Analysis of Flexible Rotating Crankshaft with Flexible Engine Block Using MSC/NASTRAN and DADS

Lee S. Mayer
Computer Aided Design Incorporated
2651 Crosspark Road
Coralville, IA 52241

Hans Zeischka
Frank Maessen
Marc Scherens
Computer Aided Design Incorporated NV
Ottergemsesteenweg 439
9000 Gent
Belgium

ABSTRACT

Mechanical engineers most commonly predict stress and vibration of components within complete mechanical systems by the use of Finite Element Analysis (FEA) techniques. The accuracy of predictions depends mainly on applied boundary and loading conditions as well as meshing techniques. Experience has shown that discrepancies between numerical prediction and test data become great when one is dealing with dynamically loaded structures within mechanical systems that undergo large rigid body motion. Such systems typically exhibit geometric non-linearity and non-linear compliance between the different bodies. This publication presents the basic theory of flexible bodies in DADS and the application in the study of interaction between crankshaft and engine block for 4-cylinder and 4-stroke engines in unfired and fired conditions.

INTRODUCTION

Mechanical engineers most commonly predict stress and vibration of components within complete mechanical systems by the use of Finite Element Analysis (FEA) techniques. The accuracy of predictions depends mainly on applied boundary and loading conditions as well as meshing techniques. Experience has shown that discrepancies between numerical prediction and test data become great when one is dealing with dynamically loaded structures within mechanical systems that undergo large rigid body motion. Such systems typically exhibit geometric non-linearity and non-linear compliance between the different bodies. Solution through the use of non-linear FEA has improved the accuracy of results yet fails to provide a complete description of complex mechanical systems including hydrodynamics and large rigid body motion. This has led to the development of more rigorous analysis techniques to handle such problems.

Many rigid-body dynamics tools serve to provide loading conditions for components of mechanical systems under large range of motion, yet stresses observed through application of rigid analysis loads in FEA do not accurately represent dynamic stresses. Also it is evident that in a rigid body analysis one encounters problems in deriving valid vibration data for components and for the response of the complete system unless the flexibility of the rigid bodies is included.

Multi-body dynamics with flexible bodies combines the advantages of FEA, geometric non-linear FEA, and rigid body dynamics. The modal synthesis method for flexible bodies implemented into DADS superimposes flexible effects on rigid motion. This is combined with modeling of internal loading from hydrodynamic journals. The solution of the combined flexible system and hydrodynamics is presented in a study of a flexible rotating crankshaft and engine block, combined with modeling of hydrodynamic journals._Publications on the theory of Multi-body dynamics including the effects of flexible bodies based on the modal synthesis method appear in 1985 [1,2]. Several subsequent publications about its application to industrial problems are found. Multi-body dynamics with flexible bodies received proportionally more attention from engineers in the Aerospace industry [3,4,5] than from other industries.

One numerical solution for the interaction between crankshaft and engine block was published in reference [6] through the use of FEA techniques. Yet there are few if any publications that have been noted as of the writing of this paper related to the simulation of this interaction.

A first application of DADS for flexible engine analysis has been published in reference [7]. In this publication the crankshaft was represented by a few beam elements. The deficiencies of such a coarse representation motivated the authors in developing more exhaustive methods which can directly deal with refined meshed FE-models (10,000 or more nodes) that are frequently used for engine vibration and stress analysis. These more accurate models are included with rigid body dynamic effects and combined with a hydrodynamic journal internal reactions. The equations of motion are solved in the time domain and results are utilized to obtain the best possible representation of "dynamic stress".

This publication presents the basic theory of flexible bodies in DADS and the application in the study of interaction between crankshaft and engine block of 4-cylinder and 4-stroke engines in unfired and fired conditions. Results and Graphics presented provide simulation information of combined flexible effects with rigid body motion.

DADS BASIC THEORY

The implementation of flexible bodies in DADS is based on the assumption of small linear elastic displacements, which are described in a body-fixed coordinate system (X_{fe}, Y_{fe}, Z_{fe}). The position of this coordinate system in the inertial reference frame is given by vector **r** of Figure 1.

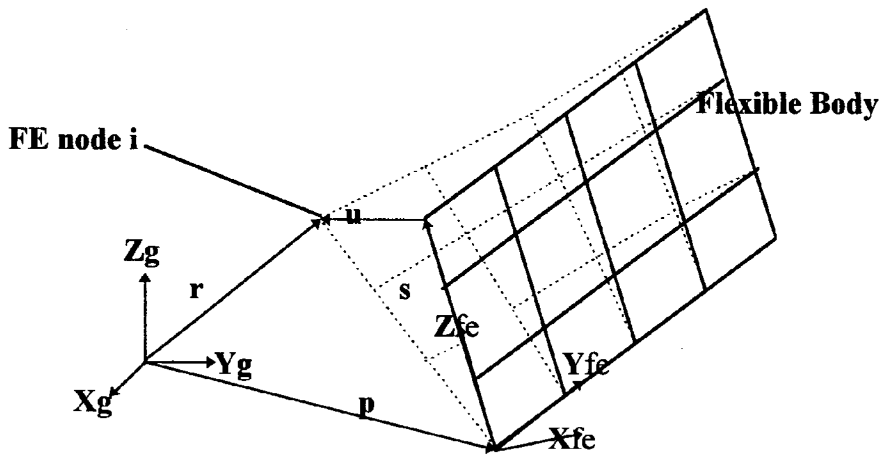

Figure 1. Reference Coordinate System for DADS.

Vector **s** defines the undeformed position and **u** the linear elastic deformation of node **i** in the finite element coordinate system. The transformation matrix **A** transforms vector **s+u** into the inertial reference system (**Xg**, **Yg**, **Zg**). The position of node **i** in the inertial reference system follows from Figure 1.

$$\mathbf{r^i} = \mathbf{r} + \mathbf{A} (\mathbf{s} + \mathbf{u^i}) \qquad (1)$$

In the modal synthesis method, the elastic deformation **u** of a flexible body is described by a linear superposition of orthogonal mode shapes. Usually the mode shapes are derived by FEA from a static solution (2b) and/or eigenvalue extraction (2c).

$$\mathbf{u} = \mathbf{f}_{st}\, \mathbf{a}_{st} + \mathbf{f}_{dyn}\, \mathbf{a}_{dyn} = \quad \mathbf{fa} \qquad (2a)$$

$$\mathbf{K f}_{st} = \mathbf{F} \qquad (2b)$$

$$[\, \mathbf{K} - w^2 \mathbf{M} \,]\, \mathbf{f}_{dyn} = 0 \qquad (2c)$$

The mode shapes can be partitioned into translational and rotational components ($\mathbf{f}_t$ and $\mathbf{f}_r$). The corresponding accelerations are obtained from differentiation with respect to time. The variational equations of motion for a flexible body with **n** nodes is given by equation 3.

$$\left[\delta r^T, \delta \pi^T, \delta a^T\right] \left\{ M \begin{bmatrix} \ddot{r} \\ \dot{\omega}' \\ \ddot{a} \end{bmatrix} + S(\omega', \dot{a}) + U(a) - Q \right\} = 0 \quad (3)$$

In this article we provide only the detailed description of matrix **M** of equation 3. In equation 4 the summation is over all nodes. The left upper partition contains the translational and rotational components which represent the inertia tensor of each body. Matrix **S** in equation 3 is of a more complex nature and contains all quadratic velocity terms.

$$M = \begin{bmatrix} \sum_{i=1}^{n} m_i & -A\sum_{i=1}^{n} m_i \tilde{s}'^i & A\sum_{i=1}^{n} m_i \Phi_t'^i \\ & \sum_{i=1}^{n} (m_i \tilde{s}'^i \tilde{s}'^i - J'^i) & \sum_{i=1}^{n} (m_i \tilde{s}'^i \phi_t'^i + J'^i \phi_r'^i) \\ & & \sum_{i=1}^{n} (m_i \phi_t'^{i^T} \phi_t'^i + \phi_r'^{i^T} J_r'^i \phi_r'^i) \end{bmatrix} \quad (4)$$

The formulation of a constrained mechanical system with flexible bodies is given by equation 5 (see reference [1,2] and [8]) where the Lagrangian multipliers **l** are used for solving the mechanical system with kinematic constraints in a combined Differential Algebraic Equation (DAE).

$$\begin{bmatrix} Mr & Mrm & \\ Mmr & m & \phi_q^T \\ \phi q & & 0 \end{bmatrix} \begin{bmatrix} \ddot{q}r \\ \ddot{a} \\ \lambda \end{bmatrix} = \begin{bmatrix} F \\ f \\ \nu \end{bmatrix} \quad (5)$$

$m = \phi^T M \phi$ M : lumped mass matrix

$k = \phi^T K \phi$ K : stiffness matrix

$f = \phi^T F_{ex} + \phi^T F_{in} - ka - ca$

In equation 5, $\mathbf{m}$ represents the modal mass matrix of flexible bodies. Vector $\mathbf{f}'$ contains the all inertia effects acting on the distributed mass of the flexible bodies, contributions of modal damping and stiffness and external forces. One factor for the numerical efficiency of the modal synthesis method follows from equation 5, as the rank of $\mathbf{m}$ is dependent only on the number of mode shapes included in analysis, not on the number of nodes in the finite element model.

MODEL DESCRIPTION AND ANALYSIS METHOD

The FEA models of a crankshaft and engine block for a four cylinder engine are depicted in Figure 2. This engine has been intentionally analyzed without the oil pan and cylinder head. The engine block is connected to the ground at its supports points. A rigid flywheel is mounted by a weld joint in DADS to the crankshaft. The flywheel is connected by a rotational spring to a body, which is constrained to rotate with a constant number of revolutions. The connecting rods and pistons are included as rigid bodies. The remainder of the system kinematics is modeled with mechanical joints defined in DADS between the different bodies of the engine.

The hydrodynamic journal effects are implemented by an impedance method [7]. The impedance charts provide the journal forces as a function of journal parameters such as dimension, oil viscosity, eccentricity and relative velocity. A detailed discussion of this implementation is provided in reference [7]. The parameters for each journal are provided by reference nodes in the DADS model for the crankshaft and engine block.

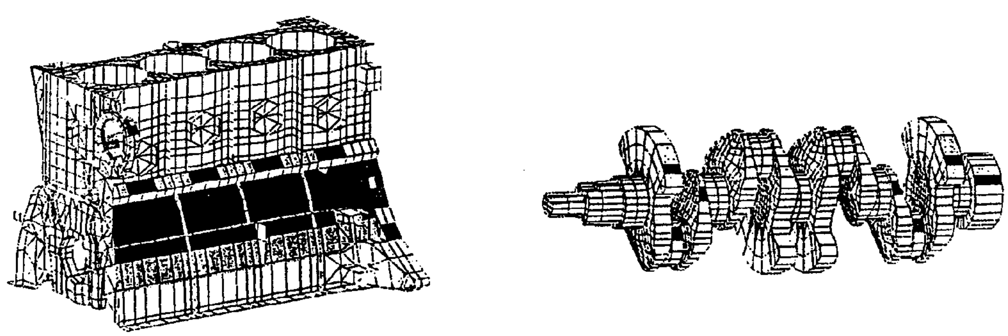

Figure 2. FEA Model Engine Block and Crankshaft.

The choice of mode shapes for the crankshaft has been verified by a correlation between DADS analysis and corresponding static FEA. For this purpose the crankshaft was supported to ground by springs representing equivalent journal stiffness. Different analyses with unit loads and arbitrary loading conditions induced at the position of the connecting rod showed negligibly small discrepancies between FEA and DADS analysis. The discrepancy can be explained by the fact that DADS executes by definition a geometric non-linear analysis, while a linear FEA was used for reference.

A typical deviation of the displacements of DADS from Static FEA is given in Table 1 for inertia loads applied to all journal points of the connecting rods.

Journal	1	2	3	4	5
Error in %	0.054	0.025	-0.007	-0.002	0.006

Table 1 Displacement Error Between DADS and Linear FEA.

The analysis procedure for DADS is shown on Figure 3. The rigid body model can be created by the DADS modeling system or imported by one of many CAD interfaces. FEA provides nodal information, a stiffness matrix, mass matrix, and mode shapes for each flexible body. A module of DADS called the Intermediate Processor (IP) reads this data from different supported FEA codes and prepares it for numerical simulation in DADS. During the simulation at every time step, the journal forces are calculated as a function of relative position and velocity of the reference nodes on the flexible engine block and crankshaft.

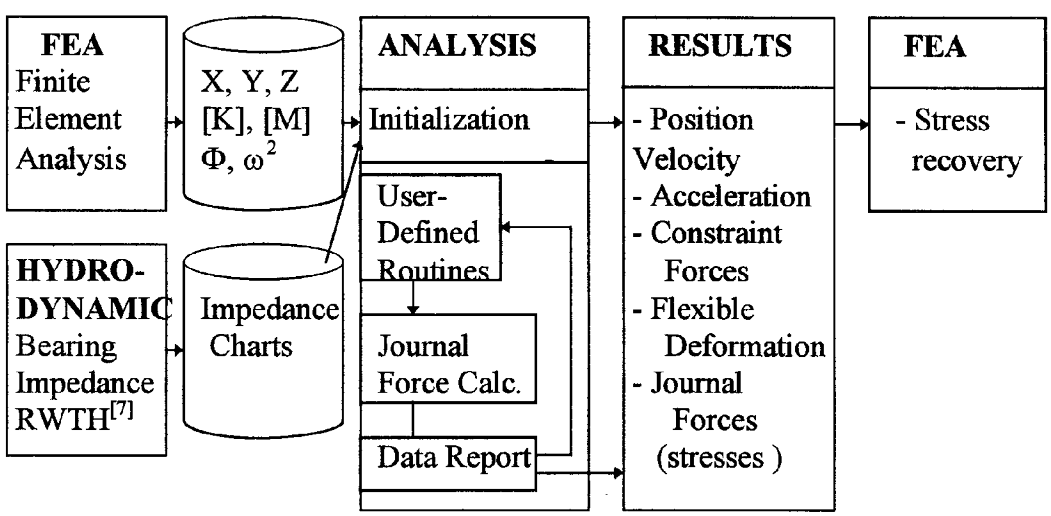

Figure 3. Analysis Flowchart.

Through solution of equations of motion, the position, velocity and acceleration of each body and reaction forces for mechanical constraints are reported to results files. For flexible bodies, DADS also provides the scale factors for the mode shapes at each time step as well as position, velocity and acceleration data for nodes of the included FEA models. These scale factors can then be utilized for efficient stress recovery through the finite element model.

RESULTS

The following results apply to an unfired engine as shown in Figure 2 at a crankshaft rotational speed of 4000 rotations per minute (rpm). The main journals are identified by 1 to 5 on depicted results , where journal 5 is close to the flywheel.

Figure 4 shows the vertical hydrodynamic forces (piston direction of motion) in all journals for four revolutions occurring between 0.0 and 0.06 seconds. One observes for example the influence of the flywheel at journal 5 (curve E) compared to the force in journal 1 (curve A). If the flywheel is omitted congruent loading conditions for journal 1 and 5 would be observed.

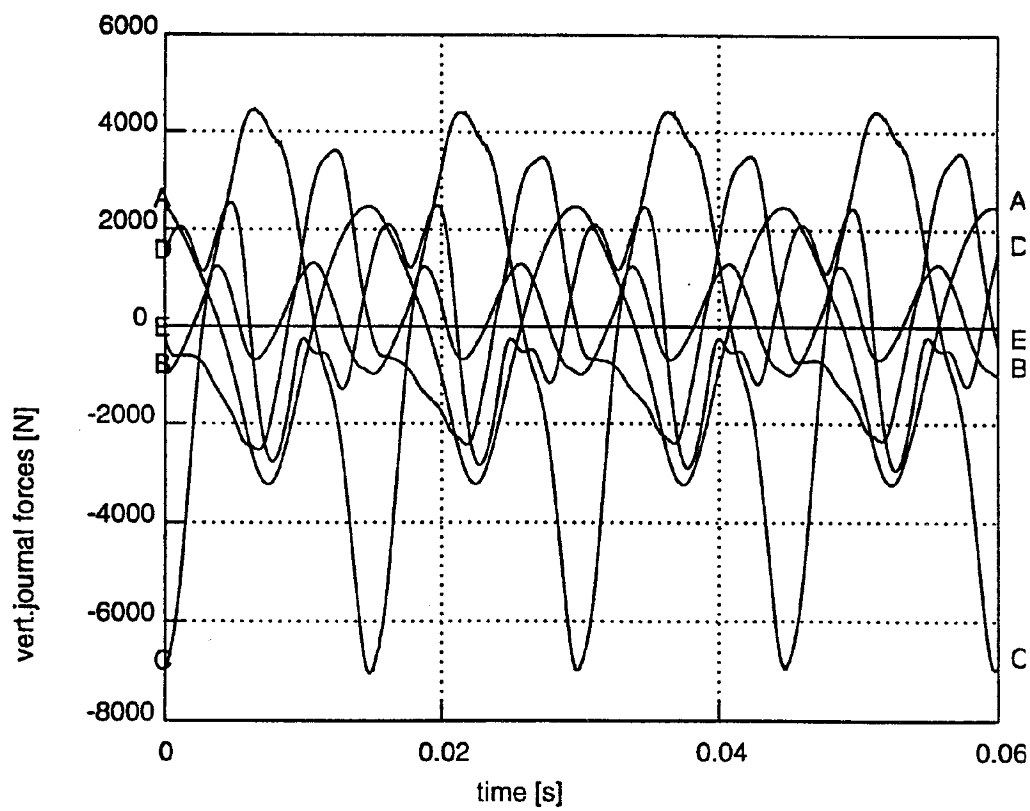

Figure 4. Vertical Journal Forces of an Unfired Engine.

Figure 5 shows the linear spectrum of the acceleration in the vertical direction of the reference node for journal 5. The acceleration of the corresponding node of the crankshaft (not shown) is similar. Note in Figure 5 a second and fourth order excitation is observed. These harmonics are not seen in the linear spectrum of the vertical journal force of Figure 6. Study of the frequencies of the system indicate that a resonance of the combined flexible system of engine block and crankshaft is being excited.

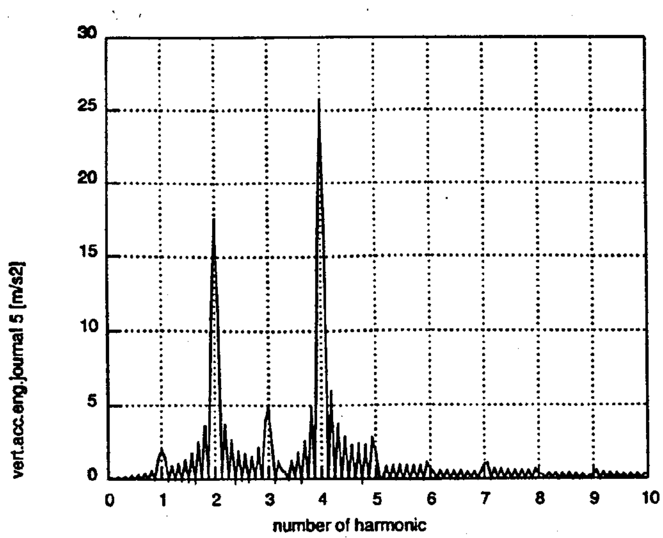

Figure 5. Spectrum of Vertical Acceleration in Journal 5.

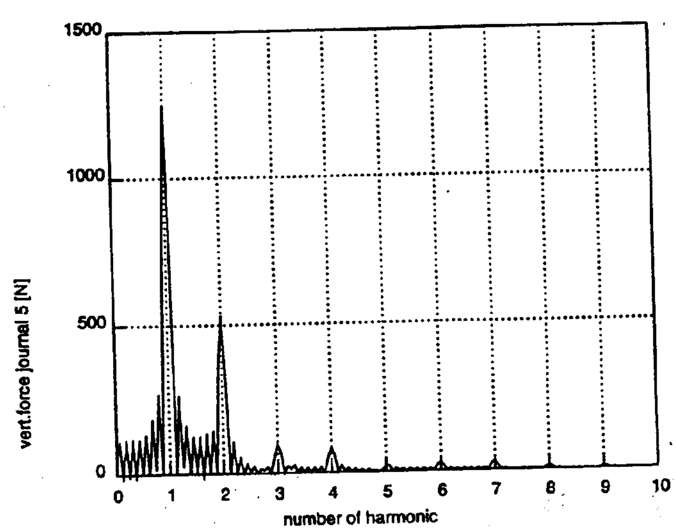

Figure 6. Spectrum of Vertical Force in Journal 5.

The relative motion of pins at the crankshaft center with to respective reference points on the engine block is shown in Figure 7 for journal 1 (curve A) to journal 5 (curve E) with intermediate journals shown (curve B, C, and D).

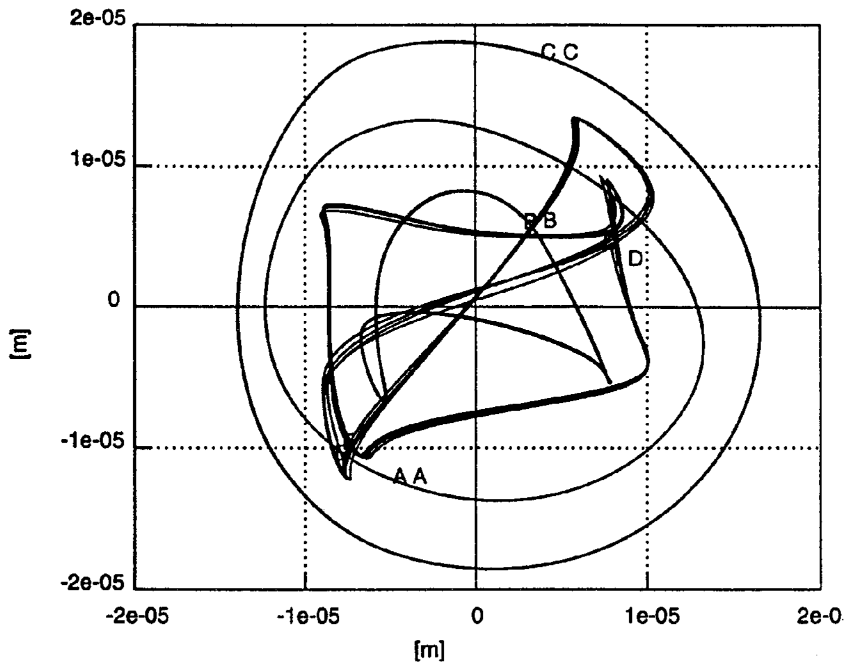

Figure 7. Polar Plots of Crankshaft to Engine Block Relative Motion.

The vertical journal forces for a separate fired engine simulation are depicted in Figure 8. One will observe nicely the effects of ignition compared to results of an unfired engine shown on Figure 4. Again, curve A corresponds to journal 1 and D to journal 5.

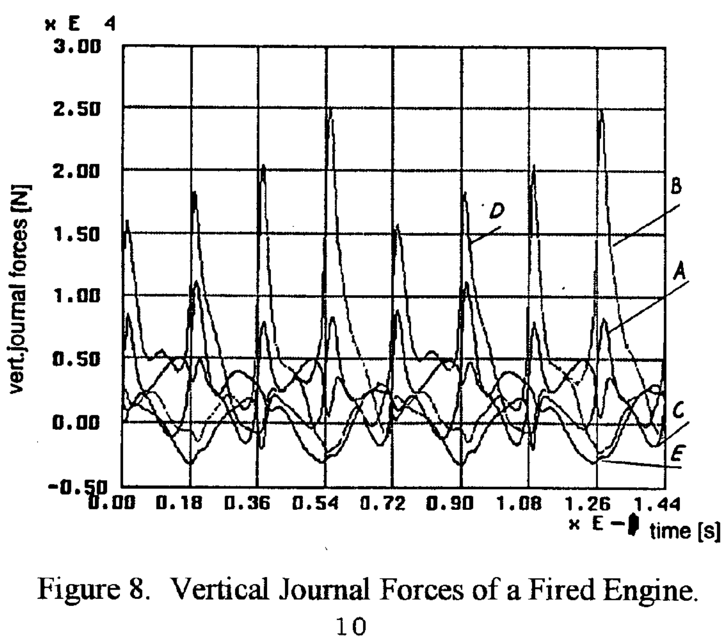

Figure 8. Vertical Journal Forces of a Fired Engine.

642

Analyses were executed for different sets of mode shapes for the engine block. Figure 9 shows for example the vertical force for journal 5 for 10 static modes (curve A), for 16 Eigenmodes (curve B) and for 10 static plus 10 normal modes (curve C). The 10 static modes correspond with the horizontal and vertical forces applied at the journals. The first sixteen Eigenmodes of the supported engine block were in the frequency range between 250 and 2000 Hz. One observes almost no difference between curve A and C, while curve B shows significantly different behavior.

It is expected that the results of curve C converge to the "exact" solution. Figure 9 shows clearly that a truncated set of dynamic modes only does not result in an accurate description of this mechanical system. In this case the truncation started from the 30th harmonic. So the result of Figure 9 indicates the need to include static modes for accurate dynamic analysis of combustion engines. This has been observed in the simulation of many other structures where it is suggested that static modes combined with fixed boundary condition normal modes provide the best approximate dynamic solutions [9]. It is noted here, that static modes can be combined with Eigenmodes in a DADS analysis as shown in equation 5.

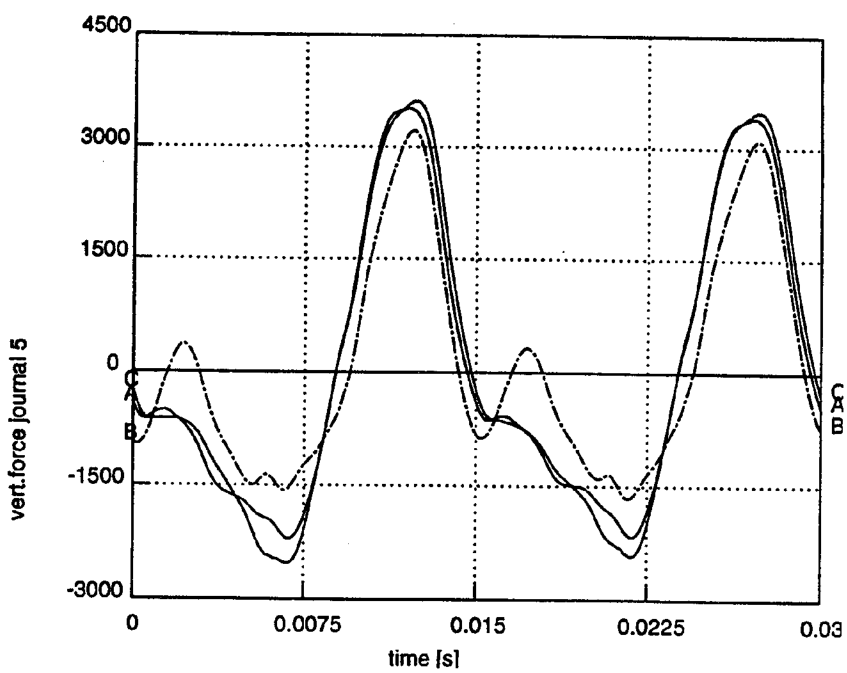

Figure 9. Vertical Journal Force for Journal 5 at modes=10 static (A), 16 Eigenmodes (B), and 10 static + 10 Eigenmodes (C).

The ability to animate flexible bodies in DADS proved to be useful in visualizing the dynamics of such a complex flexible systems. Figure 10 shows the deformation of a rotating crankshaft during a flexible crankshaft and engine block simulation.

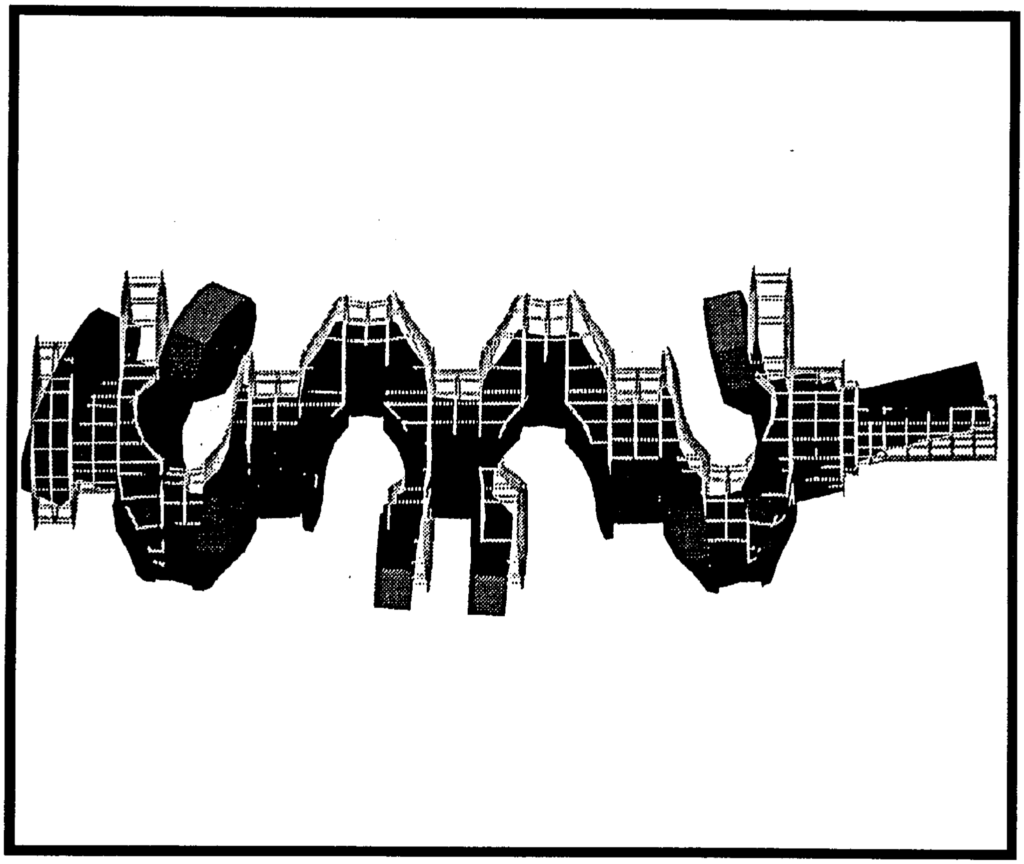

Figure 10. Deformation of A Rotating Crankshaft During DADS Simulation.

CONCLUSIONS

A description of Multi-body dynamics theory as offered by DADS has been presented. The successful application of DADS utilizing MSC/NASTRAN generated flexible models has been demonstrated for the analysis of the interaction between rotating crankshaft and engine block studies of journal forces. This problem shows non-linear behavior by kinematic motion and journal compliance.

The efficiency of DADS coupled with MSC/NASTRAN is proven by application of this study on typical workstations. It has also been shown that limitation to a truncated set of normal modes may not be sufficient in describing accurately the dynamic response of an engine block.

The results of real-life applications demonstrate that DADS and MSC/NASTRAN deliver all required data for designers of combustion engines through one single simulation model. This complete simulation model offers information for stress and vibration prediction to a high degree of accuracy. The frequency resolution is such that vibration results are valid as input for acoustic radiation analysis.

Research and development are proceeding with enhancements on the current solution method. The enhancements include representation of local journal flexibility and full three dimensional solution of Reynolds equation for hydrodynamic journal modeling.

ACKNOWLEDGMENTS

The authors would like to thank Mr. Mark Gwillim of MSC support in Dallas, Texas, and Joe Brackin of MSC support in Atlanta, Georgia for timely and expert assistance with questions regarding the integration of MSC/NASTRAN and DADS.

REFERENCES

[1] W. S. Yoo, E. J. Haug, "Dynamics of Flexible Mechanical Systems using Vibration and Static Correction Modes," Technical report University of Iowa, May 1985.

[2] A. A. Shabana, "Substructure Synthesis Methods for Dynamic Analysis of Multi-body Systems," Computer & Structures, Vol. 2, pp. 737-744, 1985.

[3] N. Vukasovic, J.T. Celigüeta, J. Garcia de Jalon, E. Bayo, "Flexible Multi-body Dynamics Based on a Fully Cartesian System Support Coordinates," Proceedings of Dynamics of Flexible Structures in Space, Cranfield, U.K., 15-18 May 1990.

[4] C. Garcia Marirrodriga, J. Zeischka, E. C. Boslooper, "Numerical Simulation of Ulysses Nutation," Fifth European Space Mechanisms and Tribology Symposium, ESTEC, Noordwijk, The Netherlands, 28-30 October 1992.

[5] O. Friberg, V. Karhu, "Simulation of Deploying Flexible Satellite Booms Using DADS and ABAQUS," Department of Structural Mechanics., Chalmers University of Technology, Göteborg, Sweden, Copyright 1991.

[6] D. J. Nefske and S. H. Sung, "Coupled Vibration Response to the Engine Crank-Block System," Engineering Mechanics Department, General Motors Research Laboratories, Warren, Michigan., 1994.

[7] G. Knoll, H. Peeken, PhD. Troppmann, J. Zeischka, F. Maessen, "Simulation und Dynamik Elastischer Kurbelwellen unter Berücksichtigung der Grundlagersteifigkeit sowie der Hydrodynamischen Schmierfilmreaktionen," 6th Internationaler Kongress: Berechnung im Automobilbau, 21.-23 September, 1992, in Würzburg.

[8] E. J. Haug, "Computer Aided Kinematics and Dynamics of Mechanical Systems," University of Iowa, 1989.

[9] Sung-Soo Kim, E. J. Haug, "Selection of Deformation Modes for Flexible Multi-body Dynamics," Center for Computer Aided Engineering, College of Engineering, The University of Iowa, Technical Report 86-22, U. S. Army Tank-Automotive Command Research and Develompent Grant #DAAE07-85-C-R046, 1986.