A Piston Secondary Motion Model Including Structural Flexibility

Mark Tussing
Cummins Engine Company

Diego Minen
Mechanical Dynamics Inc.

ABSTRACT

Liner cavitation can be a durability issue on diesel engines with wet cylinder liners. Testing and analysis at Cummins has shown that the basic design of the power cylinder components (liner and piston) can have a significant influence on whether cavitation is encountered in an engine application. In order to develop and design power cylinder systems free of cavitation problems, a piston secondary motion model that includes liner flexibility is being developed. This model is being developed in the ADAMS multibody system simulation (MSS) package. The model will include an ANSYS finite element representation of the liner, enabling the model to be used to predict liner surface velocity, which is directly correlated to cavitation. This paper documents the development of this model, and the calibration and correlation of elements of the model to experimental data.

INTRODUCTION

Liner cavitation is the process by which local pressure decreases in the coolant around the cylinder liner causes vaporization of the fluid, and creates cavitation bubbles which then implode as pressure increases above vapor pressure. When these bubbles collapse, they cause mechanical damage in the form of erosion to the outer surface of the cylinder liner. The pressure fluctuation in the fluid is a direct result of the motion of the cylinder liner. Coolant chemistry, pressure and temperature are all significant factors influencing cavitation, but variations in engine application and service practices make strict control of these factors by an engine manufacturer unlikely. Liner motion is a result of the piston secondary motion, which is largely controlled by the design of the engine and it's components.

To effectively predict whether cavitation will occur for a specific engine design, the secondary motion of the piston and the response of the liner are needed. Traditional piston secondary motion models do not predict the liner response to the piston motion. Instead, most treat the liner as rigid, and use some measure of piston-to-liner impact force as an indicator for cavitation. The model being developed here adds the complexity of a flexible liner to the secondary motion model, and thus allows prediction of liner velocity. Unlike piston impact force, liner velocity can and has been directly measured in an engine [1]. This means that such a model can actually be calibrated to measured engine operating data, providing a validated predictive tool.

1 MODEL ELEMENTS

The model includes representations for the piston, connecting rod, crankshaft, cylinder liner and crevice seal. The representation for the liner interface with the cylinder head, block and combustion seal are modelled in ANSYS as simple constraints. A proprietary combustion model was included as an ADAMS subroutine to provide forcing due to cylinder pressure on the piston. The crankshaft is constrained by a motion set to the desired engine RPM. The following sections give details on each of the model elements.

1.1 Cylinder Liner Representation

The objective for this work was to develop a model of the liner and its constraints that matched some measurable response of a real cylinder liner as constrained in an engine. The constraints on the liner are due to the clamping of the top-stop and the crevice seal around the bottom of the liner. Figure 1.1 shows a cross section of the liner with these constraints indicated. In order for the model to work correctly with the piston secondary motion model, this representation of the constrained liner needs to exist as part of an ADAMS MSS. ADAMS is capable of representing flexible bodies as a series of modes developed from a modal analysis on an FE model. This technique is generally known as the modal flexibility method.

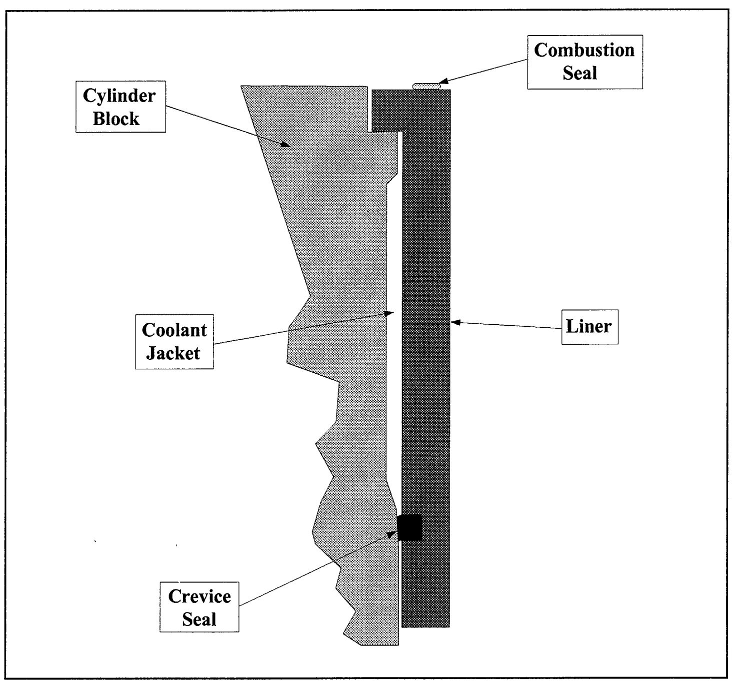

Figure 1.1 A cross section of the cylinder liner and block showing the constraints.

Figure 1.2 is a diagram that shows how the different parts of this model were developed and experimentally verified. The liner geometry was first modelled in ANSYS as an FE representation of the liner. A modal analysis of this model was conducted to determine the free-free, or unconstrained modes of the liner. To insure that the mesh density used was sufficient to capture the correct modal shapes and frequencies of the liner, the mesh was increased to about 3 times the density used in the final analysis. Modal frequencies changed less than 5% at this higher density, so the original density was used for the remainder of the analysis.

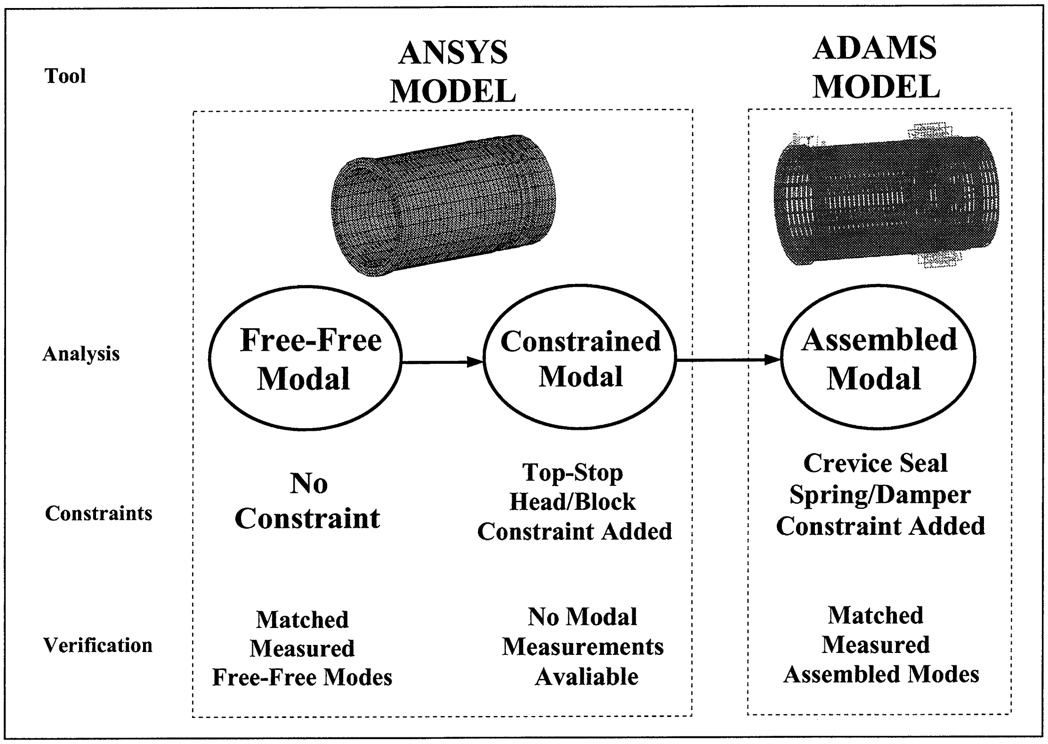

Figure 1.2 Overview of liner model development and verification.

The second step in the liner model development was to add a constraint to the liner to represent the clamping of the top stop between the block, cylinder head and combustion seal. This constraint was simulated by constraining three degrees of freedom for each of the nodes on the top surface of the liner where the liner comes into contact with the combustion seal. In addition the nodes around the outer edge of the top stop, where the liner contacts the block, were constrained in the same fashion. A modal run was then made with the specific commands in ANSYS to write the modal neutral file that is used by ADAMS to represent the flexible body.

The third step in the model development process was to import the modal representation of the liner into ADAMS as a flexible body and add constraints to represent the crevice seal. The crevice seal was modelled in ADAMS as spring-damper elements connected circumferentially between the nodes in the seal groove area of the liner to ground. Modelling of the seal in ADAMS is necessary due to the damping effect the seal has on the system dynamics. In addition, a fixed-joint, constraining 6 degrees of freedom, was attached between the liner and ground at one of the nodes previously constrained in the ANSYS model.

1.2 Crankshaft and Connecting Rod Representation

Both the crankshaft and connecting rod are currently represented as rigid bodies with mass and inertia. The crankshaft is constrained to ground with a revolute joint. The rod is also constrained to the crankshaft using a revolute. Friction and flexibility in these connections is currently neglected. The piston pin is not included as a separate part in the model. The mass and inertia of the pin is added to the piston and friction at the pin joint is applied at a revolute joint between the connecting rod and piston. Figure 1.3 shows the ADAMS model with the crank, rod and piston.

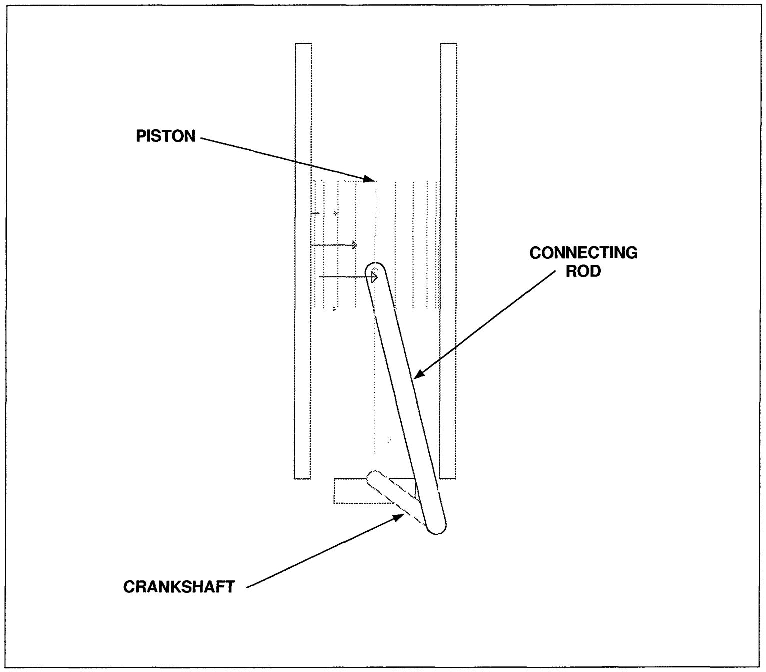

Figure 1.3 Assembled model with all components

1.3 Piston Representation

While the piston is modelled as a rigid body, flexibility of the piston is discretely accounted for in the contact between the piston and liner. Because the radial contact profile of the piston is not cylindrical but actually has a complex ovality, the contact stiffness between the piston and liner is not linear. In this model the piston is modelled with a mass and inertia at it's cg, but also with contact locations at five points on either side of the piston (see force vectors in Figure 1.3). The stiffness for the impact at these locations

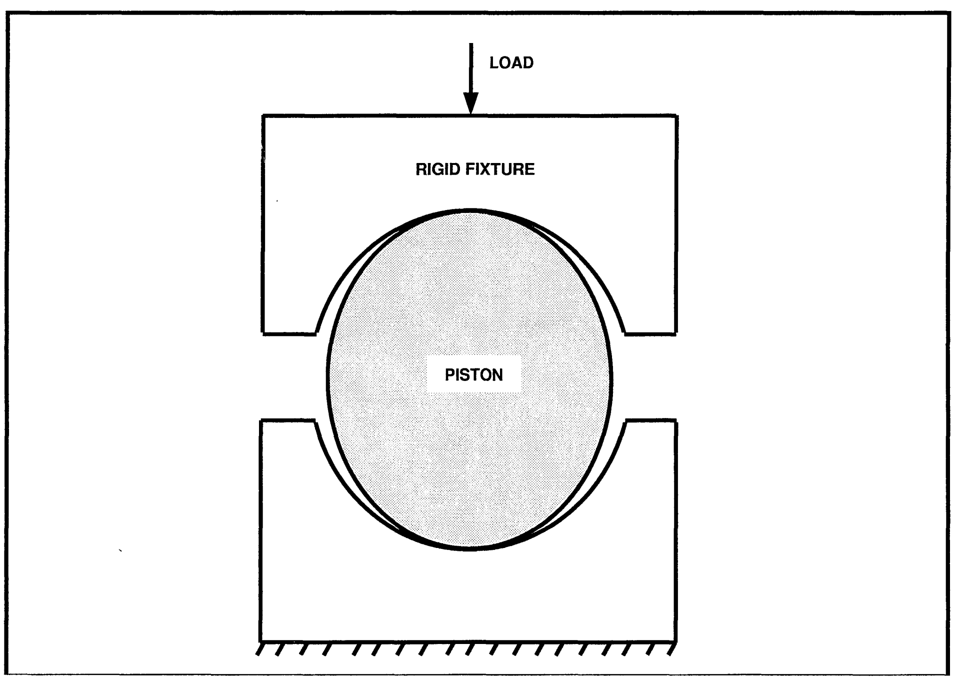

Figure 1.4 Fixture used to measure piston skirt stiffness.

can be determined either experimentally or with a a non-linear contact analysis using finite element techniques. Experimentally, a fixture as shown in Figure 1.4, is constructed and then loaded to produce the non-linear force-deflection curve shown in Figure 1.5. This measurement is taken at the five axial heights of interest on the piston. Similarly, the curve can be produced by an FE model of such a fixture. The force-deflection curve produced is then fit with a 3rd order polynomial. The coefficients for the polynomial are used as input parameters for the ADAMS model and are used in the subroutine (VFOSUB) to calculate stiffness based on the state variables during the solution.

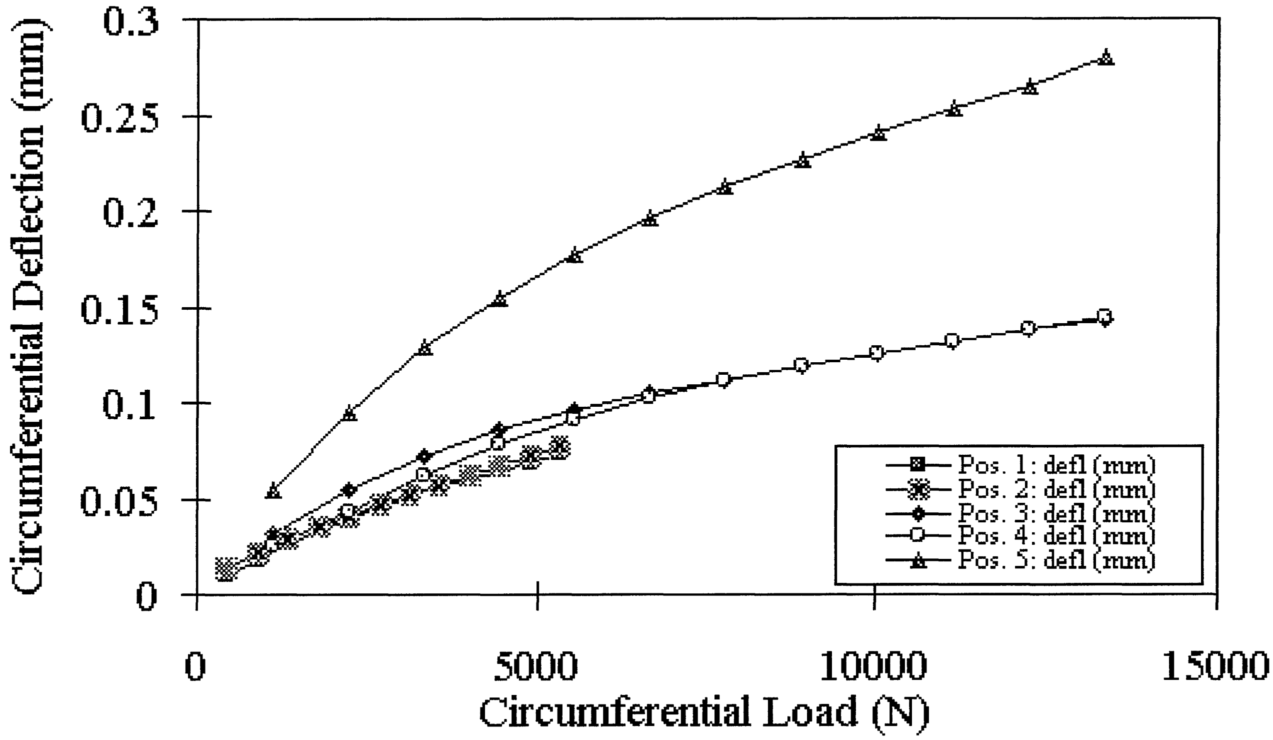

Figure 1.5 Load-deflection curves for piston skirt stiffness measurement.

2 MODEL CALIBRATION

2.1 Liner Model Calibration

Three separate pieces of experimental data were used to calibrate and verify the liner model. The first was free-free, or unconstrained modes of the liner itself, which were measured experimentally. The second was a measure of the spring stiffness of an actual rubber crevice seal. The most critical piece of data for verifying the model were the modes of the liner when assembled in the engine with the crevice seal installed.

The results of the free-free modal analysis conducted with the FE model in ANSYS, matched the measured modal results within a maximum deviation of 2.3 %. Modes for the liner with the rim constrained to simulate the clamping of the liner top stop between the block, head gasket and cylinder head were calculated with the FE model, but were not experimentally verified directly.

The crevice seal stiffness was measured by compressing a 35 mm section of the seal with a

Tinius-Olson machine. The resulting load deflection curve was used to determine the stiffness used in the linear spring-damper elements, in addition to the initial compressed length and preload. Because the stiffness of the seal is non-linear, and the ADAMS representation for the seal is linear, a point in the range of load and deflection that the seal is expected to see was selected, and a linear stiffness was calculated. The geometry of the assembled block and liner were used to calculate the initial compressed size of the seal, and the resulting preload.

The modes predicted by the MSS model compared favorably with the two modes measured on assembled system on the engine. Because there are many block modes in the range of frequencies being measured, it was not possible to measure more than two modes clearly. The two modes that were matched came within 1.5 % of the measured modes. Modal plots of these modes as predicted in the MSS are shown in Figure 2.1, below. Damping in the crevice seal was adjusted to fine tune the frequencies of the ADAMS model.

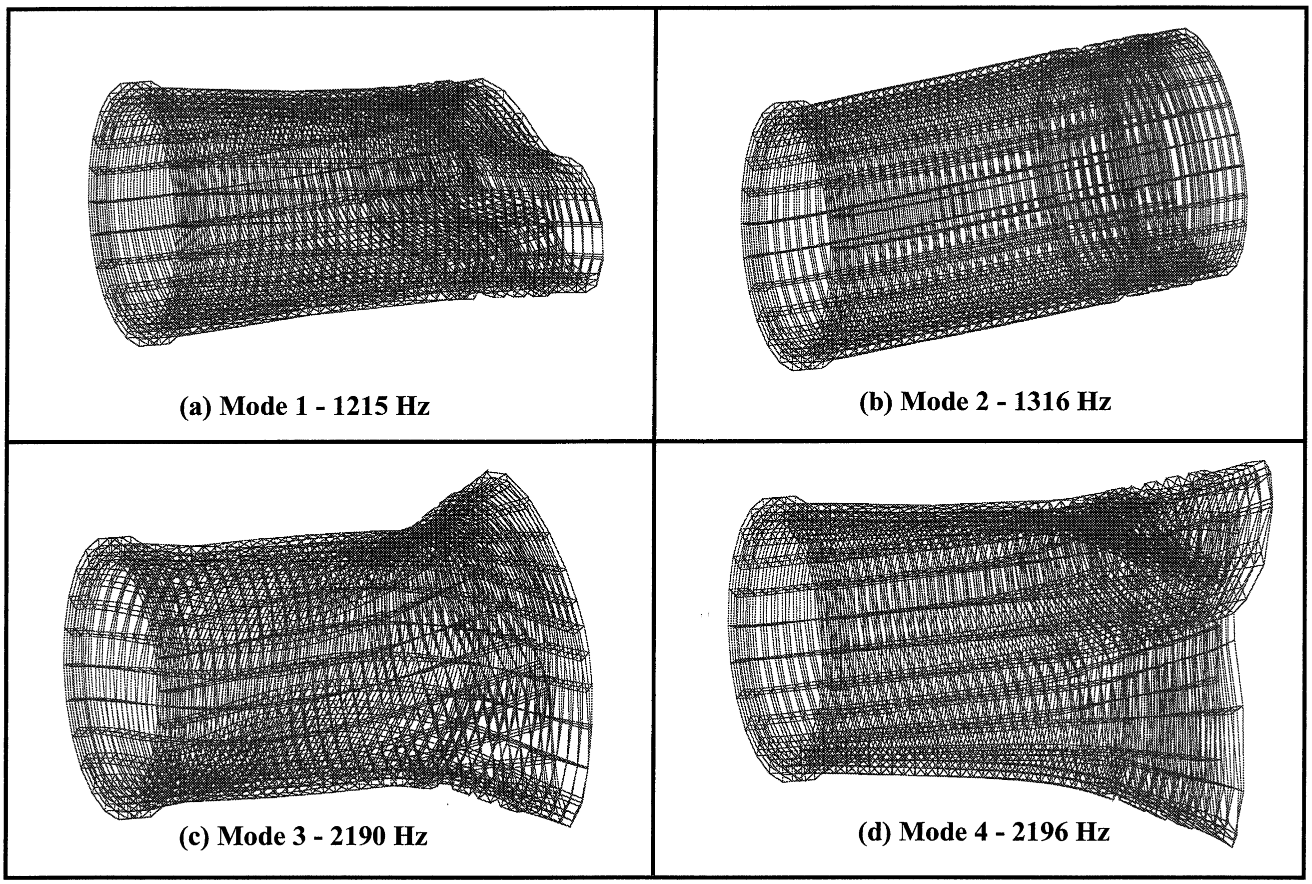

Figure 2.1 The first four modes as predicted in the MSS model including the liner and crevice seal.

Actual liner surface velocity predictions from the secondary motion model have not been scrutinized with respect to measured data at this point in time. Preliminary experimental results have shown that the liner surface velocity predicted by the model are within 30% of peak values measured in a similar engine.

3 CONCLUSIONS

The results of the analysis work documented here show a good match to experimental data. With the representation of the flexible liner in ADAMS experimentally verified, the next steps of refining the coupled piston secondary motion model and predicting liner motion will be easier to trouble shoot and verify. This analysis has established that it is possible to get good representations of flexible bodies into the ADAMS MSS software. This is an important accomplishment as a number of other projects that will combine FE flexible representations into dynamic simulations are currently planned at Cummins Engine Company.