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The F-22 AMAD Gear Drive – Optimization of Resonance Characteristics by Detuning, Coulomb Damping & Damped Force Response Analyses

by: Raymond Drago, P.E., Frederick Brown, P.E. Ashok Sane
and Dennis Stremick, Boeing Defense & Space Group



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**Raymond Drago, P.E., Frederick Brown, P.E. Ashok Sane and Dennis Stremick,
Boeing Defense & Space Group**

[The statements and opinions contained herein are those of the author and should not be construed as an official action or opinion of the American Gear Manufacturers Association.]

Abstract

Finite element analyses were performed during design and development of the F22 AMAD gearbox to ensure a design free of damaging resonant responses within its operating speed range. Ideally, design with respect to resonance characteristics would have no natural frequencies within the operating speed range. However, the wide speed range (from 1445 to 17 931 rpm) combined with the weight and geometric constraints imposed by the application (the F-22 is an advanced technology, next generation, fighter aircraft) made it impossible to attain this goal directly in the F-22 AMAD design.

The analysis and evaluation therefore involved several interrelated iterative steps which included detuning by geometric modification, evaluation of gear to forced vibration, and damping treatment to develop an optimum, lightweight design free of damaging resonant responses within 15% of the operating speed range.

This paper discusses the overall analytical approach, analytical methods and evaluation of gear response to achieve acceptable accessory gear designs. Design modifications, including developments in the modeling techniques required to accommodate the unique characteristics of these relatively small (compared to the usual helicopter main power gear) gears are also discussed. Test and analytical results are also presented and compared.

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Raymond J. Drago, PE
Technical Fellow
Boeing Defense & Space Group
Helicopters Division

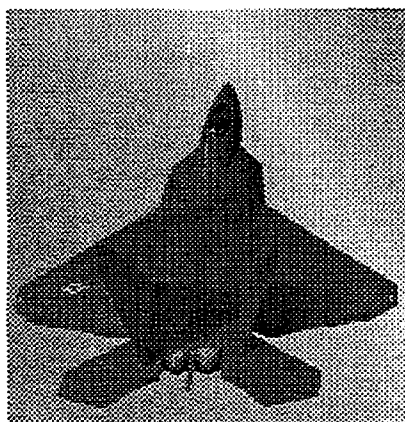
Frederick W. Brown, PE
Senior Technical Specialist
Boeing Defense & Space Group
Helicopters Division

Ashok D. Sane
Staff Engineer
Boeing Defense & Space Group
Helicopters Division

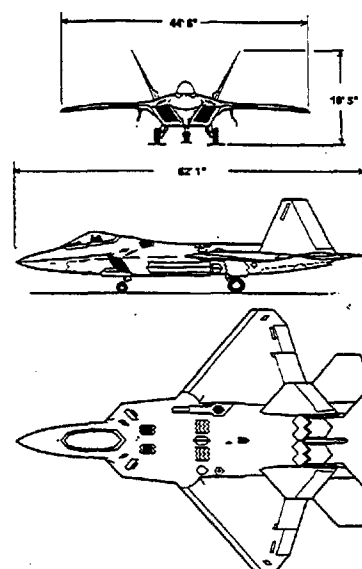
Dennis M. Stremick
Senior Principal Engineer
Boeing Defense & Space Group
F-22 Auxiliary Power Systems

NOMENCLATURE

The mode shapes referred to in this paper have been identified by the following convention:



SR	=	Shaft Rocking
UMB	=	Umbrella/Axial
WB	=	Web Bending
ST	=	Shaft Torsion
2DR	=	Rim O/P Bending with Coupled 2D
3DR	=	Rim O/P Bending with Coupled 3D
4DR	=	Rim O/P Bending with Coupled 4D
2D	=	Rim 2nd Diametral
2DSB	=	Shaft 2D @ Bearing
SB	=	Shaft Bending



INTRODUCTION

The structural criteria for AMAD (Airframe Mounted Accessory Drive) gearbox developed for the F-22 advanced technology, next generation fighter required that all gear must be free of damaging resonances within the operating speed range. Figure 1 shows an isometric views of this gearbox and a schematic of the gear train arrangement is illustrated in Figure 2. While the structural requirement of a gear free of detrimental resonant frequencies can be demonstrated by testing (bang test, dynamic strain survey) after gears are manufactured, the economics and schedule constrains dictated use of analytical methods during design and

development phase of AMAD to avoid costly redesign at the advanced design stage.

The wide speed range (from 1445 to 17931 rpm including 15% margin below and above the operating range) combined with geometric and weight constraints imposed by the application made it impossible to attain an ideal design free of all natural frequencies for all gears the AMAD design. The analysis and evaluation therefore involved several interrelated iterative steps which included detuning by geometric modifications, evaluation of gear response to forced vibration, and damping treatment to develop an optimum, lightweight

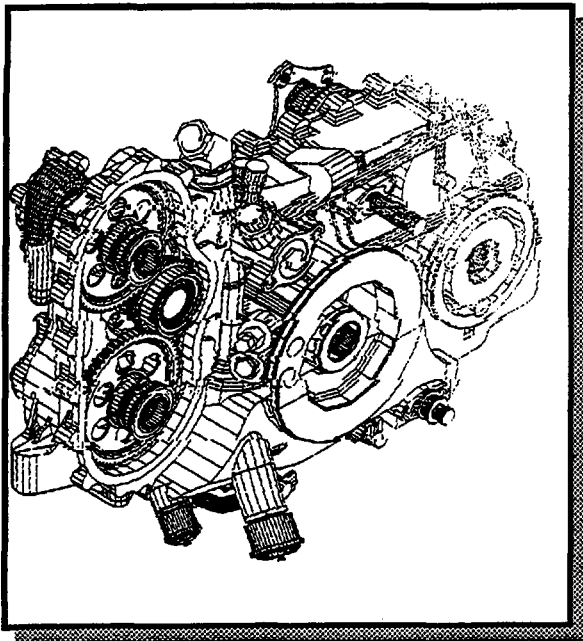


Figure 1 - AMAD Isometric View - Looking Aft

design free of damaging resonant responses within 15% of the operating speed range. The 15% margin below the minimum and above the maximum operating speed was added over the required structural criteria to allow for the uncertainties in analytically predicted frequencies.

This paper discusses the overall analytical approach and analytical methods used for evaluation of gear vibratory response to achieve acceptable accessory gear design. Design modifications, including developments in modeling techniques required to accommodate the unique characteristics of these relatively small (compared to helicopter main power gear) gears.

ANALYTICAL APPROACH

As stated previously, it was decided to use analysis rather than test to verify resonance characteristics of gears to support rapidly moving AMAD preliminary design and development phase and to avoid costly and time consuming design changes during the advanced design stage. AMAD is a compact design (Figures 1 & 2) consisting of several gears of varying size and speed with each presenting unique constraints to geometric modifications should it be required to detune damaging resonant frequencies. A preliminary modal analysis of these gears indicated that most of these gears had several natural frequencies within their operating speed range and that some gears had closely spaced multiple

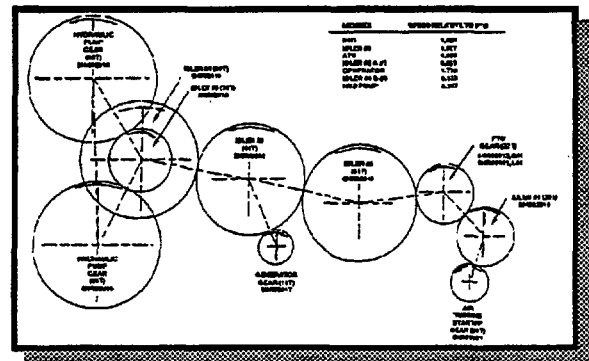


Figure 2 - AMAD Schematic Gear Arrangement

frequencies spread over the entire speed range. This was in sharp contrast to our past experience with conventional helicopter main power gears where the number of natural frequencies are usually much fewer due to narrow operating speed bands thus allowing a more conservative approach of detuning by geometric modifications to gear blank to accomplish a gear design free of any resonances within the operating range. This approach is simple, straightforward and safe; it does not require an evaluation of individual natural frequencies and their associated mode shapes to determine if they are likely to be excited by applicable gear forcing functions to cause damage. As an first attempt, we therefore decided to apply this 'Frequency Shifting' technique to all AMAD gears. However it soon became apparent that it was not a valid option for all gears of the F-22 AMAD due to wide speed range and existence of multiple frequencies over the entire range as well as just below and just above the operating speed range. The number of natural frequencies were so many for some gears that it would have been impossible to shift all of them out of the operating range without violating weight limits. We also found some gears with just 2 or 3 frequencies within the entire range. However they were placed at the upper and lower end of the wide operating band where an attempt to shift frequency at lower end caused frequencies from above the upper end to creep well inside the speed range and vice versa. Additionally, physical constraints placed by other AMAD components made it impossible to make minor geometric modifications for detuning natural frequencies judged to be damaging. A systematic analytical approach was therefore developed and applied to each AMAD gear to achieve an optimum lightweight gear design free of damaging resonant responses within the operating speed range. Our overall analysis and evaluation involved several interrelated and at times iterative steps as follows:

1) Develop an FEM (finite element model) of the gear under consideration, perform free-free modal analysis and identify natural frequencies that are candidates for potential resonances through the use of Campbell (1) Diagram within the operating speed range of the gear. If no such frequencies exist, the gear is acceptable from view point of resonance characteristics.

2) If natural frequencies for potential resonances are identified within the operating range from step 1 above, evaluate mode shapes to separate them in three categories:

A. Insignificant modes such as shaft bending, shaft torsion and complex modes unlikely to be excited by gear contact forces.

B. Complex modes and coupled modes (such as rim out of plane bending coupled with secondary diametral response) which may or may not be significant for known gear forcing functions.

C. Simple modes which are clearly gear teeth sensitive (e.g., 2D or 3D radial rim modes in toothed region, etc.)

3) No action is required on gears in category A since they involve insignificant modes or the modes unlikely to be excited by gear forcing functions.

4) Conduct forced response analyses (2) on the gears in category 2B to determine if any of the identified modes cause damaging resonances within the speed range of interest and/or large unacceptable displacement responses which are likely to be significant enough to warrant either damping or detuning.

5) For gears in category 2C and for those found unsatisfactory by the forced response analysis in category 4, define geometry changes to shift resonant frequencies out of the speed range and confirm (by forced response if necessary) that there are no damaging resonances in the operating range.

6) Define damping ring weights and locations, to damp out damaging resonant responses, for gears in category 5 which could not pass the resonance acceptability criteria by practical changes to gear blank geometry.

ANALYTICAL MODELS AND METHODS

A full 360 degree NASTRAN finite element model (FEM) was developed for each gear using 8 noded solid (CHEXA) and a few 6 noded wedge (in transition regions) elements. A 2D CATIA geometry of the gear was imported into PATRAN preprocessor for building the model. Each FEM included gear rim, web and the integral shaft. In earlier models, gear teeth were simulated by a solid ring from gear root to pitch diameter based on our past experience (mostly with Helicopter main power gears, Reference 3) that this

idealization of gear teeth was adequate for the purposes of gear resonance analysis. Subsequent analytical and experimental comparison of natural frequencies with other applications involving light weight gears (especially those with smaller rim to tooth height ratio) however, indicated that this idealization could over predict natural frequencies by as much 20% or more. All AMAD gear models were therefore revised to include a detailed finite element representation of gear teeth. The element mesh size was generally dictated by complexity of gear geometry, number of teeth and the importance of tooth mesh frequency in the gear resonance analysis rather than number of modes required and the operating speed range.

A free-free modal analysis was performed for each gear (as a first step of the overall analytical approach) using solution procedure 103 of NASTRAN. The Lanczos frequency extraction with generalized dynamic reduction was used for the modal analysis. All frequencies in 0 to 20000 Hz range were extracted to cover PTO speed range up to 15% above the maximum speed. Mode shapes were plotted by importing NASTRAN solution data into PATRAN. A modal analysis with 'Pinned' boundary conditions at bearings was also performed for those gears which required a forced response analysis evaluation. A forced response analysis was therefore performed (using 5% of critical damping) to quantitatively assess whether these modes would cause damaging resonances due to gear tooth excitation. A forced response analysis was performed for all gears involving modes which could not be classified as insignificant based on inspection of mode shapes alone. The forced response analysis was performed by simultaneously applying radial and tangential forces at the gear tooth contact. The rigid body responses were removed from the analysis by specifying restraints at the bearings and constraining one of the two bearings in UX (axial) and RX (rotation about x-axis) degree of freedom. A modal analysis with these boundary condition (pinned) was also performed to aid in interpreting forced response results.

APPLICATION

The analytical approach developed for the AMAD gears is illustrated through application to Generator, Idler 4 & 5 (also refereed to as cluster gear), Idler 1 and PTO/RH gears to illustrate typical actions taken and/or evaluations performed to demonstrate conformance of these gears to resonance criteria.

Generator Gear - A finite element model (FEM) of the

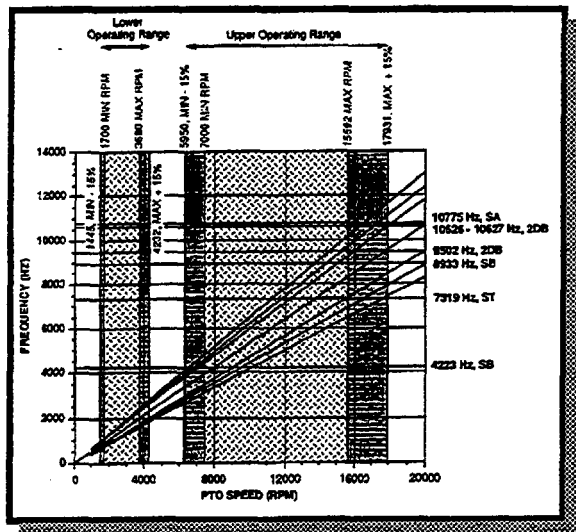


Figure 3 - Generator Gear Campbell Diagram

Generator gear was developed and a free-free modal analysis was performed as the first step of our analytical approach for evaluating these gears. The results of the basic frequency analysis, as shown graphically in the Campbell diagram in Figure 3, indicate that no potentially significant modes occur in the range of interest. For AMAD this range includes essentially all speeds from 85% of the minimum to 115% of the maximum AMAD operating speeds (that is 1445 to 17931 rpm referenced to the power-take-off shaft speed). Within this speed range, several frequencies for potential resonance are noted, however, these are not significant.

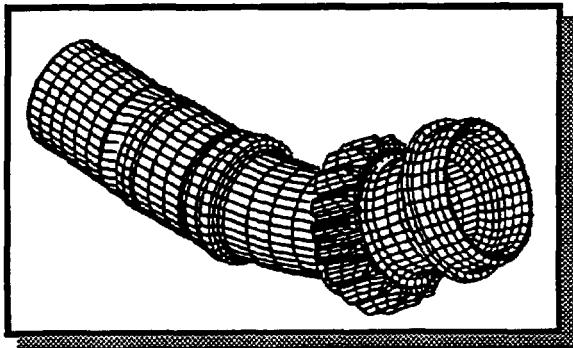


Figure 4 - Generator Gear, Shaft Bending (SB) at 4223 Hz

The shaft bending modes at 4223 and 8933 Hz (see Figure 4 for a typical shaft bending mode) are insignificant since they cause whirling response (due to gear rotational motion) rather than damaging radial or

tangential response in the critical gear tooth region apart from the fact that the frequency at which they occur are too high to be excited at lower harmonics (one or two per rev of the shaft maximum speed). Likewise, the shaft torsional mode at 7319 Hz shown in Figure 5 also is unlikely to be excited at lower harmonics to be of any concern.

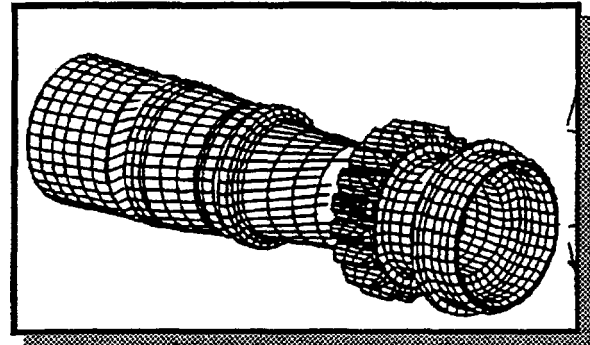


Figure 5 - Generator Gear, Shaft Torsion (ST) at 7319 Hz

The 2D modes at 9502 and 10626 Hz (Figures 6 & 7) show predominant diametral response under the bearings at the shaft end away from the gear. These will be restrained by the bearings and are therefore of no concern.

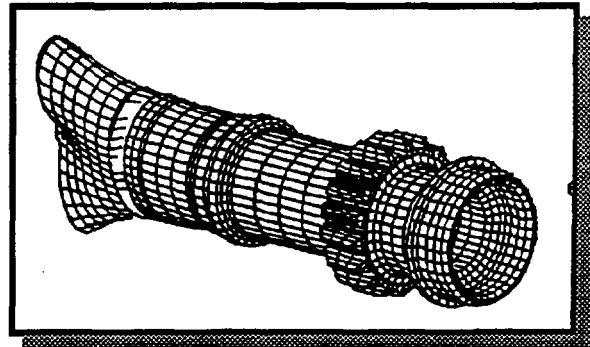


Figure 6 - Generator Gear, 2D at 9502 Hz

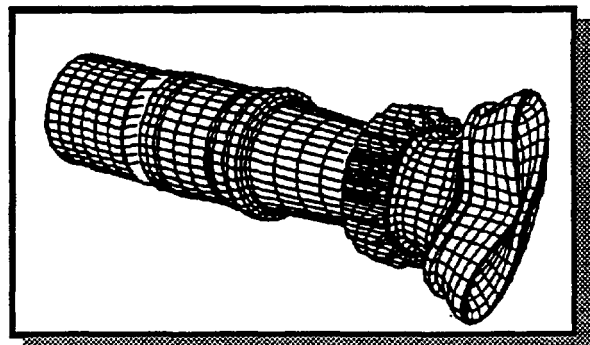


Figure 7 - Generator Gear, 2D at 10,626 Hz