MODELLING OF SNAP START BEHAVIOUR IN AN AUTOMOTIVE DRIVELINE

S A Kember – Ricardo Consulting Engineers Ltd.
N N Powell – Ricardo Consulting Engineers Ltd.
M Poggi – Ricardo Consulting Engineers Ltd.
D Ellis – Ford Motor Company
J Sung – Ford Motor Company

ABSTRACT

Snap start is the term given to sudden engagement of the clutch which can cause significant loads in the driveline and on the powertrain mounts. This paper describes an ADAMS model of a front wheel drive powertrain and driveline, and shows an effective and simple method of simulating sudden clutch engagement.

Results are presented to demonstrate how the complex interactions between non-linear components can affect the peak driveline and powertrain mount loads. This modelling approach is included in the vehicle design and development process to optimise specifications so as to minimise loads in the system and to predict maximum loads for stress analysis.

INTRODUCTION

Snap start or 'idiot start' is the action of rapid clutch engagement while the engine speed is held typically between 2000 and 5000 rev/min and the vehicle is stationary, with or without simultaneous application of wide open throttle.

When the clutch is engaged a large inertia torque is generated, due to rapid deceleration of the engine which, in addition to the normal engine torque, can induce high loads for a short duration in the engine mounts and driveline components. This condition often imposes the highest loads on the powertrain mounts and driveline components of any driving condition. Therefore the study of snap start is an important part of the vehicle design and development process.

The dynamic system involved in snap start consists of the clutch engagement process, the powertrain inertia suspended on its mounts and the rotating inertia, the clutch damper, transmission, driveshaft, tyres, vehicle body and vehicle suspension. The behaviour of the mounts, clutch, system backlashes, and tyre slip are all highly nonlinear and thus it is necessary to use time domain multi-body system analysis tools such as ADAMS to study snap start response. This paper describes a modelling approach which is now routinely used to study snap start and includes a description of the interactions between events which can affect component loads.

THE MODEL

The component parts of a snap start model are: vehicle body, simplified vehicle suspension, powertrain and mounts, flywheel, clutch engagement process and clutch damper, transmission, differential, driveshafts, CV joints and tyres. The vehicle body, powertrain and flywheel are all modelled as rigid bodies.

The model of a transverse front wheel drive powertrain is shown in Figure 1. For clarity the vehicle body and rear tyres with suspension are not shown. The main component parts of the model are described in the following sections.

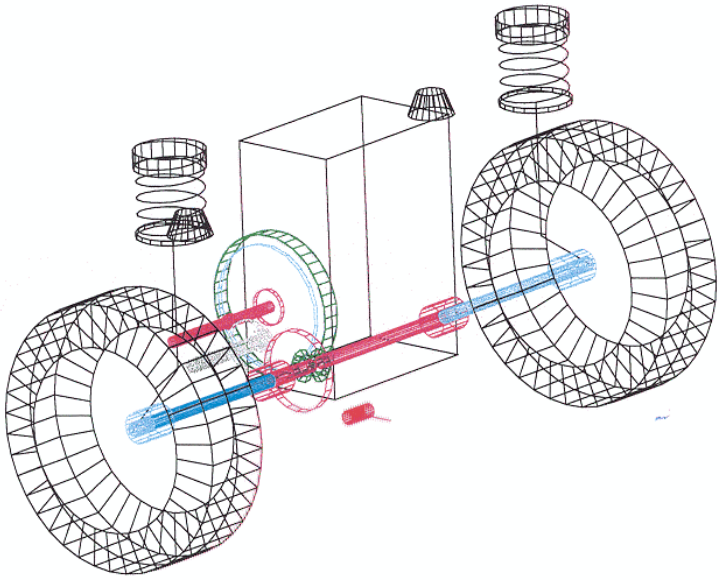

Figure 1 ADAMS Model for Snap Start Simulation

Powertrain Mounts

The powertrain mounts are modelled with nonlinear force functions in all directions. The load/displacement characteristic has a linear section with much increased stiffness at either end to represent the snubbing. A menu driven program has been developed in MATLAB to calculate a polynomial function, to represent the mount stiffness, suitable for insertion into ADAMS. Points on the load/displacement mount curve are entered via a graphical user interface, as shown in Figure 2. The user can choose the order of the polynomial and compare the result to the original specification of the mount in order to ensure an acceptable fit without overshoots. Preload is added to the engine mount stiffness functions to support the powertrain at its correct installation position. Damping in the mounts is included as a separate force function proportional to the relative velocity of the engine and vehicle body in each direction.

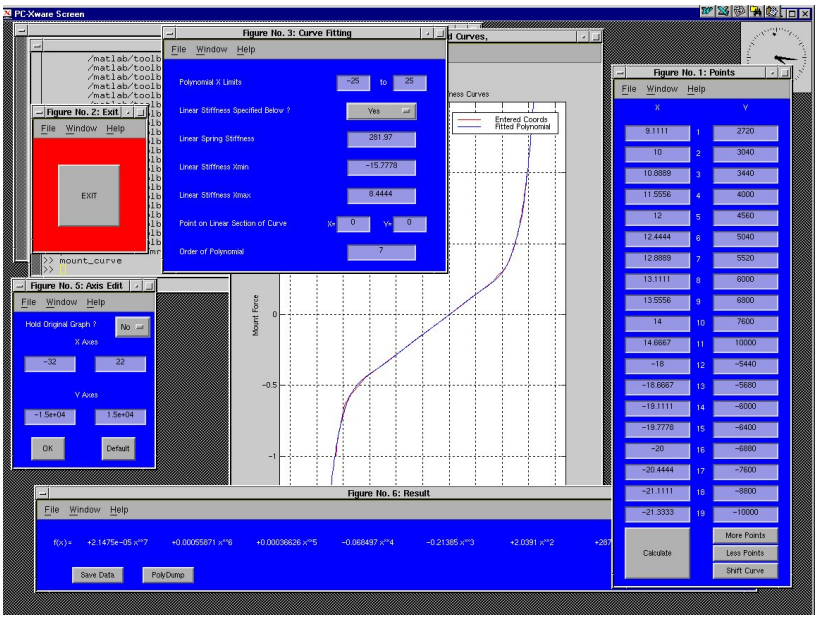

Figure 2 GUI of MATLAB Mount Curve Fitting Program

Clutch Engagement and Clutch Damper

The engagement of the clutch is an important part of a snap start model. A relatively simple method of simulating this action has been found to give good results as follows. The torque capacity of the clutch is calculated as a product of the clutch clamping force, the coefficient of friction and effective clutch radius. The actual torque transmitted is then calculated as the product of the torque capacity and a function of the relative speed between the flywheel and friction disc, i.e. the clutch slip speed, as shown in Figure 3. A small amount of slip is always necessary to transmit torque but this does not significantly affect the prediction of the snap start behaviour.

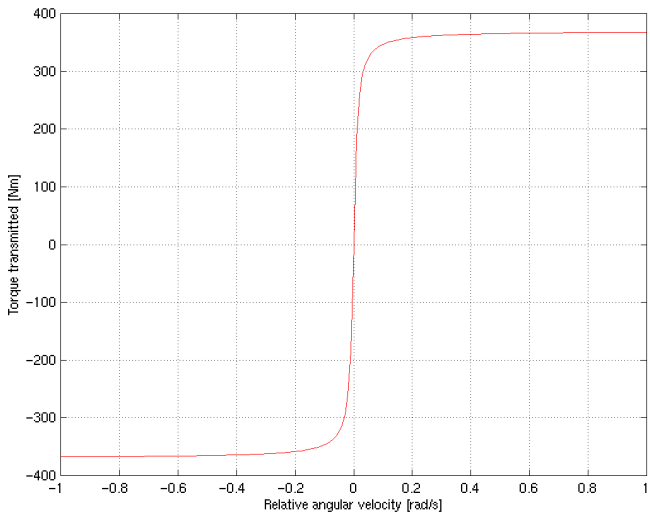

Figure 3 Clutch Torque vs. Slip Speed Curve

The clutch damper plate includes nonlinear springs with hysteresis and bump stops. These are modelled by forces which are activated with step functions dependent upon the relative angular displacement between the clutch friction disc and clutch hub, see Figure 4. The bump stops are high value stiffnesses brought in by step functions when the relative angular displacement exceeds a given angle. The hysteresis is also stepped in according to the relative displacement but the value is calculated as a function of the direction of relative angular velocity between the clutch friction disc and clutch hub.

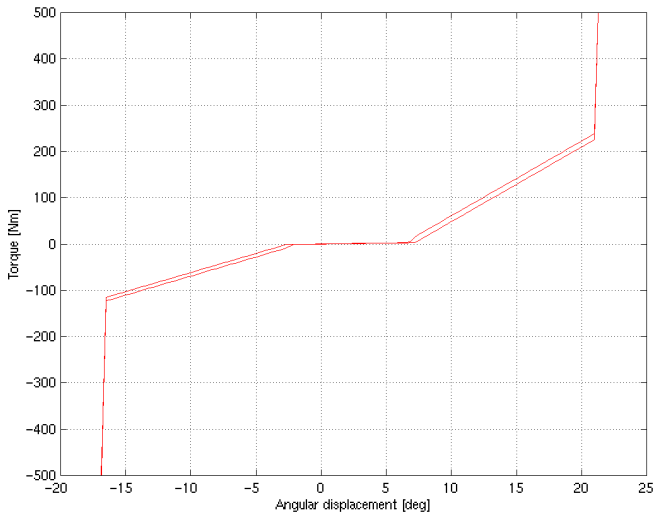

Figure 4 Clutch Damper Plate Spring Characteristic with Hysteresis

Transmission and Driveline

The transmission is represented by the first and final drive gear ratios and shaft inertias. A differential model is also included, to allow even torque distribution between the unequal length driveshafts on either side of the vehicle. The driveshafts are represented by shaft inertias and CV joints with flexibility given by torsional springs and dampers. Translational joints are also included in the drive shafts to allow plunge action as the wheels move vertically. The tyres are represented by the FIALA tyre model and appropriate resistance loads are applied on the vehicle. Backlash in the driveline is represented by a function that avoids discontinuities and overshoots which can cause integration problems [1].

Model Operation

The model is accelerated to the required engine speed by applying a torque to the flywheel which decays as the engine reaches the required start speed. The flywheel inertia includes an equivalent rotating inertia of the con-rods and pistons. The snap start condition is then initiated by applying the clutch clamp load using a step function. Thereafter the engine torque is calculated as a function of the engine speed by a spline relationship. This method is preferable to applying a motion because it allows the flywheel behaviour to be affected by the loads induced via the clutch.

SNAP START SIMULATION

By close examination of the results of the ADAMS simulation it is possible to understand the behaviour of the complete system during the snap start process and to understand the parameters which affect the maximum loads in the system. This section describes a typical sequence of events which occur immediately after clutch engagement and shows the significance of relatively small changes in design parameters on maximum loads.

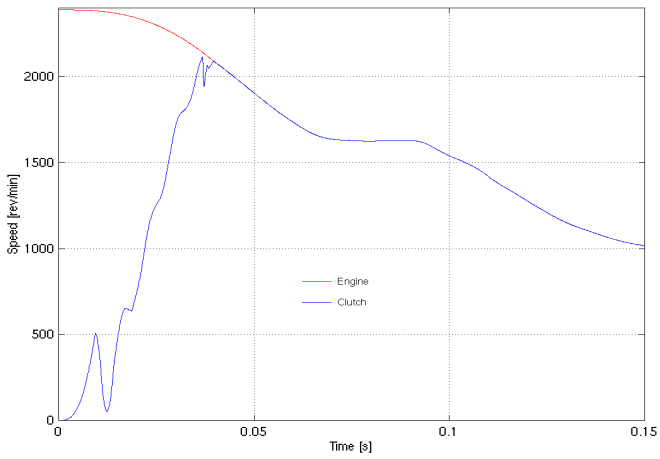

Figure 5 Engine and clutch speeds (flywheel and friction plate)

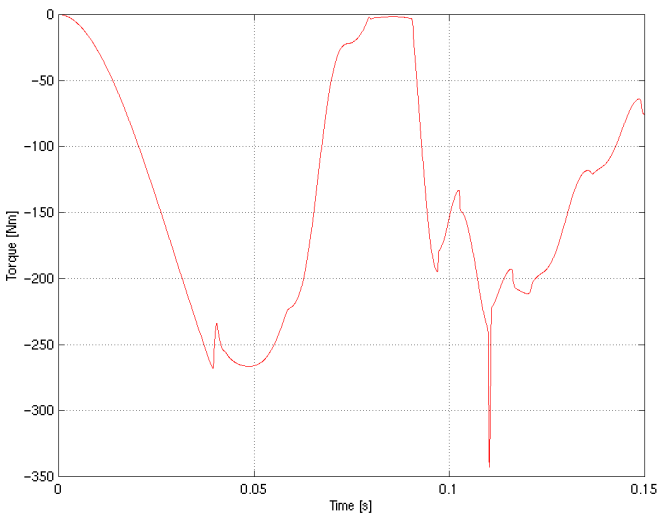

Figure 6 Clutch torque

The clutch slips during the first 0.04 seconds after engagement, as shown in Figure 5. During this period, the torque transmitted is governed by the clutch torque capacity. Subsequently, only the available torque is transmitted, without clutch slip, as shown in Figure 6.

The sudden application of torque causes the engine to rock backwards, about its roll axis until the roll restrictor reaches its snubber. At approximately 0.045 seconds after the start of clutch

engagement the engine begins to roll forward which is clearly evident in the absolute angular velocity of the inner end of the driveshafts, as shown in Figure 7.

The torque is transmitted to the wheels causing the tangential force at the tyre/road interface to increase until the tyres slip, at approximately 0.03 seconds after commencement of clutch engagement, as shown in Figure 8. The point of maximum displacement, and hence maximum driveshaft torque, occurs when the velocity of the inner and outer end of the driveshafts equate. This occurs at approximately 0.05 seconds, as shown in Figures 9 and 7.

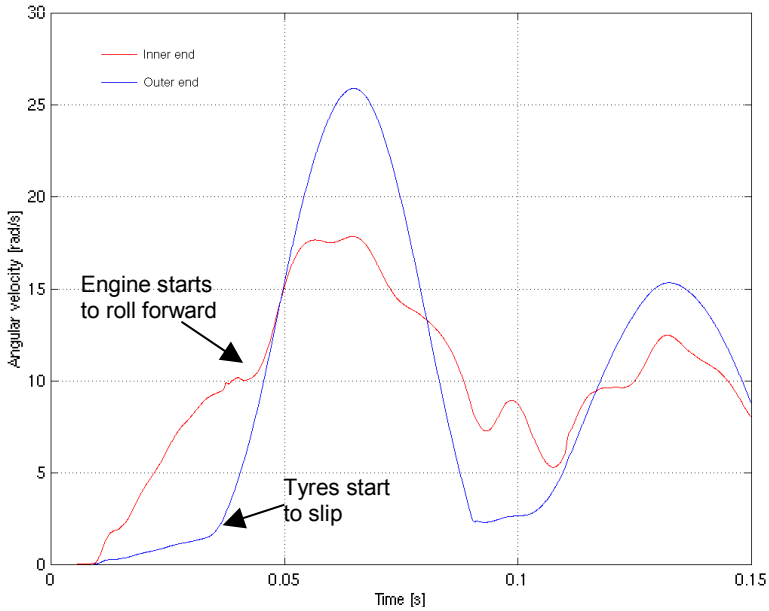

Figure 7 Driveline Velocities

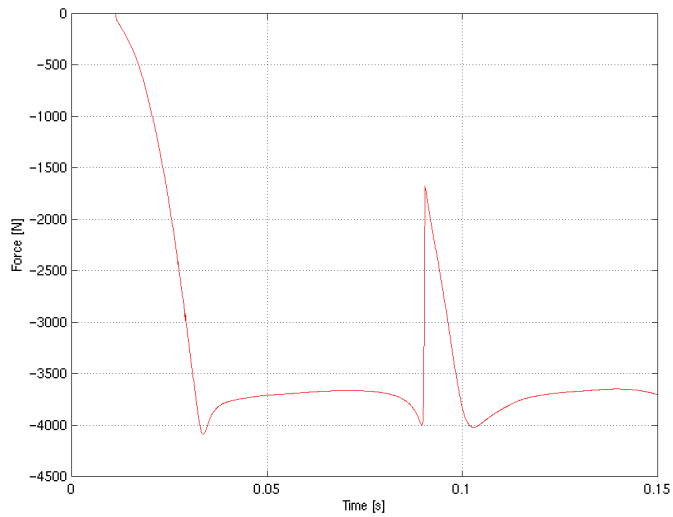

Figure 8 Tyre / Road Tangential Force

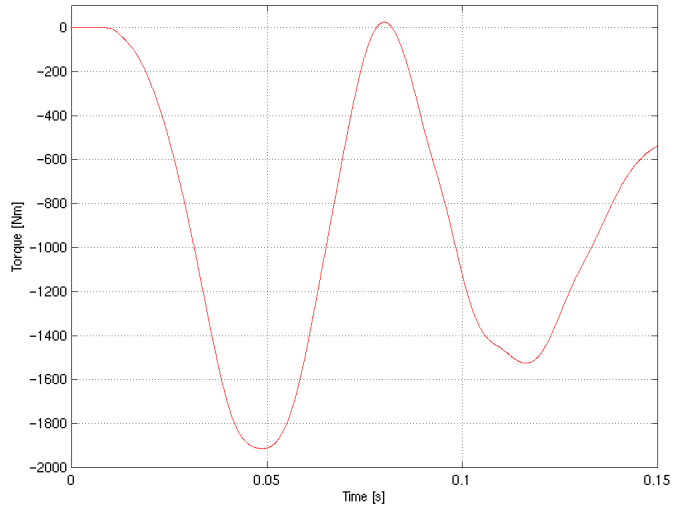

Figure 9 Driveline Torque

It can be seen in Figure 7 that factors which change the point at which the angular velocities of each end of the driveshafts match can have a significant affect on peak driveline torques. These factors include tyre friction, wheel inertia and the powertrain mount characteristics. Figure 10 shows the effect on driveshaft torque of variation in wheel inertia. Increasing the wheel inertia causes the instant at which the velocity of the inner and outer end of the driveshafts equalise to occur later, as shown in Figure 11, and hence the peak in driveshaft torque is significantly higher, as shown in Figure 10. Conversely, a decrease in the wheel inertia causes the instant of velocity equalisation to occur earlier, as shown in Figure 12, and hence the peak in driveshaft torque to be significantly reduced, as shown in Figure 10.

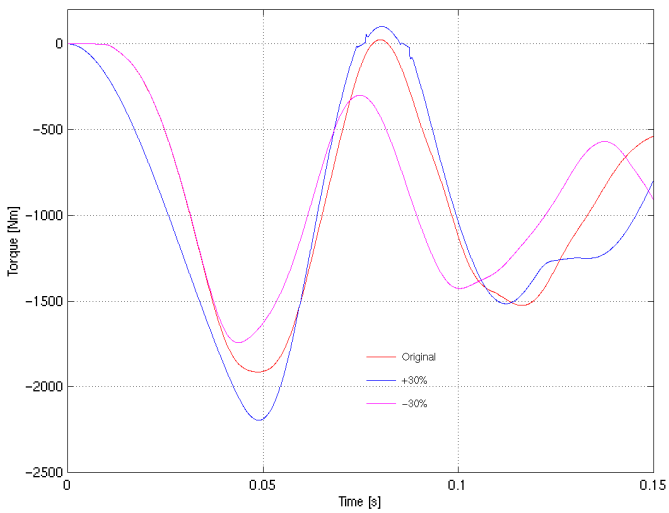

Figure 10 Variation of driveshaft torques with variation of wheel inertia

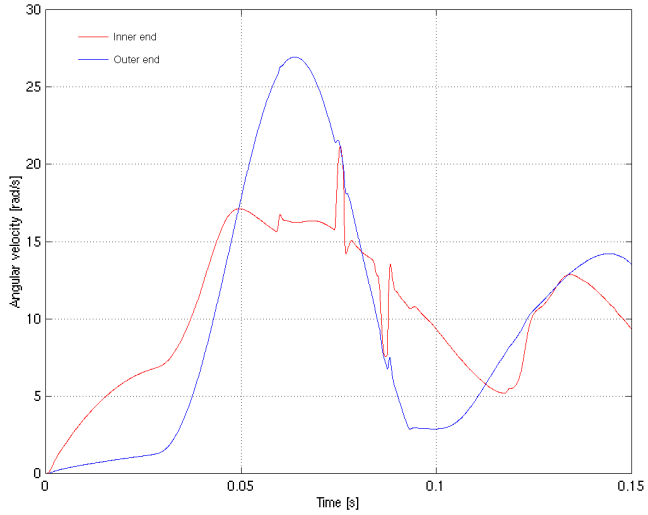

Figure 11 Driveshaft velocities with wheel inertia increased by 30%

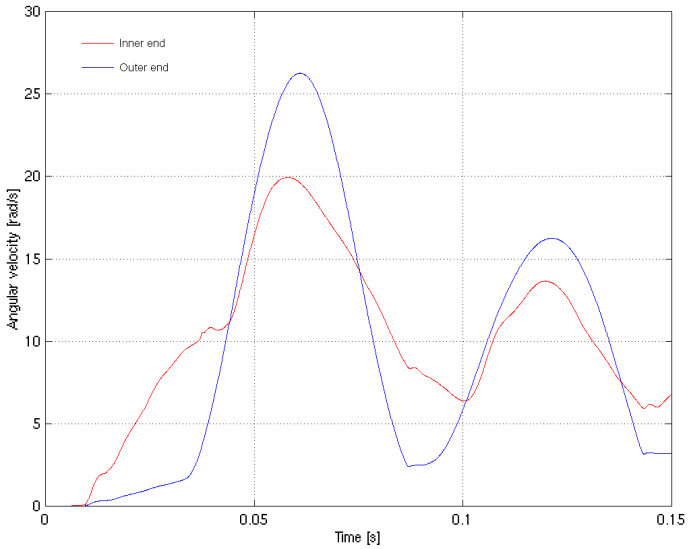

Figure 12 Driveshaft velocities with wheel inertia decreased by 30%

The model can also be used to determine peak loads in the engine mounts. For example, Figure 13 shows that the peak load in the roll restrictor occurs at approximately 0.13 seconds after commencement of clutch engagement which is not coincident with the peak driveline loads.

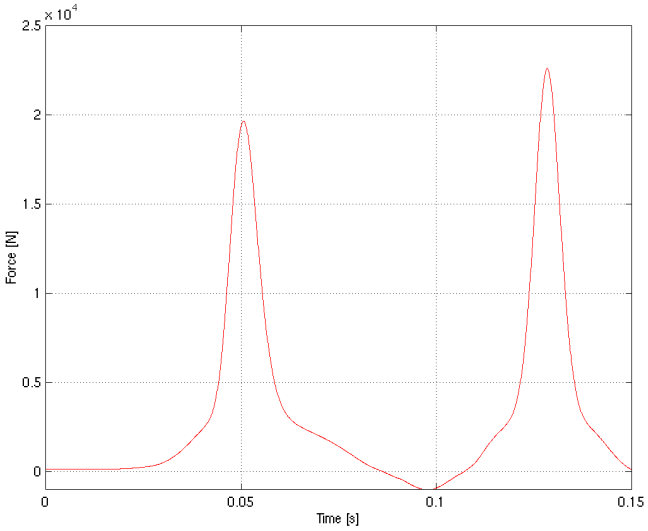

Figure 13 Roll Restrictor Load

CONCLUSIONS

This study demonstrates the use of ADAMS to model a complex system with several highly nonlinear features. In particular, a simple method to model clutch engagement has been shown to be effective.

The model provides design and development engineers with a useful tool to aid understanding of the complex interactions which occur during snap start. This modelling approach is included in the design process to optimise specifications so as to minimise loads in the system and to predict maximum loads for stress analysis.

The modelling approach described here can also be developed to study driveability [2] and gear rattle [1].

ACKNOWLEDGEMENTS

The authors wish to express their thanks to Ford and to the directors of Ricardo Consulting Engineers for their permission to publish this paper.

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2 CY Mo, AJ Beaumont, NN Powell, 'Active Control of Driveability', SAE 960046