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BLADE DYNAMICS ANALYSIS USING NASTRAN

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SUMMARY

The complexities of turbine engine blade vibration are compounded by blade geometry, temperature gradients, and rotational speeds. Experience indicates that dynamics analysis using the finite element approach provides an effective means for predicting vibration characteristics of compressor and turbine blades whose geometry may be irregular, have curved boundaries, and be subjected to high temperatures and speeds.

The NASTRAN program was chosen to help analyze the dynamics of normal modes, rotational stiffening and thermal effects on the normal modes, and forced responses. The program has produced reasonable success. This paper presents the analytical procedures and the NASTRAN results, in comparison with a conventional beam element program and laboratory data.

INTRODUCTION

Accurate prediction of blade vibration in axial-flow compressors and turbines is one of the most important design steps in the development of modern gas-turbine engines. This prediction includes the calculations of blade natural frequencies and modes, rotational stiffening and thermal effects on the normal modes, and the blade forced responses. Vibration analyses performed previously with the use of beam element theory and a lumped mass approach were found to be inadequate because of the structural complexity in advanced blade design. The NASTRAN (NASA STRUCTURAL ANALYSIS) finite element method, modeling by plate elements, has produced reasonable agreement with the measured data, and therefore a computerized blade geometry generator has been developed to reduce the structural idealization effort. This development is incorporated with other dynamic analyses using NASTRAN.

NASTRAN BLADE DYNAMICS ANALYSIS PROCEDURE

A complete dynamics analytical procedure primarily for determining blade frequencies and modes using NASTRAN (Level 12.0) has been developed at Avco Lycoming Division. An outline of the theoretical approaches is described as follows:

1) An automated blade geometry generator using streamline definition is provided as a NASTRAN preprocessor, which constructs a grid-point pattern following the blade streamline flows and/or curved boundaries. This generator produces an accurate NASTRAN model of an irregular blade configuration, and it minimizes the input data preparation.

2) After the geometry of a blade is generated, NASTRAN normal mode analysis (Rigid Format #3) is used to perform the static natural frequency and mode shape calculations (no rotational stiffening effect). The eigenvalue extraction method (Inverse Power) is selected to determine the roots within a frequency range of interest. The results of the calculation for natural frequencies and modes are examined for design use.

3) The effect of rotational (inertia) stiffening on the natural frequencies and mode shapes of a rotating blade must be considered in the analysis. This is achieved, within the NASTRAN program, by introducing the preload stiffening effects ("differential stiffness" terms) into the free-mode calculations (Reference 1).

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4) Temperature variations in a blade will affect the structural stiffness and therefore the eigenvalue solutic Temperature distribution is reflected by material property changes, so the effect of temperature gradier can then be accounted for in the normal mode analysis with or without rotational stiffening effect.

5) The calculated eigenvectors from the previous analyses may be utilized as input data to the related mo of the NASTRAN program for a forced response analysis. An example of such an application is the NASTRAN transient analysis using the modal formulation method.

DESCRIPTION OF NASTRAN BLADE MODEL GENERATOR

A computer program to automate a blade structural model has been provided as the NASTRAN preproce: The model generator provides a punchout or a printout or both for all necessary definitions in a form suitable 1 NASTRAN bulk data input (Reference 2). This input includes the GRID space coordinates, CTRIA2 definition PTRIA2 properties, MPC constraint conditions, etc.

The program takes a blade geometry defined by a set of aerodynamic flow streamlines and the associated blade profiles (airfoils) to form a NASTRAN finite element model. The model grid-point pattern follows the streamline flows or the curved boundaries, or both, of the structure. The object of the model design is to ob.. an accurate blade model definition and to minimize the bandwidth of the gridwork for best computing efficien Finite elements with nearly equilateral triangles are formed by interconnecting the grid points. This interconne tion represents the middle surface of the curved blade, which has a rectangular XYZ-coordinate system referred the axial, tangential, and radial directions of the rotating machine. The calculation procedure for finite element presentation involves the following:

The program,

1) Determines the camber-line of a blade section given on a nonplanar surface.

2) Divides the camber-line and the blade length into segments according to an input percent value.

3) Calculates the cross-sectional thickness at each grid point location, starting at the leading edge and termin-ing at the trailing edge.

4) Interconnects the grid points between the two adjace: t blade sections to form finite elements with nearly equilateral triangles, starting from the tip and ending at the hub.

5) Transfers the initial vertical axis of a section to be coinciuent with the blade stacking line forming a rec-tangular XYZ-coordinate system referred to the axial, tangential, and radial direction of a rotor.

6) Deletes the rotational degree of freedom normal to the blade surface by defining multiple-point-constraint (MPC) conditions at each grid point. This constraint will eliminate the grid-point singularities.

As a demonstration related to the above calculation procedure, Figure 1 shows: a) a typical airfoil turbi blade section, b) the composite view of the airfoil profiles, and c) the two-dimensional blade model plotted by the generator.

Another example shown in Figure 2 is an undeformed compressor blade model with a streamline grid-point pattern. This figure was generated by the NASTRAN program in orthographic projection with the use of the NASTRAN preprocessor.

RESULTS

1) Normal Mode Analysis

NASTRAN normal mode analysis (Rigid Format #3) was performed to determine the natural frequencies and modes of both compressor and turbine blades of representative configurations. The blades analyzed are variable in geometry and are assumed cantilevered at their root fixity with complete boundary single-point constraints.

A) Compressor Blade Example

A compressor blade whose characteristics are a wide chord and thin section geometry (Figure 3) was chosen to demonstrate the NASTRAN calculations. This full-size blade has an approximate geometry as follows: aspect ratio = 1.75 (blade length/chord length at tip), twisting angle = 31 degrees (at the tip), and the maximum thickness taper ratio = 0.35 (tip/hub). Table 1 presents a summary of vibration data obtained from: 1) NASTRAN (using finite plate elements), 2) lumped mass vibration program analysis (using beam theory, Reference 3), and 3) shaker test of the actual blade.

Figures 3 and 4 show the resonant frequencies and nodal patterns (zero deflection lines) determined by the shaker test while using a stroboscope, hand-held vibration pickup, and oscilloscope.

The corresponding NASTRAN orthographic projections of the undeformed and deformed models are shown in Figures 5 and 6.

B) Turbine Blade Example

A shaker test was conducted with a power turbine blade, in a manner similar to the test with the compressor blade, to determine the resonant frequencies and vibration modes of a 10X size cast-aluminum model. The measured data were then used to confirm those from the NASTRAN analysis for the actual engine blade size by applying an equivalent scale factor.

The test model on its shaker mounting and the NASTRAN model generated by the preprocessor are shown in Figures 7 and 8, respectively.

Table 2 represents the results (natural frequencies and mode shapes) obtained from NASTRAN as well as by measurements.

2) Blade Rotational Stiffening Calculations

The present NASTRAN normal mode analysis is limited to nonrotating structures. However, the effect of rotational stiffening on the natural frequencies of a rotating blade can be included by using the program's DMAP (Direct Matrix Abstraction Program) feature. This objective is achieved by altering the original computational sequences so that the terms of the "differential stiffness" can be combined with the structural stiffness matrices (References 1 and 4).

The effect of rotational stiffening in the rotational field of a compressor blade (Figure 3) has been demonstrated. The frequency increase with respect to rotational speed are plotted on an excitation diagram (Figure 9). This data is compared with the corresponding data computed by an in-house vibration program, which employ the "transfer matrix" technique applied to a lumped parameter model of the beam.

3) Blade Modal Transient Response

The performance of the NASTRAN modal transient response (Rigid Format #12) was investigated with the use of the existing compressor blade model (Figure 5). The program's general functions were demonstrated by several computer runs with simplified dynamic loadings, so that the time-dependent forced responses of a blade may be studied in plots of displacement, velocity, and stress versus time. One of such plots, illustrating transient motion resulting from an arbitrary loading and damping, is shown in Figure 10 as an example.

4) Blade Thermal Variation Effect

The combined effect of high-temperature gradients associated with rotational speed fields on the dynamic characteristics of a turbine blade must be analyzed. The steady-state thermal variations within the structure will be reflected by material property changes from element to element. By superposing the thermal and centrifugal influences, the simulation of engine operating environments for a turbine blade can be accomplished. No numerical example is presented here.

DISCUSSION OF RESULTS

1) From the results summarized in Table 1, the NASTRAN finite element method has proved to be superior in accuracy to the vibration program employing beam theory. The use of a conventional beam element to idealize a blade structure will result in two inherent restrictions related to the beam theory: 1) neglecting warping displacements (i.e., plane sections remain plane), and 2) assuming no chordwise flexibility (i.e., each section retains its cross-sectional shape). The exclusion of warping constraints has significantly decreased the torsional rigidity, and therefore the torsional frequencies, of the beams. For the compressor blade analyzed, deviations of 29 and 30 percent compared with NASTRAN were found for the first and second torsional frequency respectively. However, a torsional frequency increase of more than 100 percent has been reported in thin-walled beams with open cross sections due to the inclusion of the warping effect (Reference 5).

The second restriction (above), assuming no cross-sectional deformation of the beam elements, introduces considerable errors in bending modes of higher order. The errors are particularly high for the blades with low aspect ratio and thin cross section, where the blade chordwise deformations must not be neglected.

2) The overall correlation between the laboratory measurements and NASTRAN normal mode analysis has been reasonably good, especially in the case of compressor blades, and for the frequencies of lower modes. Deviation of results attributed to mechanical tolerances, methods of measurement, and thickness approximation in model idealization may be expected. One of the significant differences is the turbine blade mode No. 4 (Table 2), which has not been identified by the NASTRAN in the search of eigenvalue solutions (Inverse Power Method). However, since test modes 4 and 5 show small distinction between their nodal patterns, it suggests that the additional laboratory confirmations are desirable before any conclusions may be made regarding the missing mode.

3) In the excitation diagram (Figure 9), the rotational stiffening effects obtained from the beam element model and the plate element model (NASTRAN) are compared. To simplify the comparisons, however, the bending frequency curves (dashed) generated by the beam program are assumed to be coincident with NASTRAN data at the zero speed so that the trend of frequency increases predicted by both programs can be compared directly. The NASTRAN-computed points at 12,000 and 20,000 rpm show a reasonable relation with the results from the beam program. Variation exists in the second blade bending mode; in this case NASTRAN indicates a smaller frequency gain. The difference could be attributed to the coupling effect between the NASTRAN second bending and first torsion modes because of their closeness in frequency. In addition, the centrifugal stiffening effect on torsional modes has also been predicted by NASTRAN (neglected in the beam program because of lack of elastic axis information), although the percentages of increase are relatively smaller.

The natural frequencies are observed to increase as the product (rotational speed)^2 (disk radius) increases since centrifugal force is a stiffening influence. However, the amplitude of frequency increase is also a function of blade aspect ratio and blade setting angle (Reference 6). The present study does not have sufficient data to evaluate these individual parameters, but it is felt that the plate finite element method will reflect the similar trend as experienced by the beam theory for these parameters.

4) In the forced response plot (Figure 10), the blade is acted on by an external harmonic force (arbitrary amplitude) having a frequency of 60 Hz which is 8.8 times as slow as the first mode of the blade (529 Hz). The cosine function periodic force is applied at a grid point on the leading edge in X-direction. The predicted transient response is constructed for a selected point on the tip. As shown in the plot, the total response at any instant between 0 and 0.022 seconds consists of the damped free vibration superposed on the forced motion. The displacement of the free vibration will, after a short time, disappear due to damping effect. Only the forced motion may continue. The higher frequency (529 Hz) appearing in the response corresponds to the first mode of the blade. Two lowest natural modes were introduced into the modal formulation transient response analysis.

CONCLUDING REMARKS

The automated blade geometry generator has significantly simplified the data preparation effort for the NASTRAN program. However, due to the generalized nature of this program (Level 12.0), the computing efficiency associated with eigenvalue extraction is low so that its use is costly.

NASTRAN finite element modeling using a plate element has provided an effective means for predicting blade vibrations. This conclusion is based on a comparison of results obtained from the NASTRAN program with experimental results and classical theory.

ACKNOWLEDGEMENT

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3) Towgood, D.: Coupled Bending-Bending-Torsion Vibration of Rotating Twisted and Tapered Blading, Using The Transfer Matrix Approach. Avco Lycoming Division Technical Report No. AND 065-10, Project Contract No. 555-8100.

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5) Skattum, Knut S.: Modeling Techniques of Thin-Walled Beams With Open Cross Sections. NASA TM X-2637, NASTRAN Users' Experiences, September 1972.

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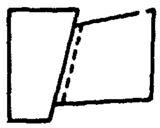
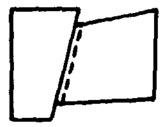
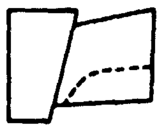
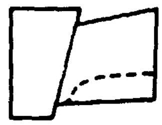
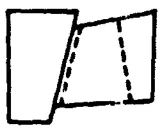
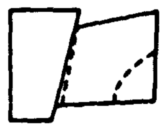
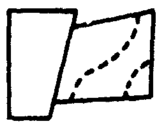
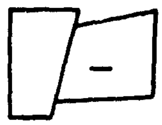
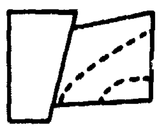
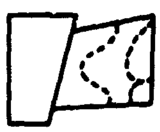
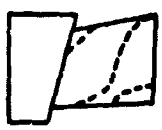
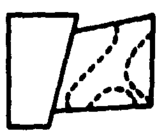
Table 1. Summary of Compressor Blade Vibration Data

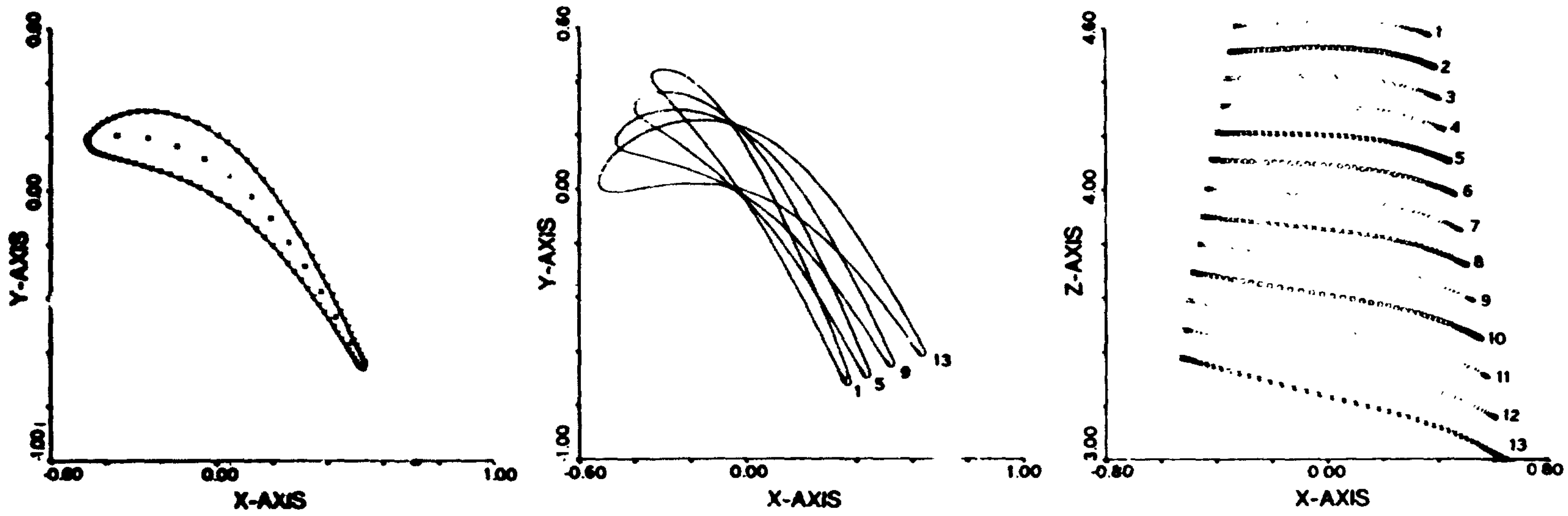
Natural Frequencies and Modes

Mode No.	Mode Name	Natural Frequencies, Hz		
		1. NASTRAN	2. Vibration Program (Beam Theory)	3. Shaker Test
1	1st Bending	529	540	486
2	2nd Bending	2026	2100	1856
3	1st Torsion	2125	1650	2130
4	3rd Bending	4483	4850	3410 3940 4200
5	2nd Torsion	4871	3740	4730 5300
6	4th Bending	6173	6400	–

Table 2. TURBINE BLADE VIBRATION DATA COMPARISONS NATURAL FREQUENCY AND MODES

MODE NO.	SHAKER TEST-TURBINE BLADE MODEL (10 x SIZE) MODE SHAPE	NATURAL FREQUENCY, Hz* (SHAKER TEST)	NATURAL FREQUENCY, Hz* (NASTRAN)	NASTRAN RESULTS ENGINE BLADE (1 × SIZE) MODE SHAPE	REMARKS
1		3.800	3.850		Test Frequency Was Adjusted to Compensate Model Root Radius Variation
2		7.940	7 700		
3		9.910	9.420		
4		12 610	–		NASTRAN Does Not Yield this Mode
5		14.620	14,180		
6		16 320	16.760		
7		23.990	21.890		
8		24 500	23.510		

*The Listed Model Blade (10 x Size) Frequencies Are in Terms of the Equivalent Actual Engine Blade at Room Temperature



a) A Typical Turbine Blade Airfoil Section Showing the Calculated Camber Line

b) The Composite View of the Airfoil Profiles

c) A Two-Dimensional Blade Model Generated by the Preprocessor

Figure 1.- Illustration of plots obtained from the blade model generator.

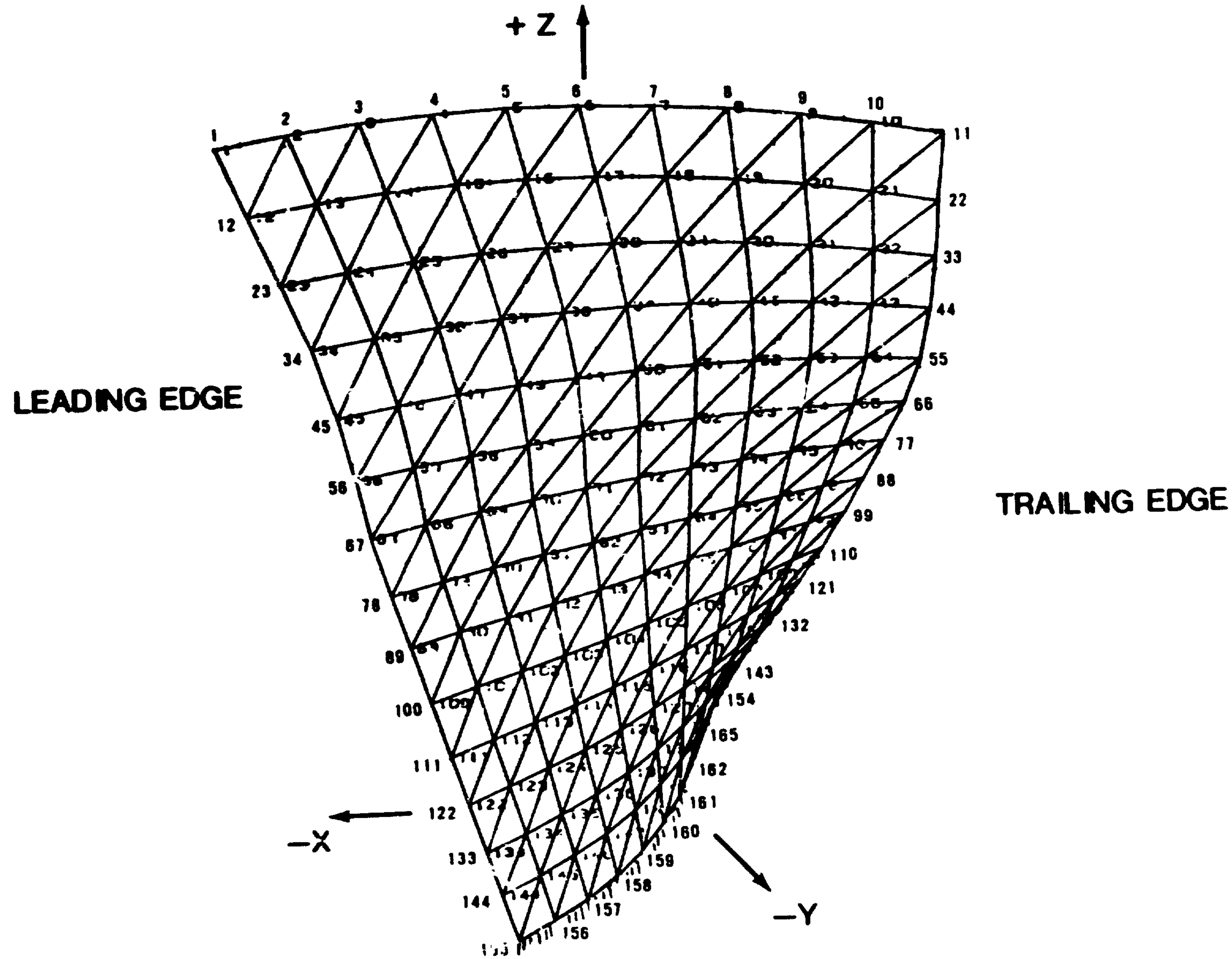
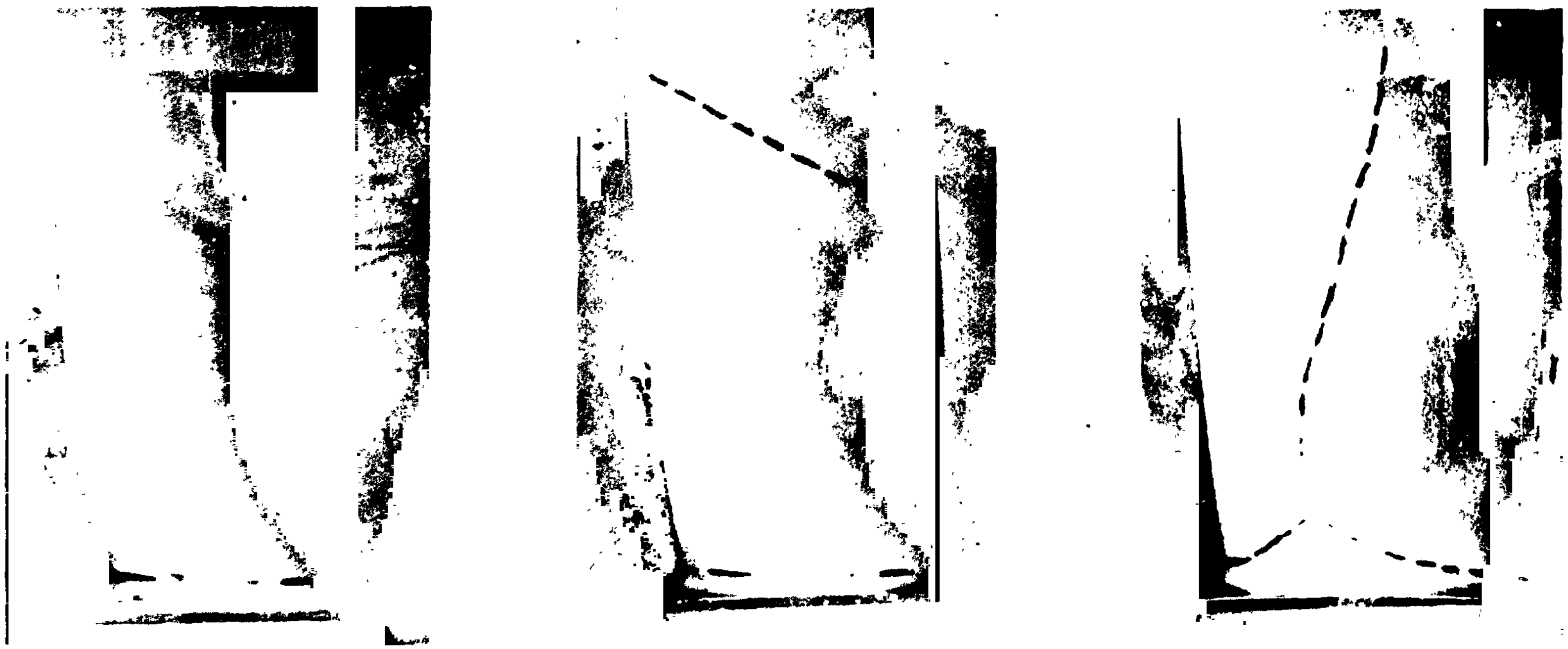


Figure 2.- Example of NASTRAN undeformed compressor blade model generated by using the blade model generator.



486 Hz
FIRST BENDING

1856 Hz
SECOND BENDING

2130 Hz
FIRST TORSION

Figure 3.- Shaker-test – wide-chord compressor-blade-measured natural frequencies and nodal patterns (modes 1, 2, and 3).

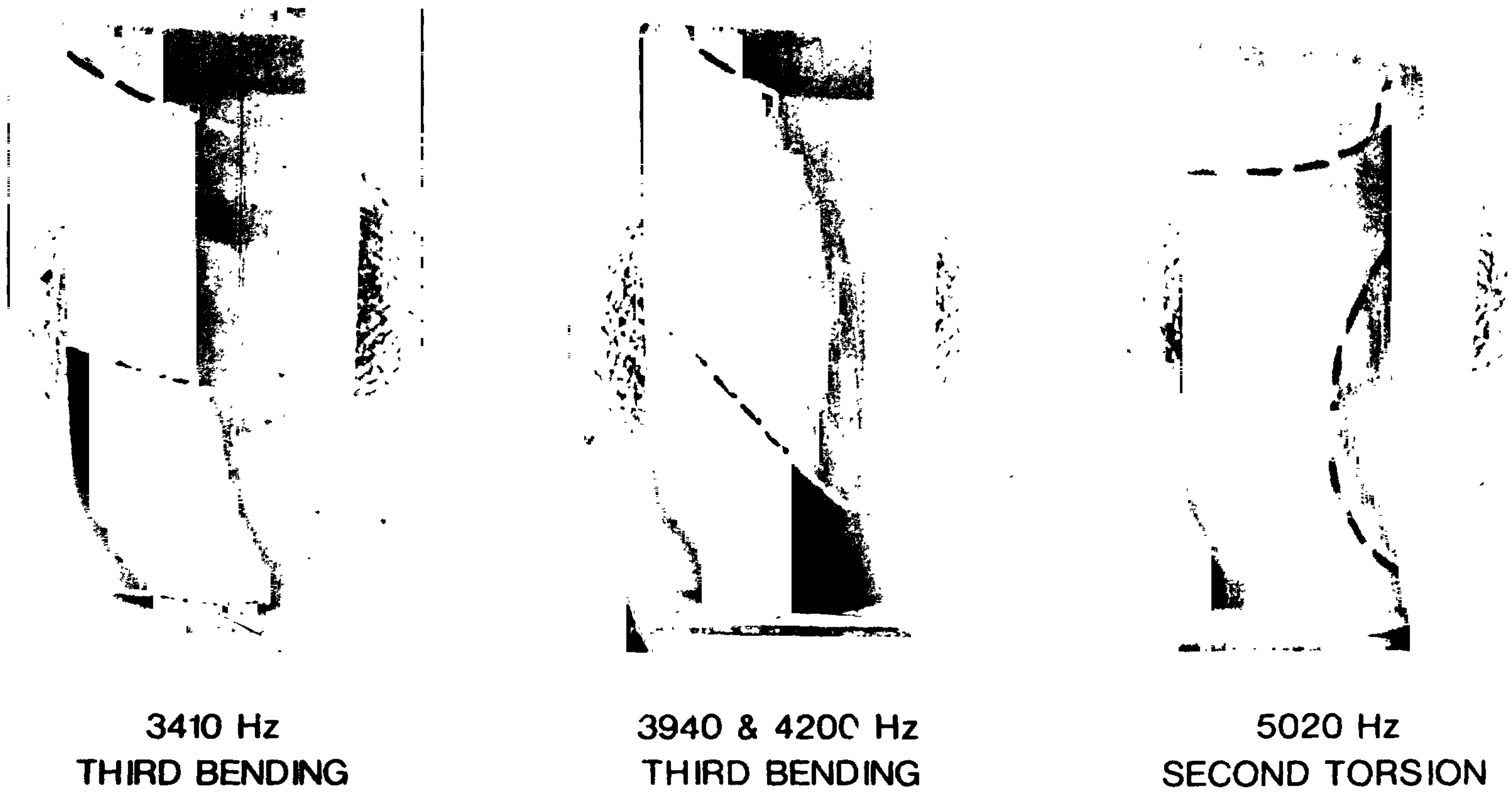


Figure 4.- Shaker-test – wide-chord compressor-blade-measured natural frequencies and nodal patterns (modes 4, 5, and 6).

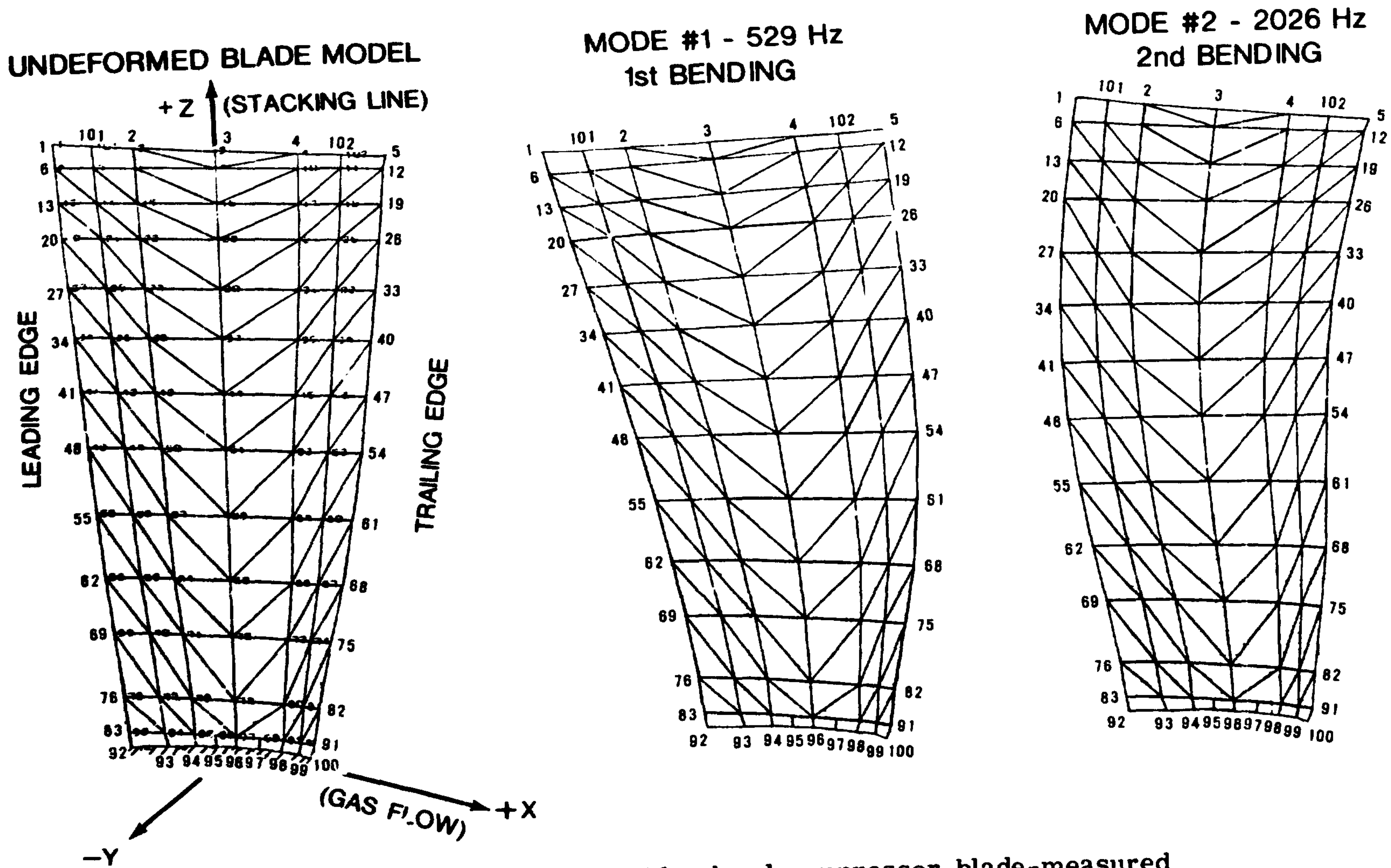


Figure 5.- NASTRAN model – wide-chord compressor-blade-measured natural frequencies and mode shapes (undeformed model, modes 1 and 2).

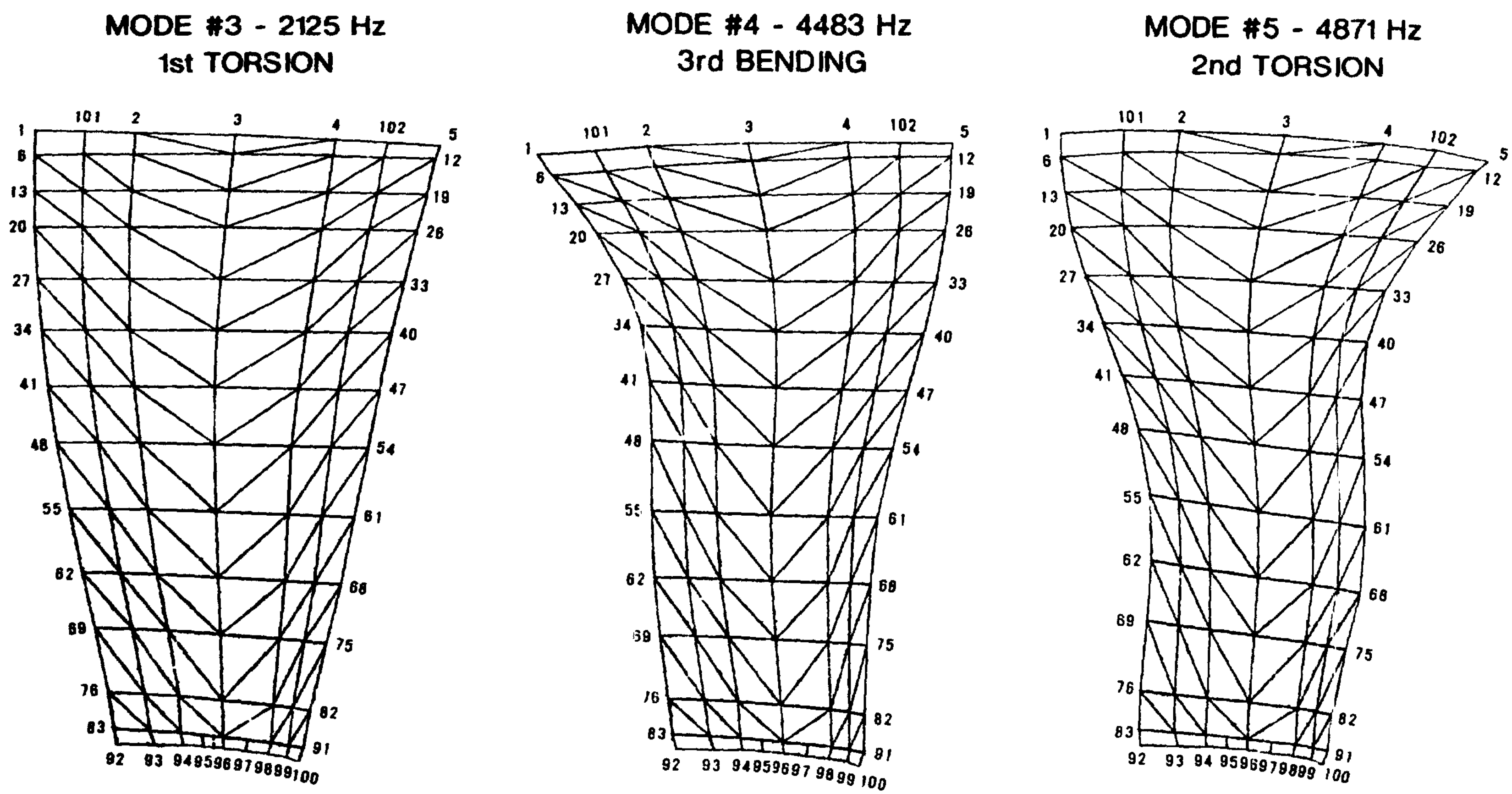


Figure 6.- NASTRAN model – wide-chord compressor-blade-measured natural frequencies and mode shapes (modes 3, 4, and 5).

Figure 7.- Vibration test setup. Power turbine blade (10 × size) aluminum model.

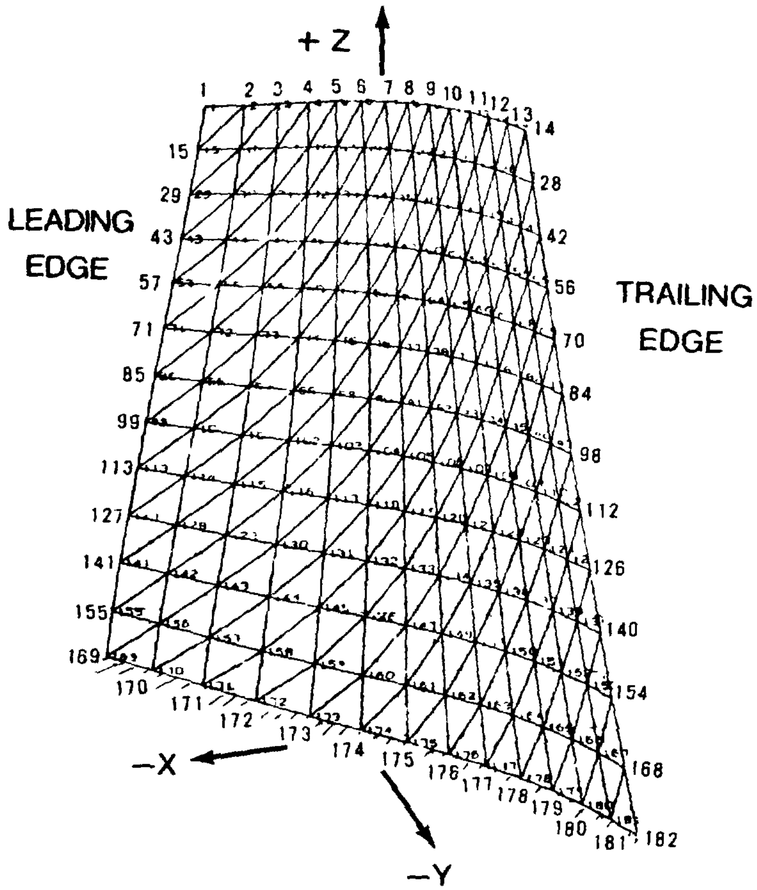


Figure 8.- NASTRAN undeformed turbine blade model.

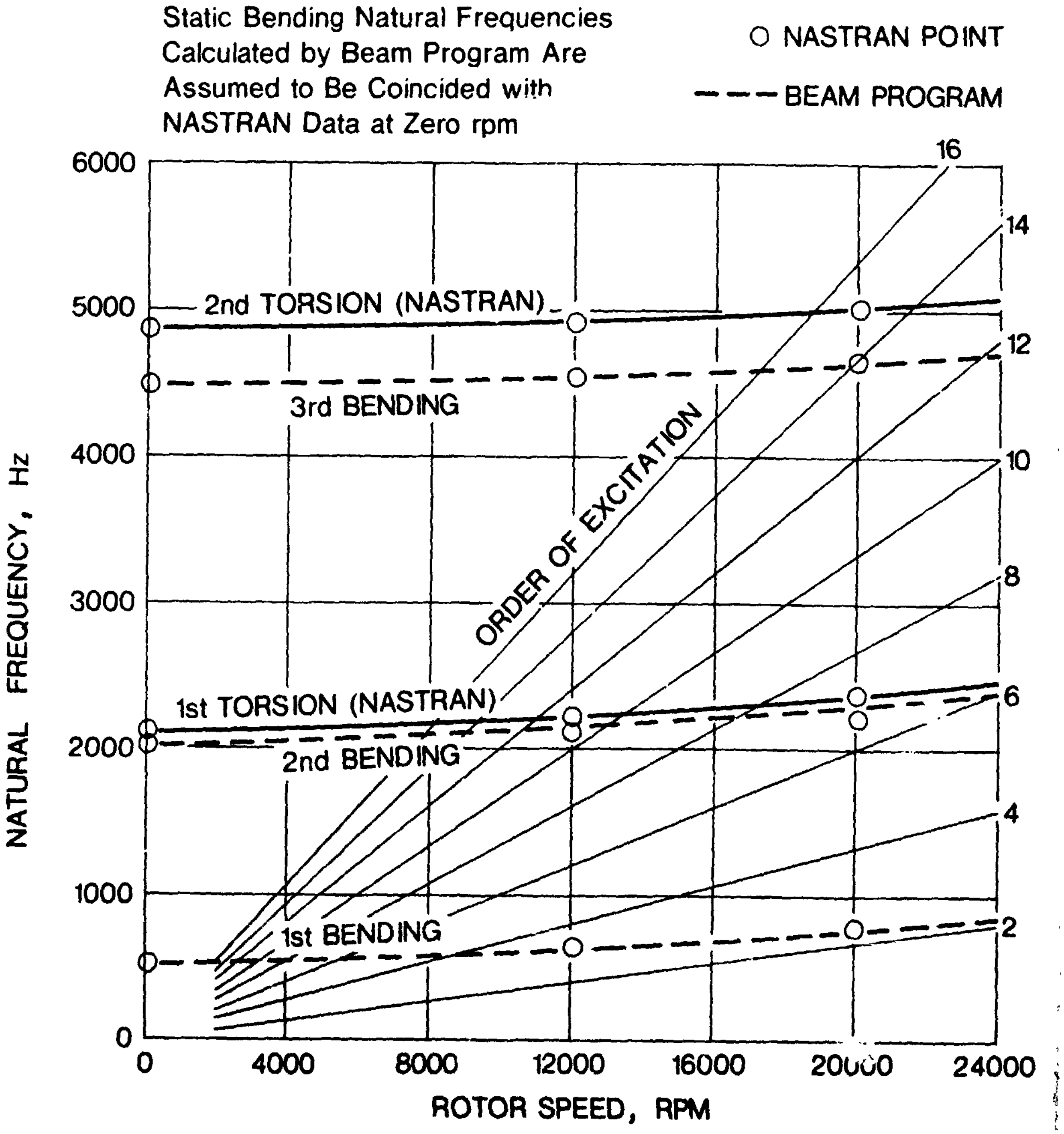


Figure 9.- Excitation diagram of the wide-chord compressor blade showing the rotational stiffening effects.

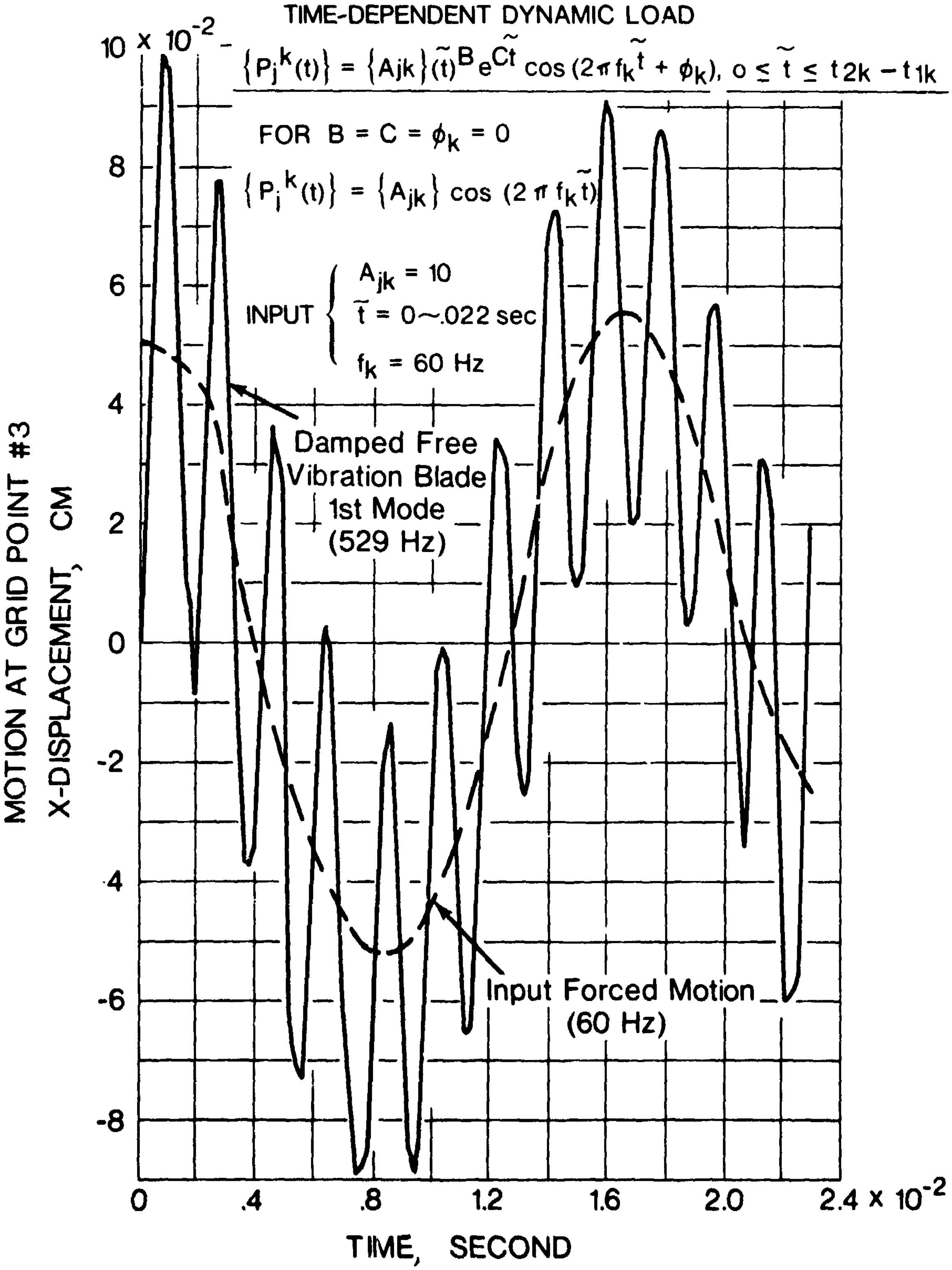


Figure 10.- Example of starting transient of the wide-chord compressor blade.