A FLEXIBLE MULTIBODY SYSTEM APPROACH TO PISTON-LINER INTERACTION DYNAMICS.

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ABSTRACT

Piston slap is considered to be one of the major sources of noise in four-cycle diesel engines. The reduction of piston slap is essential for total engine noise reduction. Furthermore, the cylinder liner vibrations due to piston slap may lead to coolant cavitation resulting in severe material damage.

The purpose of this study is to present a new modal flexibility capability of ADAMS as a tool for analyzing the secondary motion of a piston and the resulting cylinder liner vibration. By modeling the cylinder liner as a flexible body and performing a dynamic simulation of a piston traveling inside the cylinder, the vibrations of the liner due to piston impact may be computed.

The paper describes briefly the process of importing a flexible body from a MSC/NASTRAN finite element model to ADAMS and presents some approaches to ADAMS model verification. The paper discusses some modeling issues such as modal truncation, component mode synthesis and force distribution. Finally, it is demonstrated how the surface motion of the cylinder liner might be communicated back to the finite element program for stress, sound or structure-fluid interaction analysis.

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INTRODUCTION

From its inception, ADAMS has provided its users with a limited ability to model structural flexibility through a variety of force elements. The introduction of the NFORCE provided the ability to capture the influence coefficient relationship of the FEM stiffness matrix. The common deficiency of each of these force approaches was a poor, lumped mass discretization of flexible components. A recently developed functionality in ADAMS makes it possible to model complex flexible components using component mode synthesis, a method that preserves all the condensed mass properties of the finite element model.The user construct an Finite Element model of the flexible component in a Finite Element program and exports component modes for the flexible component. The ADAMS user selects the appropriate modal content for his flexible multibody dynamics analysis. Flexible multibody dynamics using component mode synthesis blurrs the lines between the Finite Element Method and Nonlinear Multibody Dynamics taking advantage of the important contributions of both disciplines.

An interesting example involving both linear vibrations and non linear motions is the piston liner interaction dynamics problem, commonly referred to as piston slap. The liner vibrations are excited by the transverse motion of the piston across the bore as the piston impacts the cylinder wall. Piston slap is known to be a major source of noise in a diesel engine. In addition to cause noiseproblem the liner vibrations induces pressure fluctuations in the coolant jacket. In a diesel engine the rapid drop in the local pressure, which leads to bubble formations, might lead to cavitation in the coolant jacket.

Several studies regarding the piston slap problem have been conducted. Usually the researcher separates the problem into two parts; analysis of the secondary motion of the piston generates the load case for the cylinder liner vibration analysis. A couple of studies have involved very complex models with flexible piston and cylinder liner. This paper describes an approach to investigate the secondary motion of the piston and the cylinder liner vibrations simultaneously using ADAMS. This integrated approach makes it possible to predict more accurate loads, making it possible to parameterize the simulation model for design studies, e.g., optimizing the wrist-pin location. Although this analysis capability was created for use in the design phase to evaluate noise and cavitation performance of different designs an enhanced version including, e.g., flexible piston, friction, tribological effects etc. could be used to analyze a complete design in more detail. The current model is simple enough to provide numerical stability and also require a minimum of input from the user.

TECHNICAL BACKGROUND

Modal flexibility in ADAMS was first introduced in the 8.1 release. Chief among the limitations of this first release was the lack of component mode synthesis. The deformations of a flexible component could only be described by its normal modes placing a burden on the analyst to properly select boundary conditions for the Finite Element normal mode analysis. Version 8.2, released later the same year, alleviated this difficulty by supporting the Craig-Bampton approach to component mode synthesis. The Craig-Bampton method defines an interface through which structural interactions with the flexible component are to take place. Deformations are modeled as a linear combination of fixed interface normal modes and constraint modes. The fixed interface normal modes are obtained by a normal mode analysis during which all interface degrees of freedom are fixed. The constraint modes are static shapes obtained by applying a unit deflection to one interface degree of freedom while fixing all other interface degrees of freedom.

One property of the Craig-Bampton method makes it inadequate for direct application to nonlinear multibody dynamics. Rigid body motion of a component modeled with the Craig-Bampton method is expressed as a linear combination of constraint modes. The large, displacements of nonlinear dynamics cannot be modeled this way. Hence, the rigid body motion contained in the Craig-Bampton basis must be isolated, rejected and replaced with nonlinear rigid body coordinates. This is accomplished by solving an eigenvalue problem for the flexible component w.r.t. its Craig-Bamton basis, effectively replacing the basis of fixed interface normal modes and constraint modes with an orthogonal basis of eigenvectors. This new basis consists of shapes that approximate the normal modes of an unconstrained system in addition to normal mode shapes corresponding to the component boundary. A beneficial side effect of this orthogonalization is the identification of natural frequency associate with each of the basis vectors. This gives the user insight into the frequency content contributed by the constraint modes and gives him the opportunity to eliminate undesirable high frequency motion in the component boundary. It should be understood that truncating boundary modes, even after orthogonalization, is tantamount to applying a constraint to the system.

THE TRANSLATION PROCESS

The task of obtaining the component modes is made simple by the use of a custom DMAP alter sequence[1] for MSC/NASTRAN. The user must simply model his flexible component as a MSC/NASTRAN super element and identify which grid points should belong to the interface. For additional user control is possible, for instance selecting only a subset of interface grid DOFs. MSC/NASTRAN writes a punch file containing grid locations and connectivity, grid mass and inertia, and modal information. The punch file is translated to the MDI Modal Neutral File format and is ready for inclusion in an ADAMS model.

SIMULATION MODEL

Flexible Cylinder Liner

There were two primary considerations when preparing the Finite Element model of the cylinder liner. As the cylinder liner is fixed to the block at one end its boundary conditions in ADAMS may be idealized by fixing the cylinder rim to ground. This is best accomplished by rigidly connecting the grid points on the rim to a central point using a MSC/NASTRAN RBAR element and applying a single point constraint (SPC). A single ADAMS FIXED JOINT attaching the central point to GROUND in the ADAMS model eliminates all the rim degrees of freedom. The other modeling consideration was the selection of the flexible body interface. The piston's transverse excursions in the bore are perpendicular to the crank shaft causing it to impact the cylinder liner at two lines of grid points. Only the transverse translations of all the grid points along these two lines of contact are promoted to the flexible body interface. MSC/NASTRAN generates constraint modes for each degree of freedom of the interface, allowing the modeling of behavior in the neighborhood of the violent piston to cylinder interaction. The fixed interface normal modes provide additional information about the dynamics of the cylinder to the desired level of detail. In this case 20 fix interface boundary normal modes where extracted from NASTRAN.

Rigid Modeling Elements

The ADAMS model includes three rigid bodies: crankshaft, connecting-rod and piston. These bodies are connected with joints according to picture below. A constant engine speed constraint is applied at the crankshaft bearing with a MOTION statement. The o-ring support of the bottom of the cylinder liner was emulated using spring dampers. The combustion force, modeled with an ACTION ONLY SFORCE, is applied at the piston crown. The combustion force is computed based on the piston crown area and combustion pressure. The interaction between the piston and the cylinder liner is modeled with VFORCE elements and IMPACT functions. The force elements are applied between nodes on the flexible cylinder liner and the piston corners. The piston is assumed to come into contact with the cylinder liner at four points only, the points furthers from the block centerline at the piston skirt and the crown. The contact force is distributed to neighbouring nodes using ADAMS HAVERSINE functions.

[1] Interfaces to other Finite Element programs are, or soon will be, available.

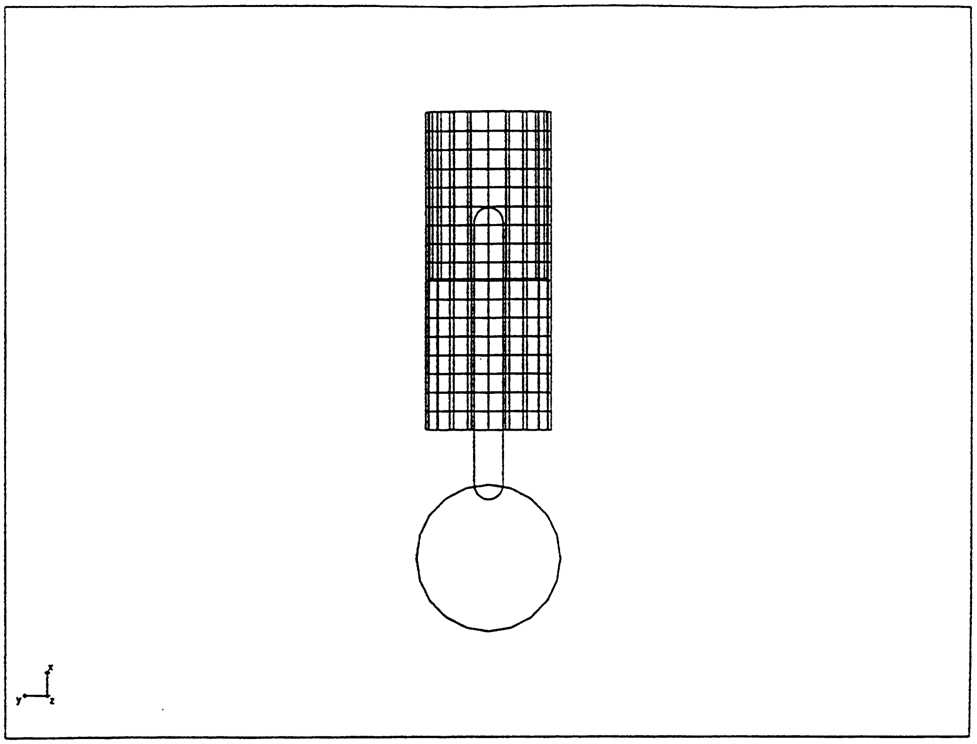

Figure showing the ADAMS model.

RESULTS

In order to eliminate start-up transients, a dynamic simulation of four engine revolutions (two cycles) was performed. This simulation of the model which contained 1301 dynamical equations took 313 CPU minutes on a 175 MHz Silicon Graphics Indy workstation.

Piston Secondary Motion

Ideally the piston motion is a crank mechanism with one degree of freedom. Due to the play between the piston and the cylinder liner a secondary transverse motion is overlapped the primary motion. The piston slap forces is generated due to the play between the piston and the cylinder liner. The rotation of the connecting rod causes a resulting transverse and combustion force to move the piston from side to side several times per cycle. In the combustion engine the transverse force caused by combustion and internal forces by impact forces occurring when the piston hit the cylinder wall. The transverse force therefore contains both high and low frequency components. Friction and tribological effects also have a significant influence on the transverse force but have been omitted from this study for the sake of simplicity.

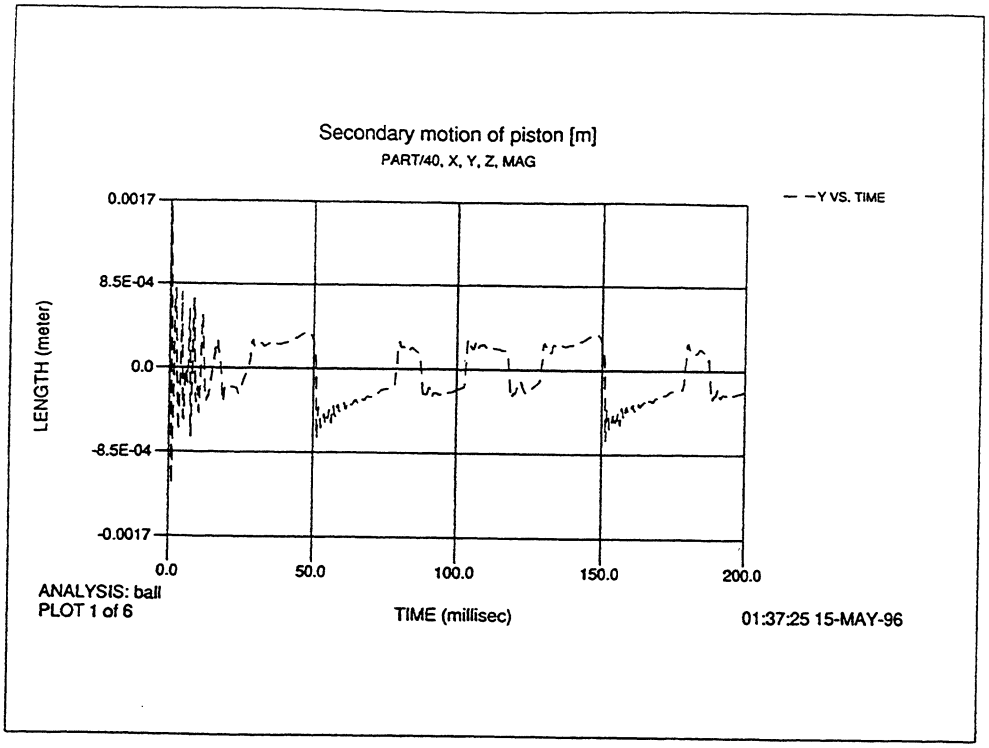

Figure showing the transverse motion of the piston

Cylinder Liner Vibrations

The impact forces between the piston and the cylinder liner cause the liner to vibrate, to varying degree, in all of its modes of vibration.

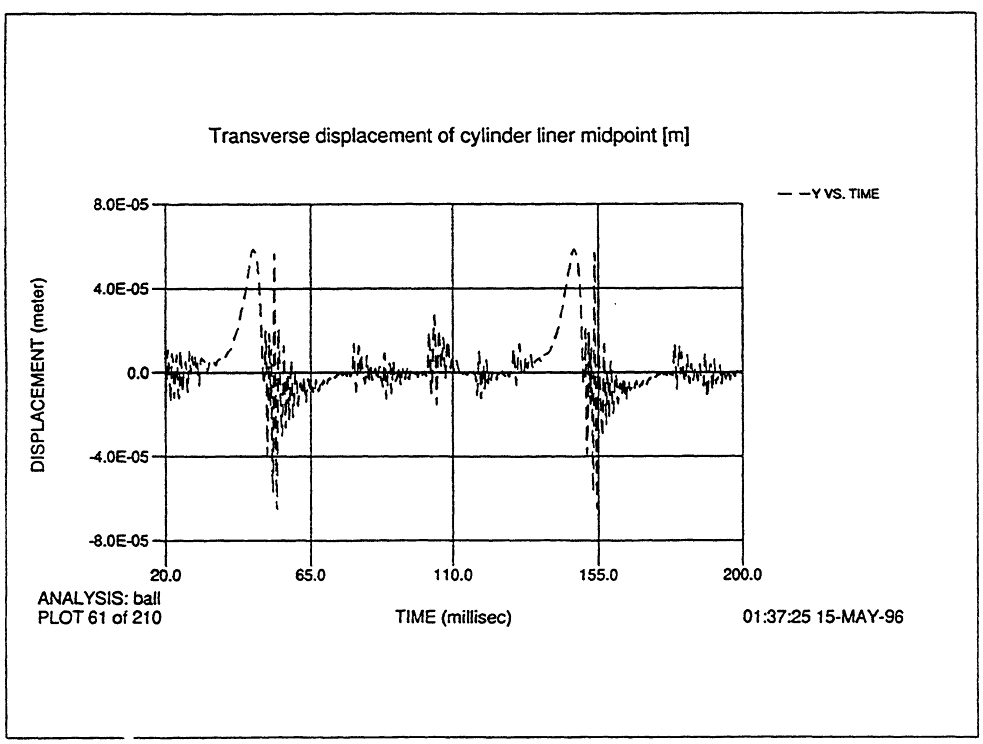

Figure showing the transverse motion of the liner

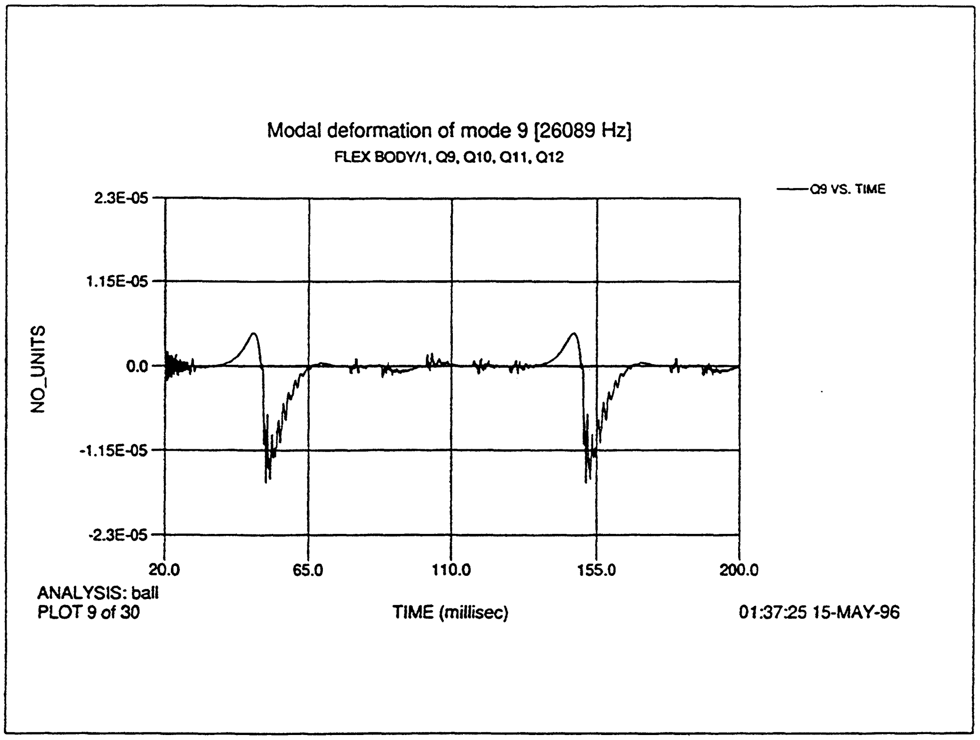

Figure showing a typical modal coordinate

FLEXIBLE ELEMENT VALIDATION

In order to verify adequate frequency behavior of the flexible cylinder liner eigenfrequencies and mode shapes computed with ADAMS and IDEAS were compared. See Table 1. Results obtained from ADAMS and IDEAS correlated very well. Good agreement between ADAMS/Linear and IDEAS results indicates that a reasonable number of modes were selected as active in ADAMS/View. No further validation of the results presented herein was performed.

1	738.7693	737.5008
2	738.7695	737.9133
3	1074.791	1066.774
4	1074.802	1067.680
5	1822.568	1812.075
6	1822.578	1825.850
7	1834.646	1834.930
8	2663.482	2660.928
9	2663.482	2689.819
10	2936.725	2861.512
11	2943.628	2942.885
12	2947.717	2982.869
13	3196.596	3125.211
14	3205.89	3326.323
15	3443.996	3420.048
16	3444.031	3479.939
17	3923.998	3854.963
18	3929.112	3977.028
19	5097.171	5017.661
20	5097.62	5141.941

Table 1: Eigenfrequencies computed with ADAMS/Linear and IDEAS.

FUTURE DEVELOPMENT

Further development of this model should include tuning of impact force parameters, incorporation of friction, oil film forces and a flexible piston to name but a few. Additional model enhancements could include more realistic geometry, measured torque resistance on crank shaft instead of constant velocity constraint, distribute forces between nodes more accurately.

Future development of ADAMS capabilities will include enhanced postprocessing capabilities to facilitate two way communication with a Finite Element program. This will allow further investigation of cylinder liner stresses, noise and cavitation analysis.

CONCLUSIONS

A approach to piston slap analysis using ADAMS/FEA with modal flexibility has been presented. It has been demonstrated that this approach is highly feasible.

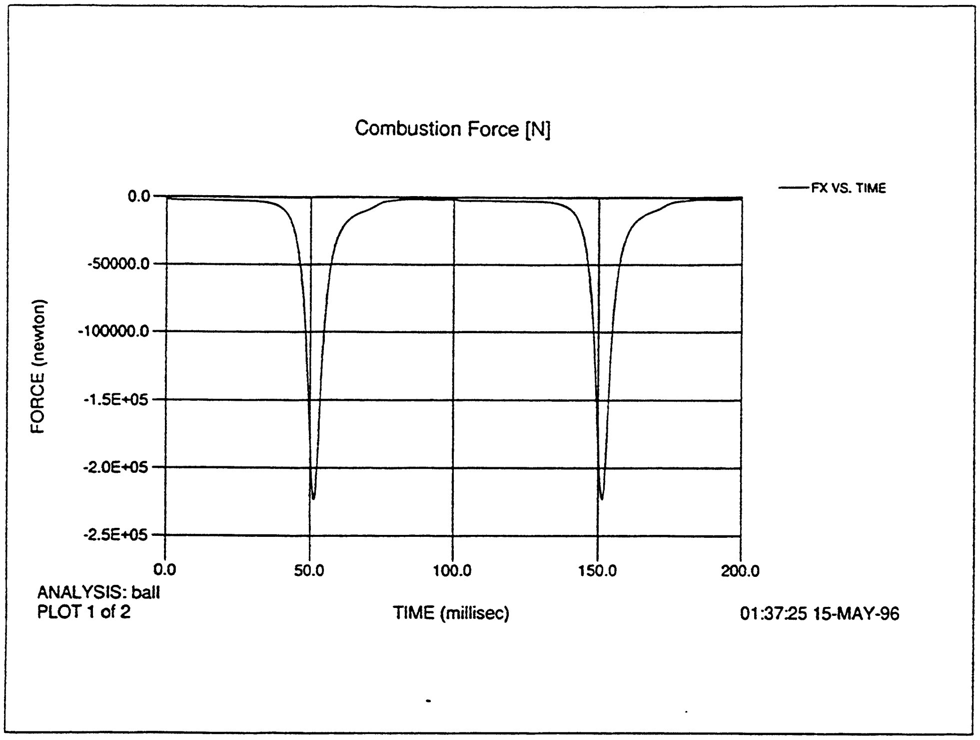

Figure showing the Combustion Force [N]

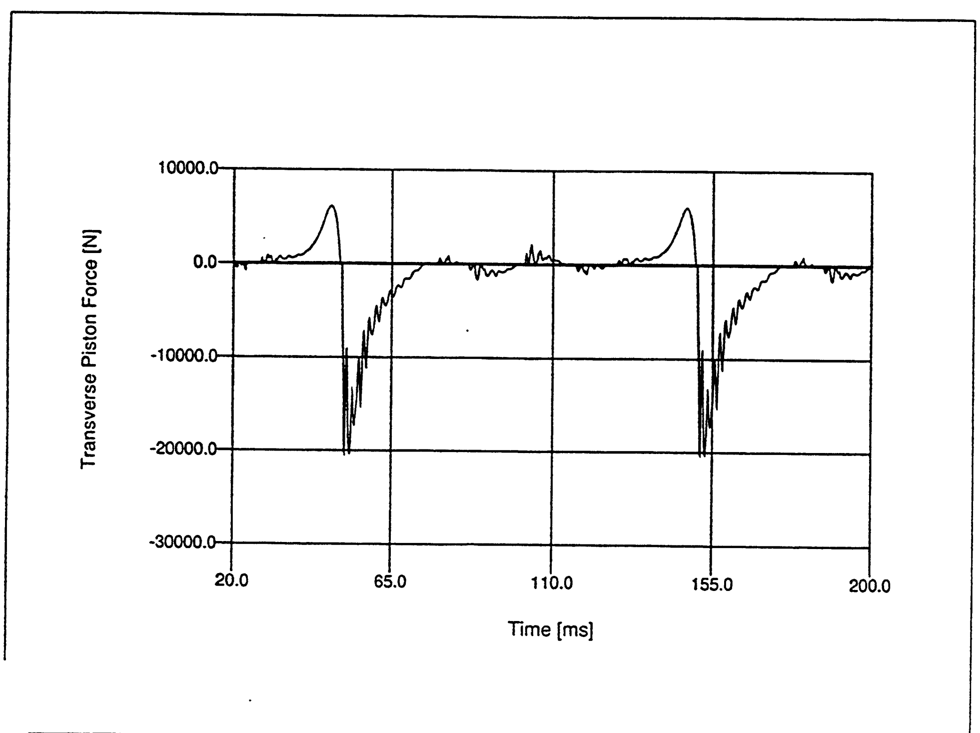

Figure showing Transverse Force [N]