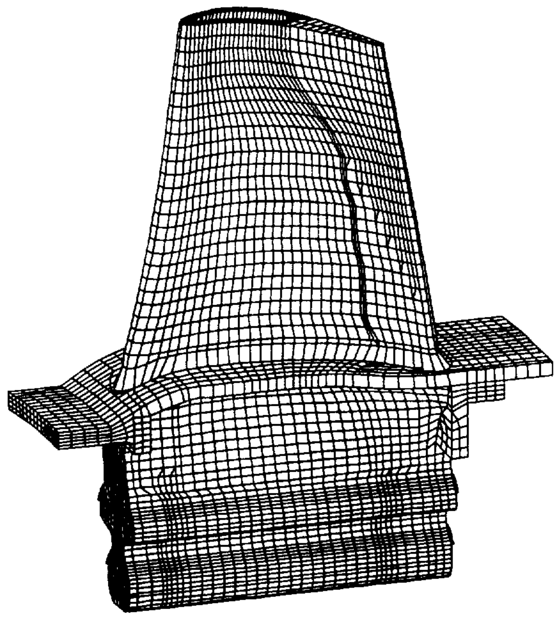

Figure 3. High-fidelity finite element model of first-stage turbine blade.

A modal analysis of the structure was performed and the modal displacement shapes and stress distributions examined in detail using the postprocessor graphics code PATRAN. The first 30 modes of the structure, up to 65,000 Hz, were obtained. This was substantially higher in frequency than the Adapco modal analysis. The first 10 natural frequencies are shown in table 1 and match the Adapco analysis within 4 percent.

Detailed examination of the modal stress plots at the crack initiation location at the inside of the tip core proved very illuminating. Previous cursory examinations of the modal stress for the entire blade missed the substantial stress concentration in many of the modes at this location; these included modes 6, 8, 13, 14, 20, 22, 23, 25, 27, and 29. The modal stress plots of modes 6 and 25 are shown in figures 4 and 5, respectively. The modal stress analysis was important in ruling out the identification of a "culprit" mode due to failure location and modal stress alone, as is possible in many cases.

Table 1. Natural frequencies of nominal first-stage turbine blade.

Mode	Adapco Frequency	MSFC Frequency
1	5,084	4,991
2	9,761	9,980
3	14,483	15,044
4	16,965	17,988
5	17,825	18,226
6	22,662	23,658
7	24,685	24,612
8	25,654	26,393
9	27,259	26,735
10	28,346	28,384

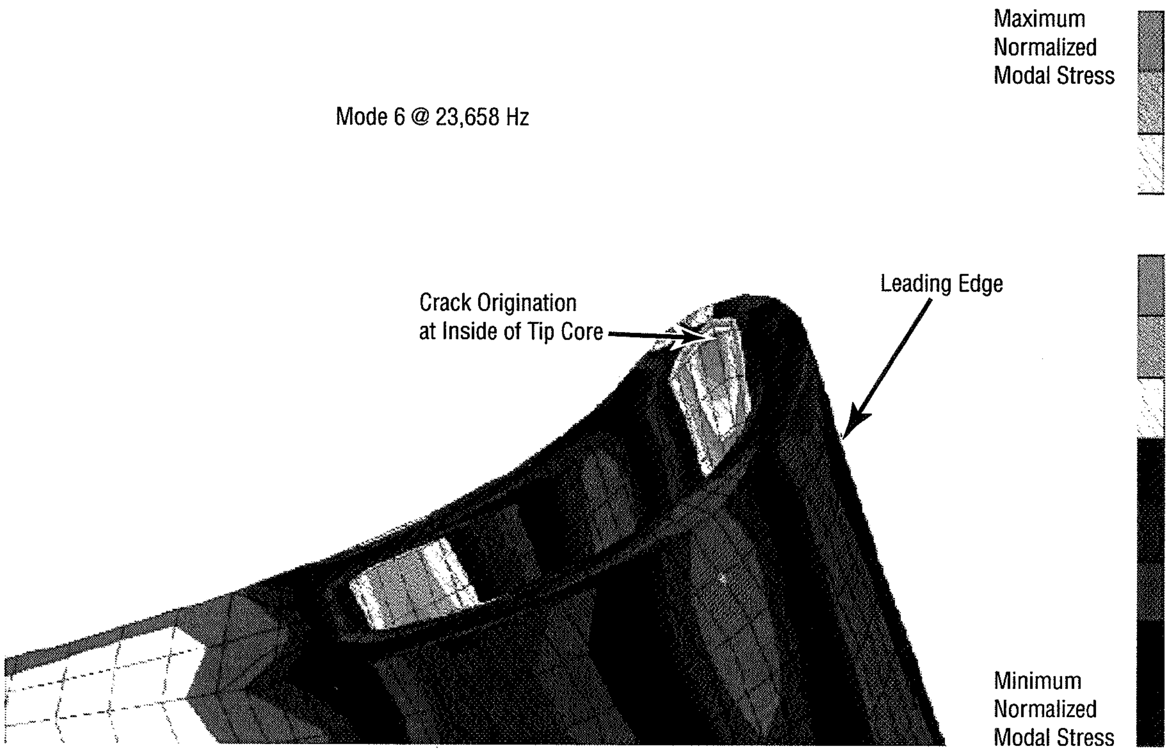

Figure 4. Mode 6 modal stress fringe plot looking at inside of leading edge tip core.

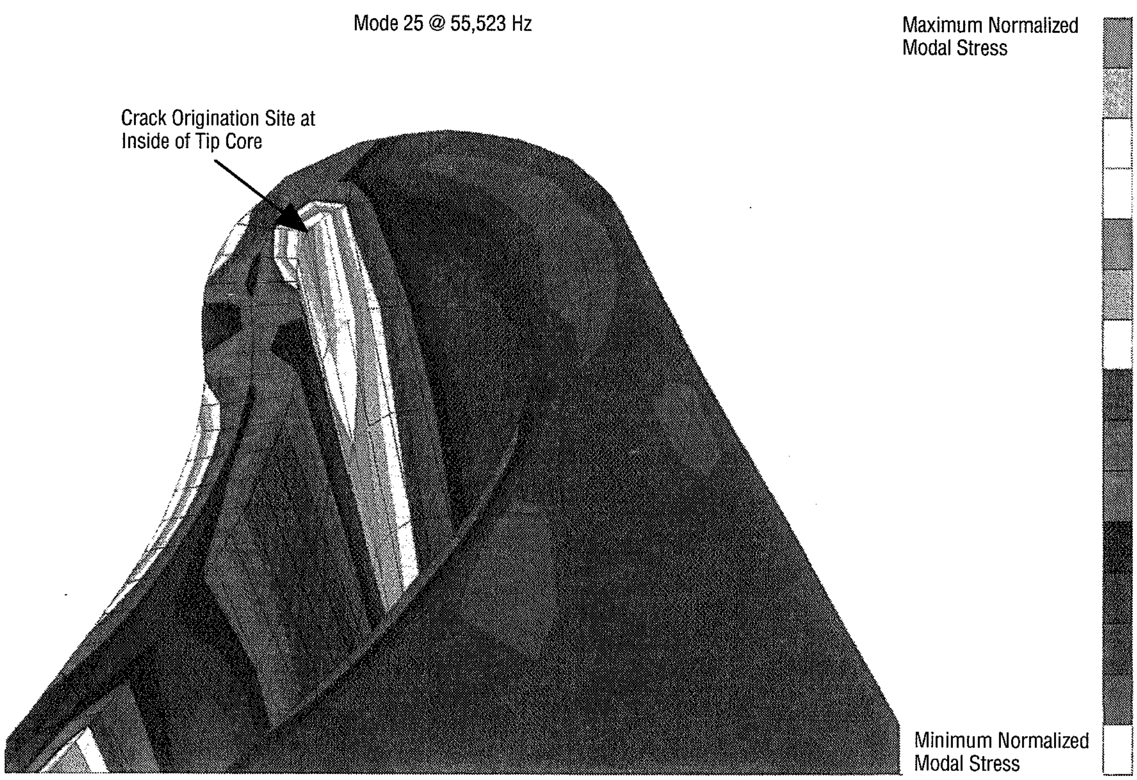

Figure 5. Mode 25 modal stress fringe plot looking at inside of leading edge tip core.

3. FORCED RESPONSE ANALYSIS

Previous forced response analyses of the blade had included loadings from the upstream and downstream stator vanes. In addition, an unquantified loading exists due to the drop in pressure on the blade that occurs as the tip passes underneath each of the small 0.063–0.076 cm (0.016–0.030 in.) wide gaps between 13 symmetrically placed 6.305-cm (2.482-in.) long BOGS, shown in figure 6.[1] This pressure drop occurs because the gap leads to a lower pressure cavity outboard of the blades. It was theorized that the evenly spaced BOG gaps would impart a harmonic load on the blade and induce resonance with a structural mode. A frequency response analysis to simulate this condition was therefore performed, using a unit tip load on the top row of blade elements. For this type of analysis, the loading is in the form of a purely sinusoidal wave, and the response of a given location is determined for the entire range of frequency excitation. One main result of the frequency response analysis is to show the relative stress at the tip cracking location, due to the tip loading for each of the modes. The frequency response for the 30–65 kHz range is shown in figure 7. The results from the analysis show that modes 1, 4–6, 13, 14, and 22–29 have high levels of stress response at the crack location.

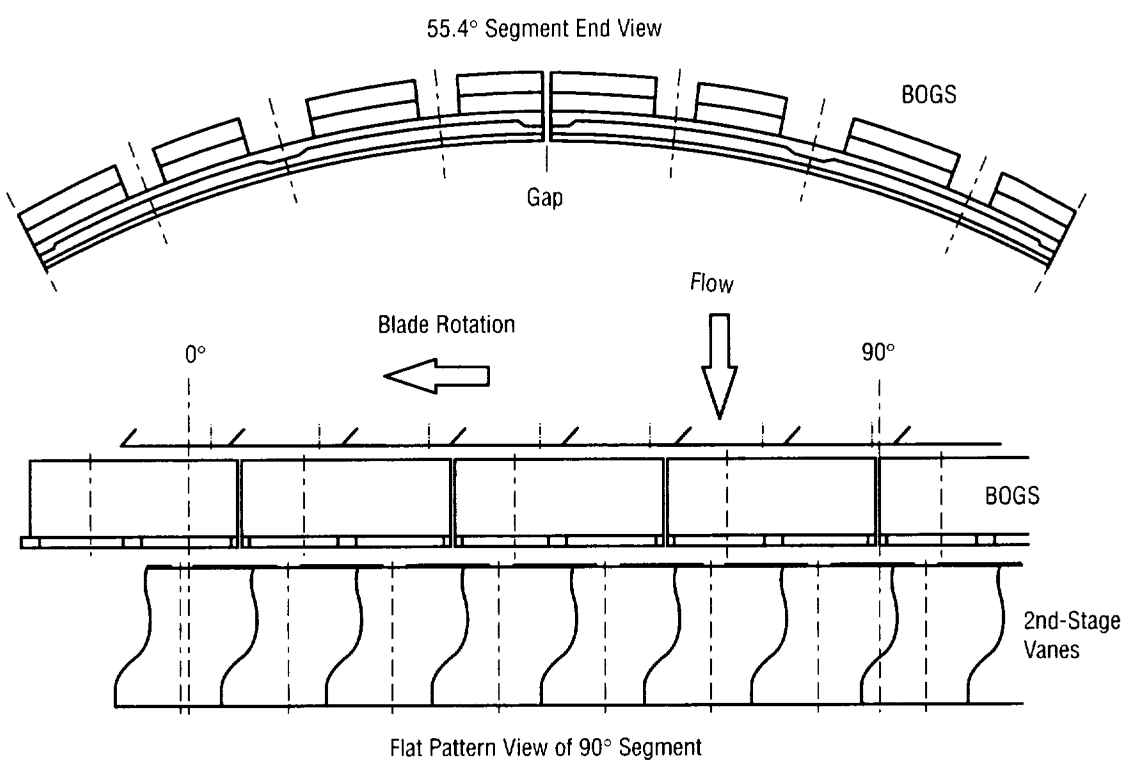

Figure 6. Blade outer gas seals (BOGS).

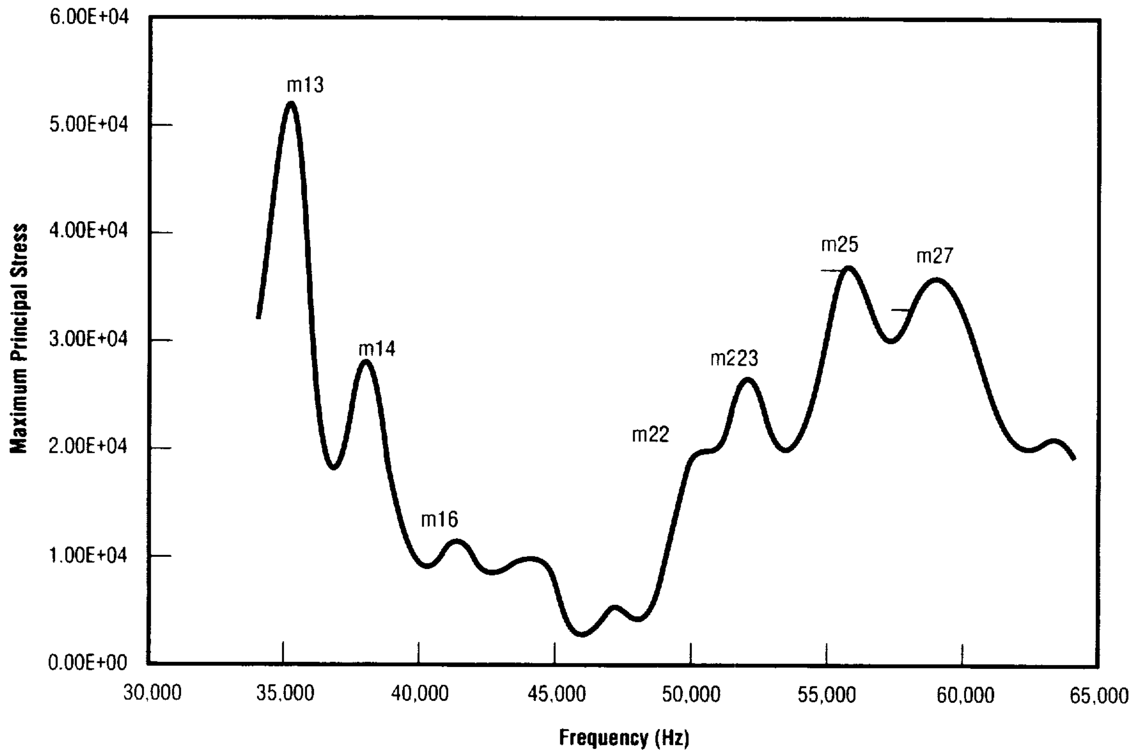

Figure 7. Frequency response to sinusoidal tip loading, 34–64 khz.

One proposed solution for the postulated resonance condition was to configure the seals with BOGS of unequal length to form an asymmetric pattern. This asymmetric configuration would supposedly result in a lower response because the resonance would not tune up with the regular excitation from the gaps. This design had been implemented previously on the second stage of the turbine after a similar failure. It was recognized at this time, though, that the excitation shape was much closer to a short triangular pulse than a sinusoid, so a fourier analysis was performed to quantify the relative strength of the harmonics of the pulse for different BOGS configurations. Since the magnitude and shape of the actual load were unknown, a FORTRAN program was written to generate an assumed loading, which consists of a series of unit triangular loads of length equal to four times the gap width, for any desired combination of different length BOGS. The program then calculates the fourier sine and cosine coefficients for the periodic pattern of period equal to one complete rotor revolution. The root sum square of these coefficients are then plotted to show the relative harmonic strength of excitations for multiples of an engine rotation. The results indicate that the existing configuration of 13 symmetrically placed BOGS has a very high harmonic content, e.g., the ninth harmonic of the primary 13E (E=rotor speed) excitation at 117 times the rotor speed (117E) has a strength equal to 86 percent of the primary. The 7/13+12/26 BOGS configuration (i.e., 7 BOGS with a length equal to 360°/13 combined with 12 BOGS with a length equal to 360°/26) has a very high strength at even multiples of 13, medium high strength at odd multiples, and introduces some energy at 1E, 13E,14E, 25E, 27E, etc. (see figure 8). This harmonic information can be combined with the frequency response information to identify engine speed levels of expected high response. This results in 104E being a large source of response because of its high relative loading strength in all the configurations interacting with the highly responsive modes 22–29. A partial Campbell diagram of the 7/13+12/26 configuration indicating the important resonant crossovers in the 31–36 kHz operating range resulting from this interaction is shown in figure 9.

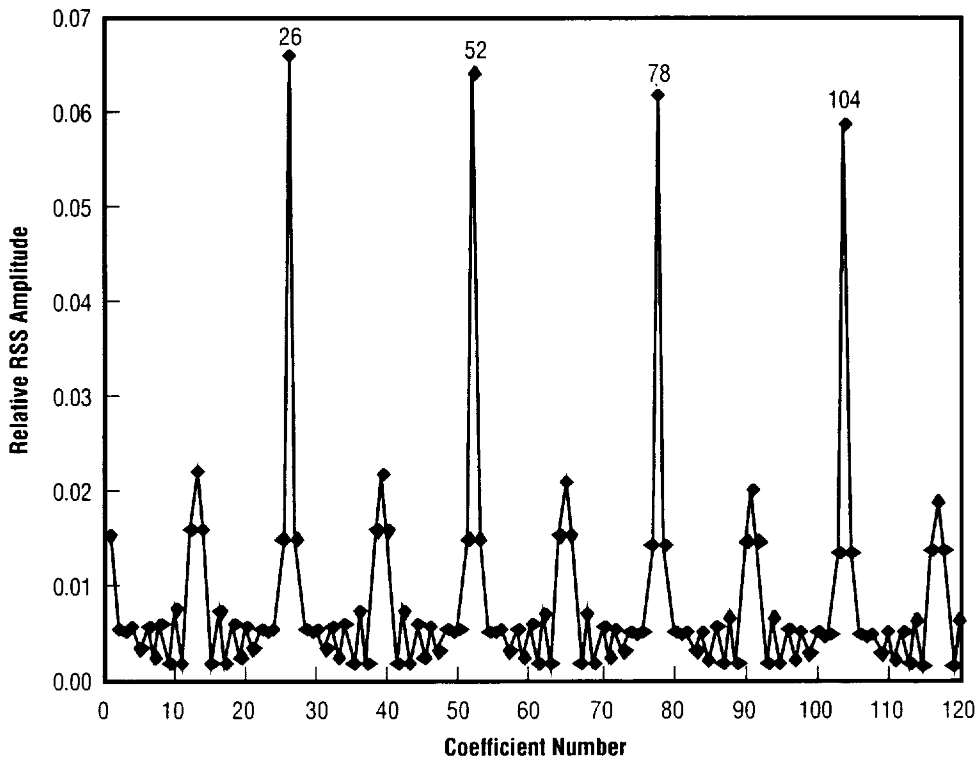

Figure 8. Fourier decomposition of 7/13+2/26 BOGS excitation.

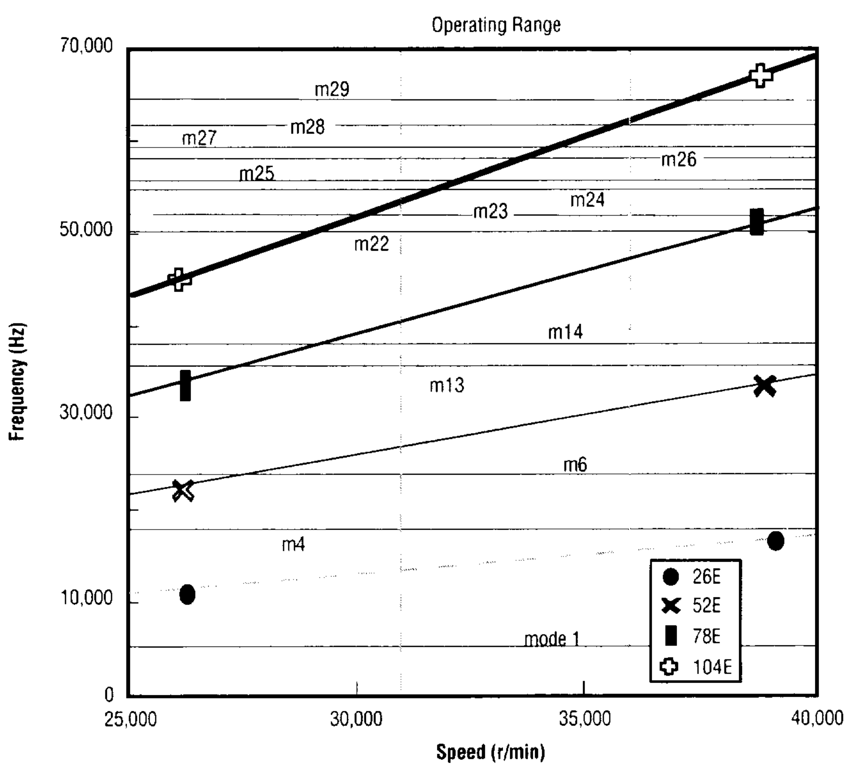

Figure 9. Partial Campbell diagram for 7/13+12/26 BOGS.

4. TRANSIENT RESPONSE AND SHOCK SPECTRA ANALYSIS

A transient response analysis of the stresses in the cracking location in the blade, as a result of the BOGS loading, was performed to simulate as closely as possible the circumstances of the cracking event. This type of analysis takes into account the complete energy content of the loading and the complete modal content of the structure. A FORTRAN program was written to create the NASTRAN TLOAD cards to specify the time history of the loading described in the last section. The desired rotational speed, BOGS gap width, and BOGS configuration can all be input into this program. The load was applied in a positive direction onto the top row of suction side elements, corresponding to a pressure drop on the pressure side of the blade as the tip passes under the BOGS gap. Since the elements at the tip will feel the pressure drop only when that element passes under the gap, a time delay was also calculated based on the circumferential location of each tip node and applied in the analysis. This loading is shown schematically in figure 10.

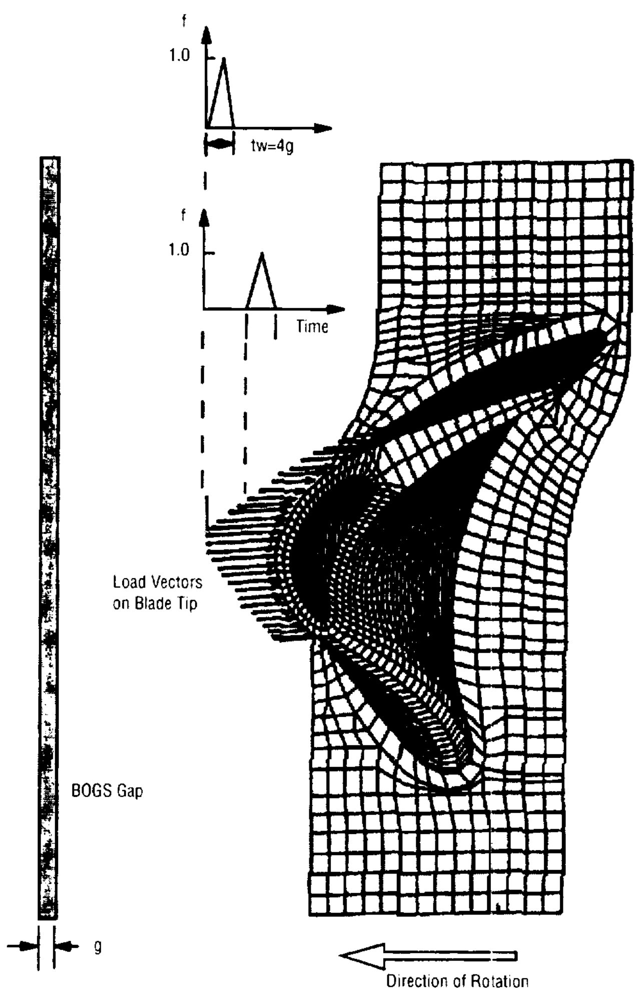

Figure 10. Loading of blade tip as it crosses under BOGS gap.

The analysis was performed for engine speeds which were expected to have high levels of response based on the previously performed analyses. To save computer time, the NASTRAN modal analysis was performed first and the datablocks saved. The transient analysis, which uses a modal representation of the structure and so is much more efficient than the original eigensolution, is then run using the RESTART capability, which uses the previously determined modal information. The timestep for the transient analysis was refined until the final answer stabilized. This resulted in a capability to run 30,000 timesteps covering 14 complete engine revolutions. A value of 2 percent structural damping was used rather than the standard 0.5 percent damping to reduce the required number of cycles in the run. For each run, a time history of the stress is plotted and the peak steady-state stress obtained. Examples of the loading history on one of the nodes at the tip and a stress time history of an element at the crack location are shown in figure 11.

A "sanity check" on these time histories was performed by applying a shock spectra analysis on selected cases. This type of analysis creates a one-degree-of-freedom spring/mass system that will have a natural frequency that varies over an entire frequency range, and applies a selected time history as the excitation of that system. The results essentially yield the spectral content of the time history, as the spring/mass system will have the largest response at significant frequencies in the signal. A sample spectra for the symmetric 13 BOGS configuration superimposed with the 7/13+12 configuration at a speed of 33,433 r/min is shown in figure 12. The peaks in the response match with the harmonics of the loading, and the maximum peaks occur when there is a significant modal response at that frequency. The maximum response is at 65E because of the resonance at that speed with modes 13 and 14 for both configurations, but the 7/13+12 asymmetric has a higher peak at 104E than the 13 configuration because it has a much higher fourier component at that harmonic. This result is consistent with the solutions obtained using the transient analysis.

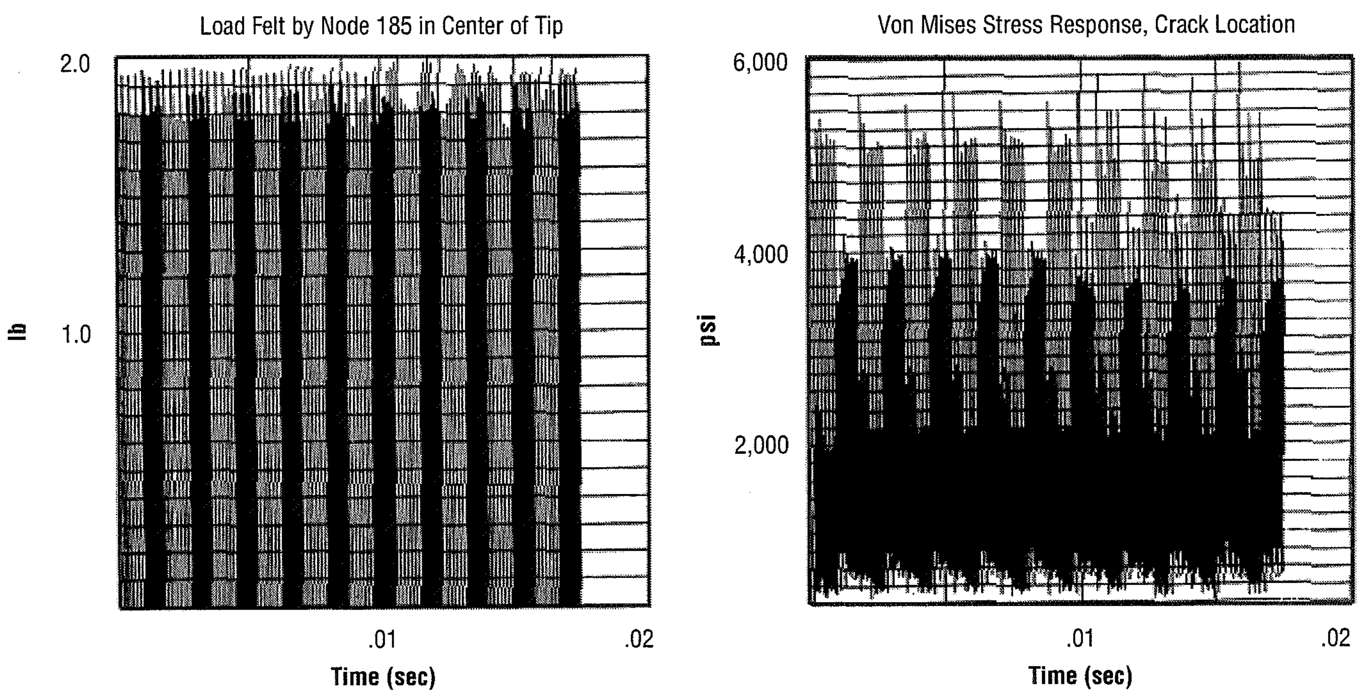

Figure 11. Time history loading and response of blade.

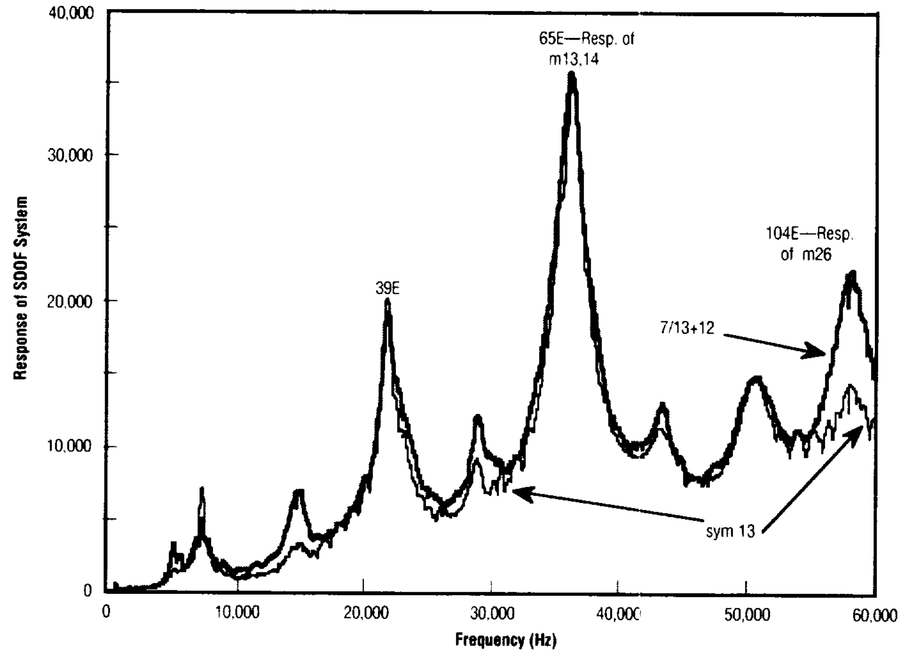

Figure 12. Comparison of shock spectra for symmetric–13 BOGS versus asymmetric 7/13+12 at 33,433 r/min.

The final results of the transient analysis, presented in figure 13, show that both proposed asymmetric configurations produce more stress at the tip than the existing symmetric configuration does. In particular, the asymmetric configurations show a significantly higher response in the 33,000–34,500 r/min range. Although there are some significant assumptions in the analysis involving the loading, the relative response of the different configurations should be accurate.

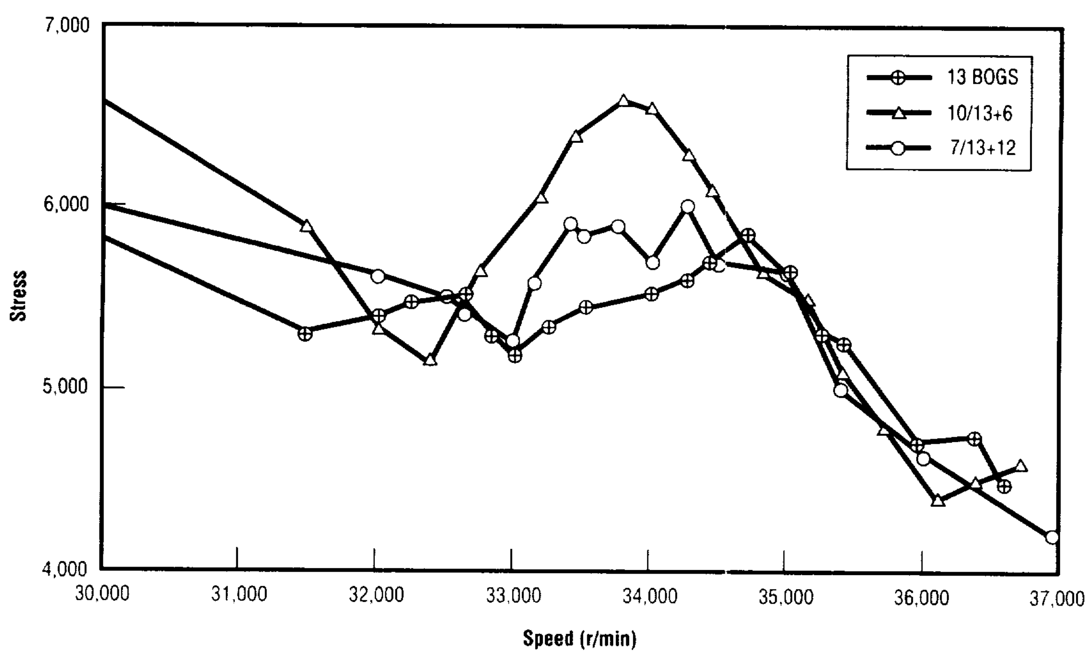

Figure 13. Stress response of tip element to tip load for different BOGS configurations.

5. DISCUSSION OF RESULTS

This result was somewhat unexpected based on the standard "Campbell diagram resonant crossover" method of dynamic analysis. Using this reasoning, the response would ramp up over several BOGS loadings due to a harmonic resonance with a specific mode of the turbine blade. The excitation from the gap, though, is actually more characteristic of a pulse than a pure sine wave. Therefore, the actual response, as approximated by the transient response analysis, is impulsive in nature, not harmonic. Each pulse excites many modes of the structure, as clearly indicated by the fourier analysis, and many of these modes exhibit a peak stress at the crack location. In addition, the peak response is reached within a few gap crossings, not after a slow ramp-up. The response is much more a function of the length and shape of the pulses than their spacing.

The impulsive nature of this response can be verified by examining the theory of impact loading.[2] The maximum response of a structure undergoing impact is less than twice that of the static response. A static unit load applied in the same manner on the blade resulted in a stress of 6,100 psi, which is at a comparable level with the response found in the transient analysis, as predicted by the theory.

Finally, the fact that this type of loading results in a fairly constant level of dynamic response relatively independent of speed is backed up by the fact that blades on several units, each of which had different amounts of run times at different speeds, showed extensive cracking. In addition, the stage 2 high-cycle fatigue cracking exhibited is indicative of a low level, constant level of dynamic stress as compared with a higher level of dynamic stress that would be exhibited in a purely resonant-type phenomenon.

6. CONCLUSIONS

Because of the nature of the loading on the turbine blades, the highly responsive modes could not be detuned by simply changing the BOGS count. Therefore, the resolution of the problem required that both the magnitude of the loading and the stress response in the cracking location be reduced. The reduction in load was accomplished by introducing a "shiplap" on the BOGS, which makes the path through the gap to the lower pressure cavity zigzag instead of straight. This limits the sudden drop in pressure as the blade passes underneath the gap. The blade stress concentration was reduced by redesigning the contour of the core of the blade. The implementation of these design changes has apparently resolved the issue. By performing comprehensive structural dynamic analysis on the blade, an expensive and detrimental asymmetric BOGS configuration was avoided, and successful design changes that solved the cracking problem were identified.

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2. Clough, R.W.; and Penzien, J.: *Dynamics of Structures*, 2nd Ed., McGraw Hill, NY, pp. 73–84, 1993.

APPROVAL

COMPREHENSIVE STRUCTURAL DYNAMIC ANALYSIS OF THE SSME/AT FUEL PUMP FIRST-STAGE TURBINE BLADE

A.M. Brown

The information in this report has been reviewed for technical content. Review of any information concerning Department of Defense or nuclear energy activities or programs has been made by the MSFC Security Classification Officer. This report, in its entirety, has been determined to be unclassified.

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13. ABSTRACT *(Maximum 200 words)*

A detailed structural dynamic analysis of the Pratt & Whitney high-pressure fuel pump first-stage turbine blades has been performed to identify the cause of the tip cracking found in the turbomachinery in November 1997. The analysis was also used to help evaluate potential fixes for the problem. Many of the methods available in structural dynamics were applied, including modal displacement and stress analysis, frequency and transient response to tip loading from the first-stage blade outer gas seals (BOGS), fourier analysis, and shock spectra analysis of the transient response. The primary findings were that the BOGS tip loading is impulsive in nature, thereby exciting many modes of the blade that exhibit high stress at the tip cracking location. Therefore, a proposed BOGS count change would not help the situation because a clearly identifiable resonance situation does not exist. The recommendations for the resolution of the problem are to maintain the existing BOGS count, eliminate the stress concentration in the blade due to its geometric design, and reduce the applied load on the blade by adding shiplaps in the BOGS.

14. SUBJECT TERMS

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