Analysis of diesel engine crankshaft torsional vibrations

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INTRODUCTION

The development and application of a technique for the transient analyses of diesel engine crankshaft torsional vibrations is presented in this paper. Crankshafts in emergency diesel generators undergo torsional vibrations due to the effect of cylinder firing pressure and the inertia of the reciprocating parts. Techniques have been developed for the analysis of torsional vibrations under the steady-state conditions of constant speed and constant load. The work presented here extends these techniques for application to transient conditions. A diesel engine crankshaft is subjected to transients during startups, coastdowns, and load changes. Often the transients may result in torsional stresses that far exceed those normally designed for at constant speed and constant load. Computer program STAMS (Shaft Transient Analysis by Modal Superpostion) has been developed at Failure Analysis Associates (FaAA) for the dynamic transient torsional analyses of crankshafts. This program is based upon a modal superpostion approach, also known as the normal mode method, and provides an efficient means for the solution of forced vibration problems. STAMS predicts the free-end amplitude of vibration and the stress amplitudes at each shaft section in a crankshaft. The modal superpositition method is briefly discussed, followed by a detailed description of STAMS. Comparisons of the analytical results obtained using STAMS with test data obtained during testing of diesel generator engines is included. A brief description of the use of the results from STAMS is then presented.

MODAL SUPERPOSITION METHOD

Modal Superpostion is a technique for solving the equations of motion for a system [1]. This method makes efficient use of the eigenvalues and eigenvectors. Modal superposition utilizes the principal vibration modes as generalized coordinates to uncouple the equations of motion. By uncoupling the equations of motion in this manner, a single degree of freedom equation for each mode is obtained and the total response of a particular mode or a number of modes can easily be calculated. The eigenvalues and eigenvectors of the structural system are obtained by solving the undamped eigenvalue problem and normalizing the mode shapes with respect to the mass matrix. Then, they are utilized to transform the nodal mass, damping, stiffness and loading matrices defined utilizing physical coordinates and physical degrees of freedom, into normal (modal) coordinates. The solutions to the equations of motion are obtained using these generalized displacements. The solution is then transformed back into physical coordinates.

The shaft system is first discretized into a series of torsional springs and lumped mass moments of inertia. For the crankshaft, the inertia for each cylinder is lumped at the crankpin location. Figure 1 shows such a system consisting of a front-end gear, a twenty cylinder V-engine and a flywheel attached to a generator rotor. For each cylinder, the mass of both banks is lumped at one degree of freedom.

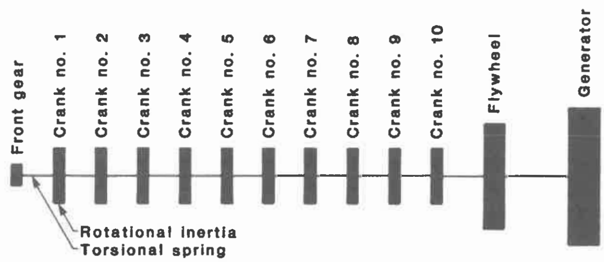

Figure 1: Discretized model of ten throw crankshaft.

The equations of motion for this discretized system with an applied load, $\underline{A}$ may then be expressed in terms of the displacement vector, $\underline{X}$, as

$$\underline{M}\,\ddot{\underline{X}} + \underline{C}\,\dot{\underline{X}} + \underline{S}\,\underline{X} = \underline{A} \tag{1}$$

where $\underline{M}$, $\underline{C}$, and $\underline{S}$ are the mass, damping, and stiffness matrices respectively. The system may then be reduced to a series of one degree of freedom equations in normal coordinates, for example for the i^{th} mode

$$\ddot{X}_{Ni} + 2\omega_i\gamma_i\dot{X}_{Ni} + \omega_i^2 X_{Ni} = A_{Ni} \tag{2}$$

where ω_i and γ_i are the natural frequency and damping for the i^{th} mode. The load for the i^{th} mode in normal coorindates is obtained from

$$A_{Ni} = \underline{\phi}_i^T \underline{A} \tag{3}$$

where $\underline{\phi}_i$ is the mode shape for the i^{th} mode which has been normalized with respect to the mass matrix. The displacement vector for the i^{th} mode is obtained from the transformation

$$\underline{X}_i = \underline{\phi}_i X_{Ni} \tag{4}$$

The loading vector, $\underline{A}$, consists of three types of forces, reciprocating inertia, friction, and cylinder pressure. Reciprocating inertia forces are handled as a force contribution since the accelerations that cause them are related to the engine speed with only a small contribution due to the vibratory accelerations. Similarly, friction forces are primarily related to the engine speed and are thus best treated as a right hand side or force contribution. Friction or damping contributions that are related to relative velocities are included in the damping matrix, $\underline{C}$. In practice, the damping for each mode, γ_i, is specified rather than $\underline{C}$. The two are related since

$$2\omega_i\gamma_i = \underline{\phi}_i^T \underline{C}\,\underline{\phi}_i \tag{5}$$

For analysis of transient response of the crankshaft, the torsional loads due to reciprocating inertia, friciton, and cylinder pressure must be determined at each time step and then transformed to normal coordinates. By pre-multiplying the load vector by the transpose of the eigenvector matrix in normal coordinates, solutions of each single degree of freedom equation are found using step-by-step response calculations based on Duhamel's integral. The responses are finally back transformed and summed for each mode.

Alternatively, the equilibrium equations may be solved using one of the direct numerical integration techniques such as Newmark's method. Again, this approach allows for a more general damping matrix, but does not make use of the extracted natural frequencies and mode shapes.

PROGRAM STAMS, SHAFT TRANSIENT ANALYSIS BY MODAL SUPERPOSITION

The torsional vibratory response of crankshafts during start-up and coastdown transients is of interest since the torsional stresses may far exceed those normally designed for at constant speed and load. The number of loading cycles imposed on a crankshaft during these transients is usually small; however, the magnitude of the loading may be large.

The computer program STAMS calculates the torsional natural modes of vibration of an axial system of springs and masses and the response of the system to applied loads. The solution obtained using STAMS sums the response of specified harmonic orders for each mode of vibration taking into account the correct phase angles for all cylinders. The summation is performed for each inertia and shaft section, respectively. The cylinder pressure and reciprocating inertia loads are generated at each time step. The reciprocating inertia is calculated based on the current value of angular velocity of the crankshaft and the current crank throw orientation for each cylinder. The transient solution is developed at each time step using a Duhamel integral for accurate integration of load through time [1].

Time histories, maximum and minimum values of vibration and nominal shaft stresses are provided in the output for each inertia and shaft section. Nominal torsional stresses, τ, are calculated from torques, T, by the equation $\tau = Tr/J$ where r is the journal radius and J is its polar moment of inertia. Results for the contributions due to selected orders or modes may also be obtained. STAMS includes internal modal damping in the analysis.

STAMS requires the following input: the number of shaft sections, shaft stiffnesses and diameters, lumped mass moments of inertia, number of banks, cylinders and firing order. V-engine angle, engine speed data, reciprocating mass per cylinder, stroke and connecting rod length, analysis type, cylinder pressure loading, the number of modes, and the damping ratio.

The engine speed and firing pressure data may be provided in a number of ways depending upon the type of transient. For a coastdown analysis, an indicator diagram of a typical cylinder without fuel and the relationship between engine speed and time are required. Test data from several diesel engines indicate that for a coastdown, the engine speed reduces linearly with time. If engine speed versus time data is unavailable, STAMS has an option allowing this linear decrease with time. For a start-up analysis, the angular velocity of the crankshaft must also be specified as a function of time. This may be done by measuring the speed of an engine as it passes through the start-up. The cylinder pressure may be specified either by supplying pressure versus crank angle data, which is then normalized to speed, or by measuring the cylinder pressure as a function of time in one cylinder and using it to interpolate the pressure in other cylinders. Ideally, a complete pressure trace for all cylinders is required; however, an analysis can be performed with less accuracy if pressure data from only one cylinder is available. For fast and slow starts, such as are required in emergency standby service for nuclear power stations, the initial position the crankshaft may also be important and is required as input.

<u>STAMS Results Compared to Test Data</u> An analysis of the transient torsional stresses in a DSRV-20 diesel generator crankshaft at San Onofre Nuclear Generating Station during

fast and slow starts and coastdowns was performed utilizing STAMS [3,4]. The DSRV-20 engine is a twenty cylinder V-engine with a 13×13 inch crankshaft. This crankshaft underwent transient testing in September 1984, as a part of diagnostic tests being conducted on the diesel engines. Time histories of cylinder pressure, engine speed, and crankshaft torsional vibration were recorded, for this engine.

A steady state analysis was peformed on these crankshafts utilizing computer program SHAMS- Shaft Harmonic Analysis by Modal Superposition. SHAMS was written by FaAA to analyze torsional vibrations of an axial or branched system of springs and masses due to harmonic loads. The results of the steady- state analysis indicate that the maximum stress amplitude in the crankshaft during normal operation is approximately 4000 psi.

The calculated natural frequencies of the shaft system were compared with torsiograph test data, and were found to be in good agreement. This comparison is used to indicate that the crankshaft structural system has been appropriately modelled.

The response for both coastdown and startup was determined based on cylinder pressure for one cylinder. For the coastdown analysis, a cold compression pressure curve was used with a peak pressure of 450 psi. The coastdown was assumed to take 70 seconds (based upon test data), and it was further assumed that the angular velocity decreased linearly from 450 rpm to 0 rpm during this time. The test data showed that this was approximately true although a more accurate representation of rate of decay of angular velocity would be slightly parabolic. A comparison of the coastdown response with torsiograph test data is shown in Figure 2.

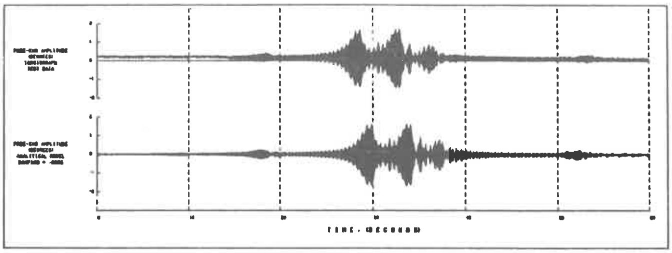

Figure 2: Comparison of test and analytical results for coastdown.

Good correlation was obtained using a low value of damping equal to 0.006. The calculated nominal torisonal stresses at each shaft secton are shown in Table 1. It is seen that the maximum stress levels during a coastdown are almost three times those attained during steady state operation.

The fast start transient was modeled and tested under two different conditions corresponding to the positions measured during the torsiograph test. The analysis was performed using the pressure-time data recorded during the testing in cylinder 8 LB. The pressure at other cylinders was obtained by interpolation form the recorded pressure-time trace. A conservative damping value of 0.6 percent of critical modal damping in each mode was used. A uniform time step of 0.00074 seconds was used for the numerical integration procedure.

Figure 3 shows the results for fast starts in which the crankshaft was initially positioned so that cylinder 3 RB was at BDC of its exhaust stroke.

It is seen that there is some variability between test measurements for the two starts in Figure 3. The response of this crankshaft under fast start conditions is sensitive to the initial crankshaft position. The simulated model does not check the fast start "A" or fast start "F" completely in every detail. When the T_n input to the STAMS is adjusted

Table 1: Nominal Torsional Stresses During Coastdown

Main Journal Location	Torsional Stress (psi)	
	Maximum	Minimum
Between Cylinder 1 and Cylinder 2	2350	-2340
Between Cylinder 2 and Cylinder 3	4310	-4520
Between Cylinder 3 and Cylinder 4	6150	-6070
Between Cylinder 4 and Cylinder 5	6310	-7370
Between Cylinder 5 and Cylinder 6	8650	-8350
Between Cylinder 6 and Cylinder 7	9900	-9630
Between Cylinder 7 and Cylinder 8	10980	-10970
Between Cylinder 8 and Cylinder 9	11470	-11720
Between Cylinder 9 and Cylinder 10	11660	-12060
Bewteen Cylinder 10 and Cylinder 11	11700	-11910

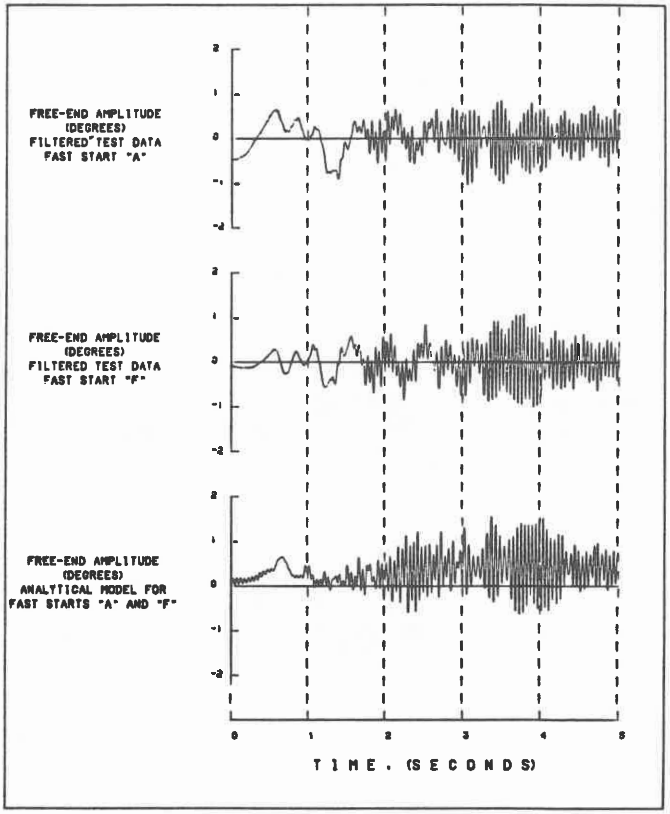

Figure 3: Comparison of test and analytical data for fast start.

to have the cylinder 8RB starting at BDC of its exhaust stroke, the simulation becomes very close to the processed acceleration test data for fast start "D" as shown in Figure 4. The two traces in the figure show excellent agreement with the analysis calculating a peak-to-peak vibration amplitude of 5.1 degrees and the accelerometer data showing 5.0 degrees. Analyses were performed varying the initial positon of the crankshaft section. The calculated nominal torsional stresses at each shaft section for the worst case postion (initial position that produces the highest stress levels), cylinder 10 RB at BDC exhaust stroke are shown in Table 2. It is seen that the stress levels during a fast start can reach up to five times those attained during steady-state operation.

Table 2: Nominal Torsional Stresses During the Worst Case Fast Start

Main Journal Location	Torsional Stress (psi)	
	Maximum	Minimum
Between Cylinder 1 and Cylinder 2	5730	-4050
Between Cylinder 2 and Cylinder 3	9410	-7630
Between Cylinder 3 and Cylinder 4	12430	-9380
Between Cylinder 4 and Cylinder 5	14400	-12220
Between Cylinder 5 and Cylinder 6	15670	-13160
Between Cylinder 6 and Cylinder 7	18870	-14220
Between Cylinder 7 and Cylinder 8	21570	-16730
Between Cylinder 8 and Cylinder 9	22830	-18540
Between Cylinder 9 and Cylinder 10	23170	-184200
Bewteen Cylinder 10 and Cylinder 11	23820	-16740

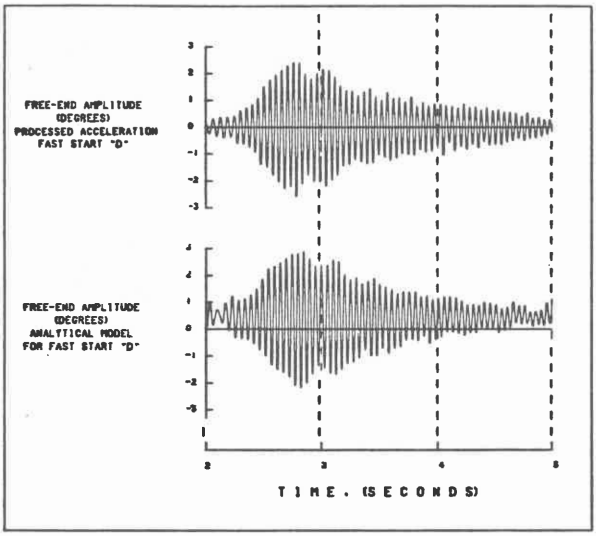

Figure 4: Comparison of test and analytical data for fast start.

Including the effect of stress concentrations which exist in fillet regions and oil holes on the crankshaft can increase the stress levels in the crankshaft to levels where crack initiation and propogation is of concern.

Utilization of Analysis Results — The results of STAMS yields time histories of the maximum and minimum values of vibrations and nominal shaft stresses. These results can be used for numerous applications including parametric studies varying engine and event parameters, and fatigue life predictions. A brief example of how the results of STAMS have been used in these applications follows.

Based on both analysis and testing, it was determined that the maximum amplitude of vibration in the crankshaft during engine start up was dependent upon the initial starting position of the crankshaft and startup duration. At San Onofre Nuclear Generating Station, the maximum amplitude of vibration occurs as the engine passes through three closely spaced critical speeds, the five-and-a-half order at 217 rpm, the fifth order at 240 rpm, and the four-and-a-half order at 264 rpm. These three closely spaced orders lead to the high stress levels observed during startup.

It is expected that the startup duration would have a significant effect on the amplitude of vibration, and two competing influences can easily be identified. First, in a fast start there is more cylinder pressure available to excite the crankshaft at the critical speeds compared with a slower start. Second however, the fast start engine spends less time in the speed range of these three criticals giving less time for the amplitude to build and producing fewer high stress cycles.

STAMS was used to perform analyses to study the impact of using slower engine starts (10 to 15 second range versus the 6.2 second fast start) [4]. It was found that for the worst initial position (that producing the highest stress levels), the maximum stress amplitudes decreased by approximately 10 percent. For the initial position producing the lowest stress amplitudes, this was not always the case with some of the slower startups producing best start stress levels higher that the best condition fast start stresses.

The influence of the initial starting position of the crankshaft is not so easily identified. Nevertheless, in a six-second fast start, the engine spends only a little over one half of a second in the speed range 216 rpm to 264 rpm. This provides only enough time for each cylinder to fire about once in this speed range. Thus, it seems likely that the order in which they fire during this critical time would be important. If starts are repeatable, the order of firing as the engine passes through these criticals is dependent on the initial starting position of the crankshaft. As the startup duration becomes longer, more and more cylinder firings occur in the speed range of the three criticals. It is therefore expected that the order of there firings, and hence the initial crankshaft postion, will become less important for longer startups.

STAMS was used to perform a parametric study to quantify the effect of initial crankshaft position on the response of the crankshaft. Ten fast start analyses were performed with the crankshaft's inital position altered by 72 degrees each time, thereby covering two revolutions (720 degrees) of the crankshaft. The maximum peak-to-peak vibration from each of the these conditions is shown in Figure 5. It is seen that the start with the lowest vibration occurs when cylinder 3 RB started at BDC during its exhaust stroke. This condition was measured twice. during the transient testing mentioned above. The highest level of vibration was obtained with cylinder 10 RB starting at BDC in its exhaust stroke. It is seen that the ratio of peak-to-peak vibration for the highest and lowest response is approximately 2.5 .

Based upon the operating history of a crankshaft (number of fast or slow starts, coastdown, and steady-state operation), the results of STAMS may be used for life predictions. The stress time histories provided by STAMS have been utilized to estimate crack growth rates using the BIGIF fracture mechanics code [5]. As input BIGIF requires the following; a geometric model of the region of interest, a stress time history (a stress block can be defined as a start-up and a coastdown), material properties, and stress distributions

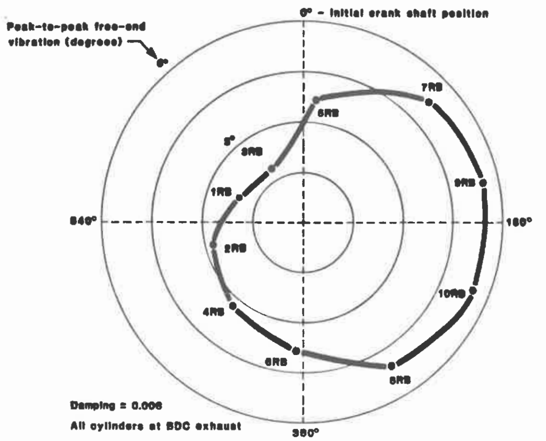

Figure 5: Effect of initial crankshaft position on free-end vibration during a fast start.

around the regions of interest (univariate and bivariate stress fields can be applied). An initial crack size, often the size that can be reliably detected during inspections, is also assumed.

To correlate with the engine history data available, crack propogation rates are determined as a function of the number of start-up and coastdown combinations. The crack growth propogation rates provide the information required to estimate life predictions of the component being analyzed.

CONCLUSIONS

A technique utilizing modal superposition has been developed for the analyses of torsional vibrations under transient conditions. Computer program STAMS was found to accurately predict the dynamic torsional response of crankshafts during transient conditions which include startups, coastdowns, and step changes in load. The results obtained using STAMS may be used for numerous applications which include parametric studies of various engine and analysis parameters and for life predicitons.

Analytical results from STAMS have been compared with test data obtained during testing of diesel generator engines. Good agreement was found between the analytical and test results.

REFERENCES

1. Weaver, W. Jr., and P.R. Johnston, *Structural Dynamics by Finite Elements*, Prentice-Hall Inc., 1987.
2. Johnston, P.R., and L.M. Shusto, "Analysis of Diesel Engine Crankshaft Torsional Vibrations," SAE Special Publication on Crankshafts.
3. " Evaluation of Transient Conditions on Emergency Diesel Generator Crankshafts at San Onofre Nuclear Generating Station," Failure Analysis Report Number FaAA-84-12-14, Revision 1.0.
4. " Impact of Slow Starts on the Fatigue Life of Emergency Diesel Generator DSRV-20 Crankshafts at San Onofre Nuclear Generating Station, Unit 1," Failure Analysis Associates Report Number FaAA-PA-R-86-07-21.
5. "BIGIF- Fracture Mechanics Code for STructures," Manuals 1-3, Failure Analysis Associates, April, 1981.