Transient Analysis of an Integrated Powertrain and Suspension System

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ABSTRACT

An ADAMS model of an integrated powertrain and suspension system for a heavy truck is presented in this paper. The ADAMS model includes an engine, clutch, 10-speed transmission, driveline, drive axles and differentials, and a suspension system for a tandem rear-axle tractor. The model was developed in order to predict the dynamic performance of the powertrain and suspension system during transient events. Specifically, the model is used to predict the variation of the universal joint angles in the driveline as well as the effect of universal joint angles on powertrain vibration during acceleration and deceleration. The model can also be used to predict overall vehicle performance during transient events such as clutch engagement and gear shifting.

Introduction

Powertrain vibration and driveline dynamics are major concerns in heavy truck component reliability, NVH performance and ride quality. In this paper, powertrain refers to everything from the engine to the driven wheels, including the clutch, transmission, driveline, gear reduction and differential, and axle shafts. Powertrain vibration plays a crucial role in the reliability of powertrain components because the service loads that act on the components are dominated by the dynamics of the powertrain. Similarly, noise due to resonance between the engine's firing frequency and powertrain's natural frequency is dependent on the dynamics of the powertrain system. Furthermore, powertrain vibration may affect heavy truck ride in the form of fluctuating traction forces on the ground, thus generating fore-aft vibrations in the vehicle, as well as fluctuating reaction forces at the engine mounts, which in turn generate vibrations in the roll direction.

The most common sources of powertrain vibration are driveline excitation and engine torque fluctuation. Driveline excitation may be classified into first harmonic torsional vibrations due to mass imbalances and second harmonic excitations due to the presence of universal joint angles. Engine torque fluctuation occurs because the torque delivered at the engine's crankshaft is in the form of pulses coming from the ignition of fuel in each cylinder as well as the inertia of the crankshaft and the pistons. The engine flywheel absorbs most of this pulsation, but some of the lower frequency components of this pulsation is transmitted throughout the powertrain. The above sources of excitation are present in both transient and steady state conditions. Transient conditions such as vehicle starts or gear shifting, however, may produce more severe torsional vibrations than those found in steady state conditions due to larger universal joint angles during axle windup and also due to other sources of excitation such as clutch friction torque fluctuations during clutch engagement.

The objective of this paper is to present an ADAMS model developed for conducting a transient analysis of a heavy truck undergoing vehicle start and acceleration along with gear shifting. The goal of the transient analysis is to estimate the variation of the universal joint angles during the transient event, as well as to determine the effect of the universal joint angles on the powertrain vibration. Cognizant of the fact that the universal joint angles depend on the amount of axle windup during transient events, a suspension system was also included in the model in order to properly estimate the amount of windup in the drive axles.

Model Description

A graphical representation of the integrated powertrain and suspension system of a heavy truck is shown in Figure 1. The heavy truck has a tandem rear drive axle. The front and rear suspensions consist of nonlinear spring forces with friction damping and nonlinear shock absorbers. In addition to axle bounce and axle roll degrees of freedom, the rear axles are allowed to rotate in the pitch mode in order to simulate the drive axle windup. The default ADAMS tire model is used to model the tires. The powertrain consists of the following: engine and flywheel, clutch, 10-speed transmission, driveline with center bearing and cross-type universal joints, inter-axle differential, forward-rear gear reduction and differential, inter-axle slip tube and universal joints, rearward-rear gear reduction and differential, and drive axle shafts.

The engine and flywheel are represented by a lumped inertia and a simple engine torque map which is a function of the input throttle and the wide-open-throttle engine torque-velocity curve. The clutch is divided into two discrete inertias: one representing the clutch disks and the other representing the clutch hub. The friction torque between the disks and the friction plate is a function of the clutch disk geometry, clutch friction coefficient, and the normal force coming from the diaphragm spring. The torque transmitted from the disks to the hub is represented by a torsional spring and a viscous damper. The 10-speed transmission is represented by a set of spur gears and the following compliant shafts: transmission input shaft, main countershafts, main shaft, auxiliary countershafts, and transmission output shaft. Shaft compliance is modeled by distributing the shaft inertia into a set of discrete inertias which are connected in series by torsional springs and dampers. The main gears and countershaft gears are connected to the shafts through force (torque) elements which in turn are activated according to a prescribed gear shifting schedule.

Torque is transmitted from the transmission output shaft to the driveline by a cross-type universal joint. The driveline consists of a compliant non-slip tube connected to a compliant slip tube by another universal joint. The non-slip tube is hanging from the vehicle frame through a compliant "center bearing." The slip tube is connected to the inter-axle differential input shaft by a third universal joint.

The inter-axle differential and drive axle differentials are modeled by a train of bevel gears. The ADAMS model is capable of simulating both locked and unlocked differentials. When a differential is unlocked, the output velocities are not equal, but the torque is equally split between the two output gears. The forward-rear axle and the rearward-rear axle are connected by the inter-axle slip tube through universal joints. The slip tube allows articulation between the two drive axles. When a differential is locked, the torque is distributed between the two output gears in a manner such that they have equal velocities. In the present study, the differentials are left unlocked.

Model Validation

Keeping in mind that the objective of the present modeling effort is to determine the transient response of the powertrain during vehicle start up and acceleration, the ADAMS model is validated by evaluating the system natural frequencies corresponding to the first powertrain mode. This is so because we know from previous experience that the first powertrain mode dominates the transient response of the powertrain from external inputs. Therefore, validating the first powertrain mode natural frequency and mode shape for each gear setting in the transmission would enhance our confidence in the results of the transient analysis. For this purpose, we compare the first powertrain mode natural frequency obtained from the ADAMS model with that of Rockwell's in-house Drivetrain Vibration Program (DTV) which is a frequency-domain program designed for evaluating the frequency response of powertrains. The table below shows the comparison of undamped natural frequencies associated with the first powertrain mode for various transmission gear settings:

Transmission Gear Setting	ADAMS	DTV
1st	0.54	0.67
2nd	0.81	1.04
3rd	1.37	1.37
4th	1.89	1.81
5th	2.66	2.57
6th	3.63	3.52
7th	5.01	4.84
8th	5.64	6.38

Table 1: Natural Frequency (Hz) of First Powertrain Mode

We can see from the table above that the results obtained from ADAMS (wherein the response is obtained in the time-domain) agrees well the frequency-domain program DTV, in both magnitude and trends as the gear settings are changed. The small discrepancies are due to the fact that the distribution of the system's inertia and shaft compliances are different between the two models.

Simulation Results and Discussion

In this section, we present the simulation results for the transient response of the powertrain and suspension system during vehicle start-up and acceleration. We did not include engine torque fluctuation in the model because our goal in the present analysis is to isolate the effects of universal joint angles on powertrain vibration during vehicle start-up and acceleration. Figure 2 shows the vehicle speed and vehicle's fore-aft acceleration as the vehicle goes through the start-up and gear shifting process. We can see from this plot that the effect of universal joint excitation on the vehicle's fore-aft vibration becomes more pronounced at higher vehicle speeds, *i.e.*, higher driveline angular speeds. This is expected because we know that the amplitude of the excitation is proportional to the square of the driveline's angular velocity. Figures 3 and 4 show the time history of the universal joint angles in the main driveline and in the inter-axle driveline during the vehicle start-up and gear shifting process. In particular, we can see from Figure 4 that the universal joints in the inter-axle driveline undergo significant changes due to drive axle windup during vehicle start-up, and especially in the lower transmission gear settings. Figure 5 shows the angular velocity of the transmission's input and output shafts as the gear settings change during the acceleration process. In this plot, we can see the effect of gear shifting on the vibration of the transmission's input and output shaft. Closer inspection of the plots reveals that the transient response during gear shifting is dominated by powertrain's first torsional mode and by the drive axle windup mode. After the transient response damps out, the fluctuation in the angular velocity of the transmission input and output shafts are governed by the universal joint excitation. Figure 6 shows the friction torque transmitted from the friction plate to the clutch disks. We can make the following observations from this plot. First, the friction torque capacity of the clutch disks is saturated during gear shifting at the lower gear settings. This results in slipping between the clutch disks and the friction plate. Second, the torque fluctuation at the clutch is characterized by a transient response which is governed by the powertrain's first torsional mode. Third, the steady-state torque fluctuation at the clutch is governed by the universal joint excitation, and this excitation becomes more pronounced at higher driveline angular velocities. Fourth, at the 7th gear setting, the universal joint excitation frequency sweeps through the natural frequency of the second powertrain mode. The universal joint excitation is then in resonance with the powertrain's second torsional mode, and this results in large torque fluctuations across the clutch. This phenomenon has been observed in field testing of heavy trucks. Finally, at the 8th gear setting, the universal joint excitation frequency is above that of the second powertrain mode. Resonance does not occur, but the torque fluctuations across the clutch is also significant because of even higher driveline angular velocities.

Conclusions and Future Work

In this paper, we have presented an ADAMS model of an integrated powertrain and suspension system for the transient response analysis of a heavy truck undergoing vehicle start-up and acceleration. The purpose of the analysis was to determine the effect of drive axle windup and universal joint excitation on powertrain vibration. Future work will be directed towards the steady-state response of the powertrain to the combined effects of universal joint excitation and engine torque fluctuation. In order to achieve this goal, a detailed engine model needs to be added to the existing ADAMS model. This engine model should include a characterization of the amplitude and phasing of the torque produced by each piston. This engine model should also include a closed-loop control system that will represent the governor in the engine.

References:

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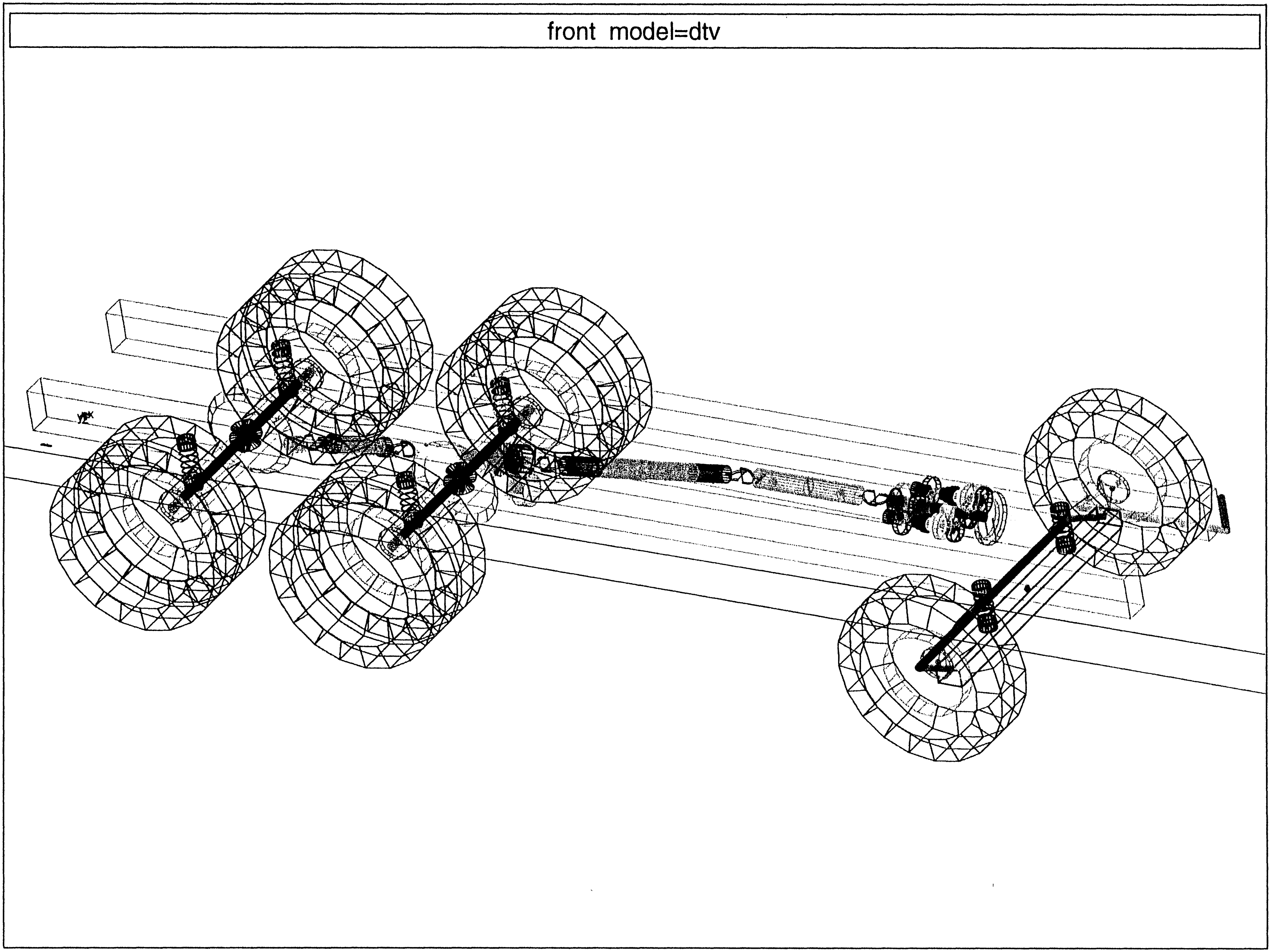
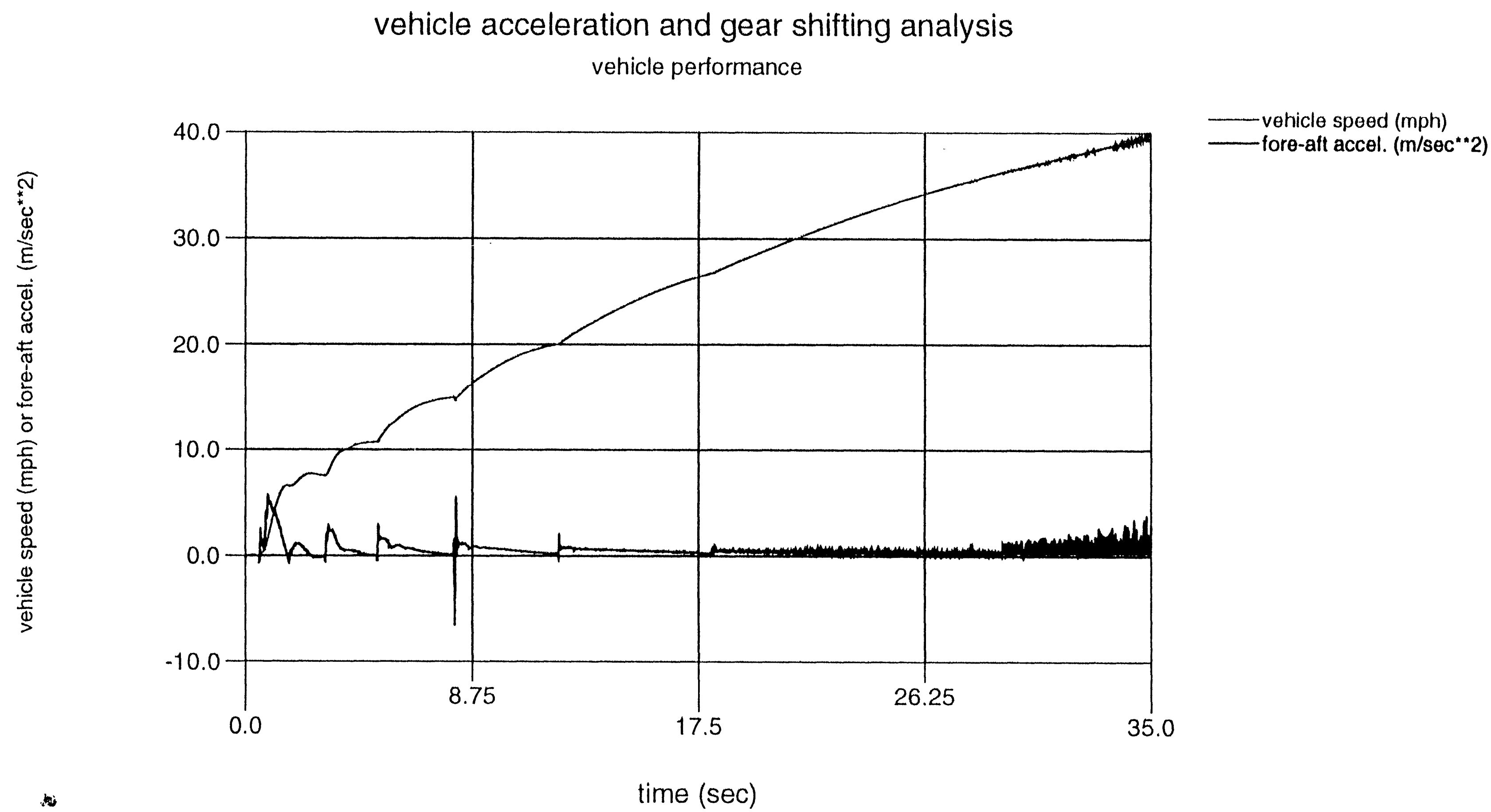


fig. 1



vehicle acceleration and gear shifting analysis
vehicle performance
vehicle speed (mph)
fore-aft accel. (m/sec**2)
vehicle speed (mph) or fore-aft accel. (m/sec**2)
40.0
30.0
20.0
10.0
0.0
-10.0
0.0
8.75
17.5
26.25
35.0
time (sec)

fig. 2

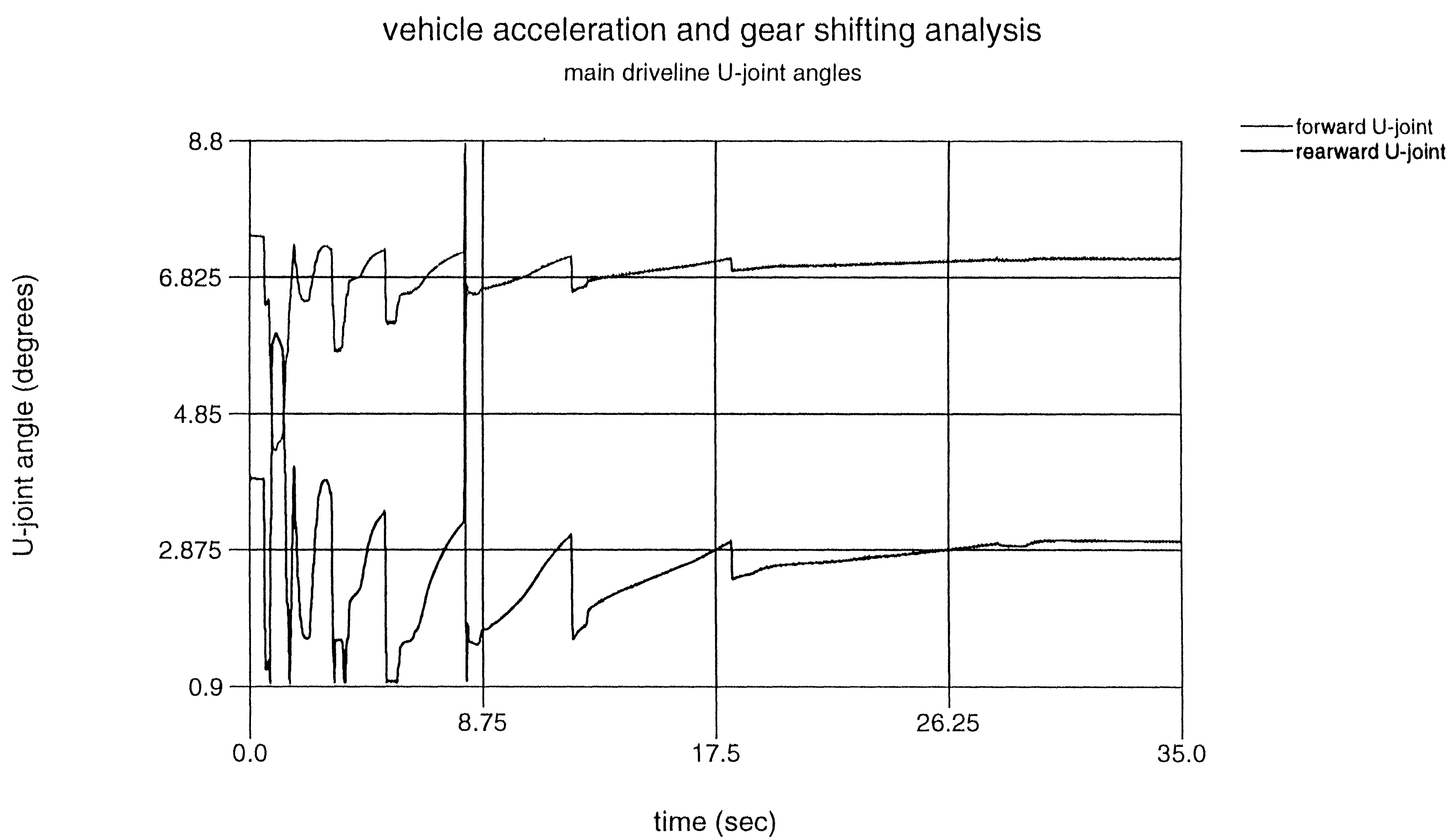
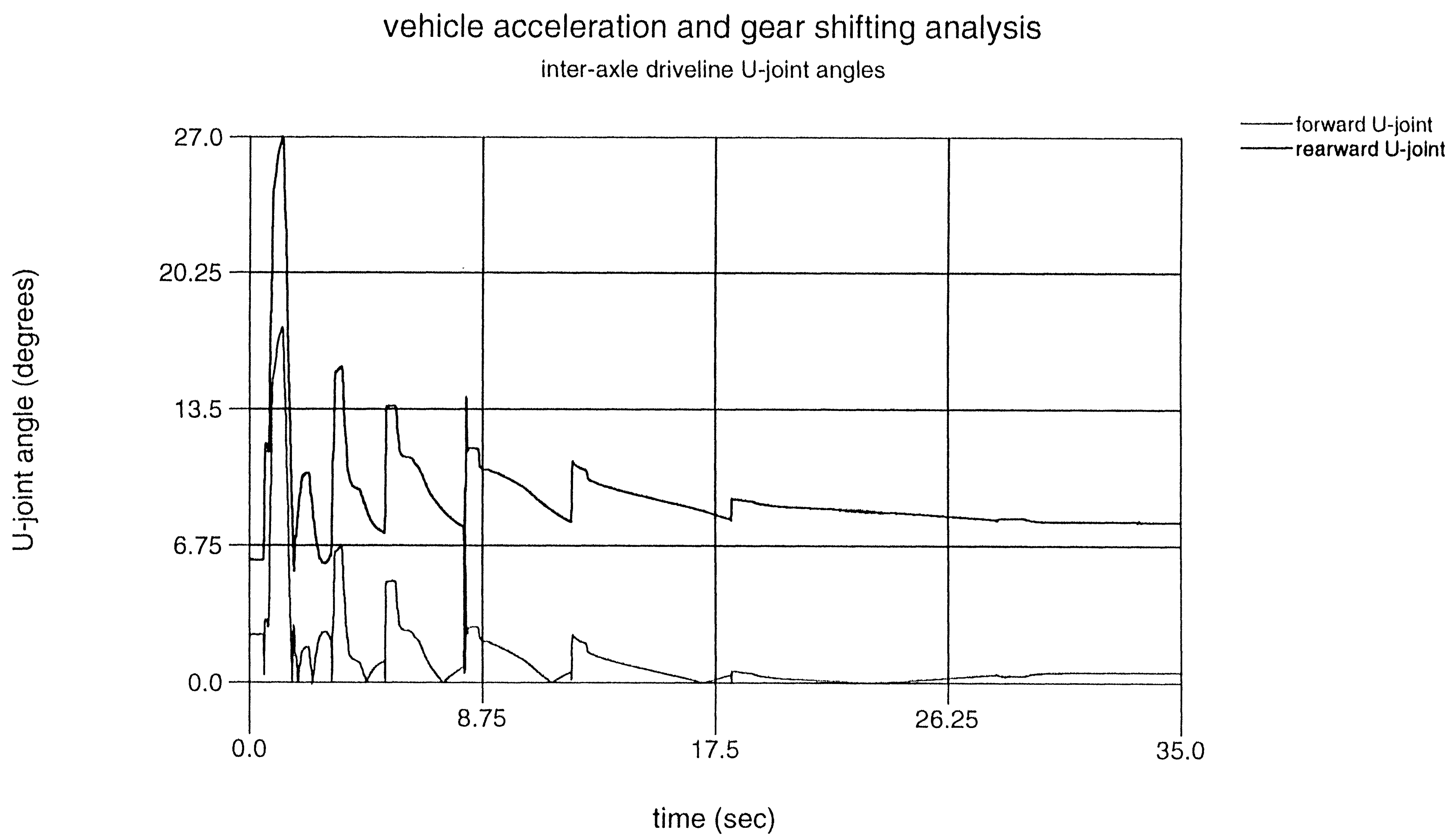

vehicle acceleration and gear shifting analysis
main driveline U-joint angles
forward U-joint
rearward U-joint
U-joint angle (degrees)
8.8
6.825
4.85
2.875
0.9
0.0
8.75
17.5
26.25
35.0
time (sec)

fig. 3



vehicle acceleration and gear shifting analysis
inter-axle driveline U-joint angles
forward U-joint
rearward U-joint
U-joint angle (degrees)
27.0
20.25
13.5
6.75
0.0
0.0
8.75
17.5
26.25
35.0
time (sec)

fig. 4

vehicle acceleration and gear shifting analysis

transmission input and output shaft rpm

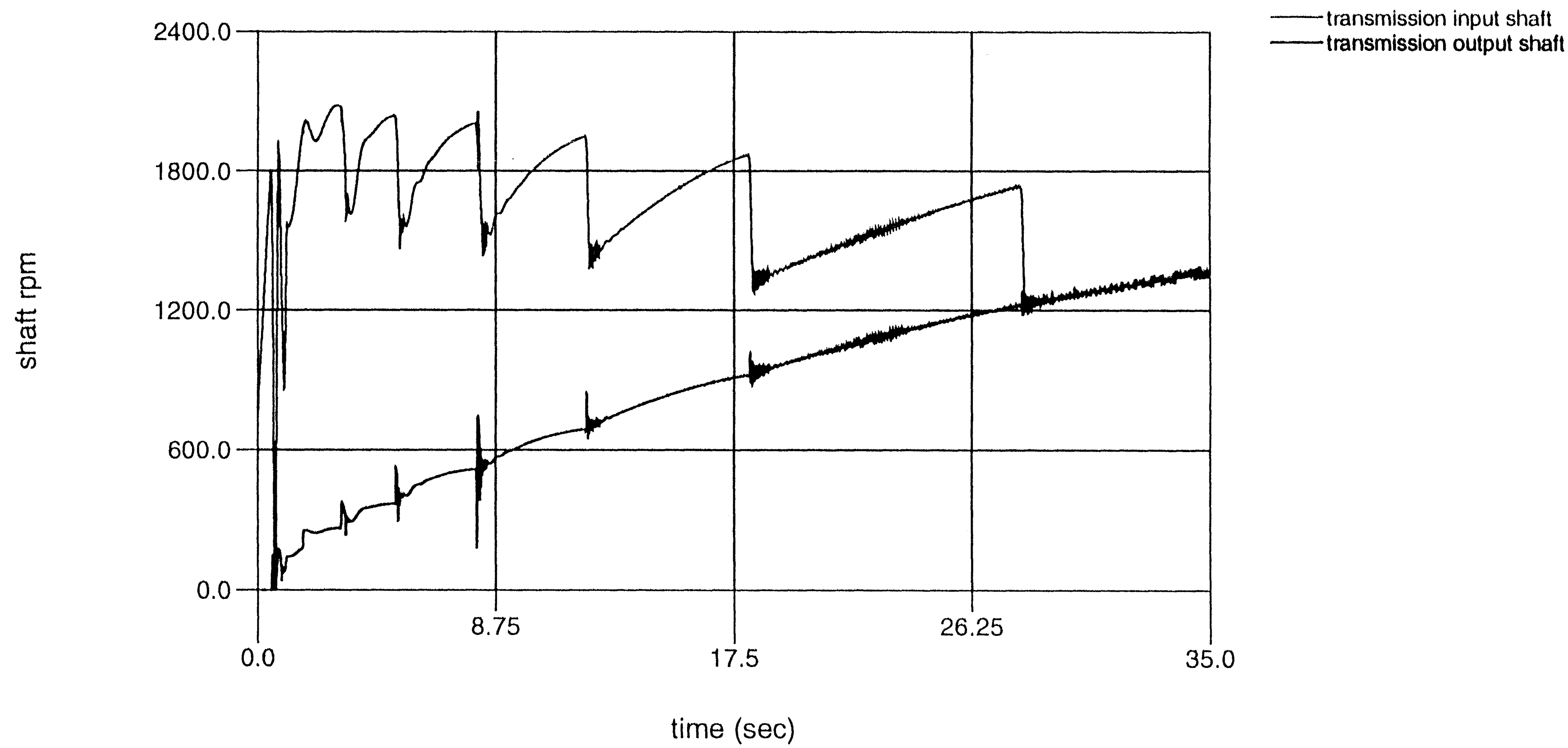


fig. 5

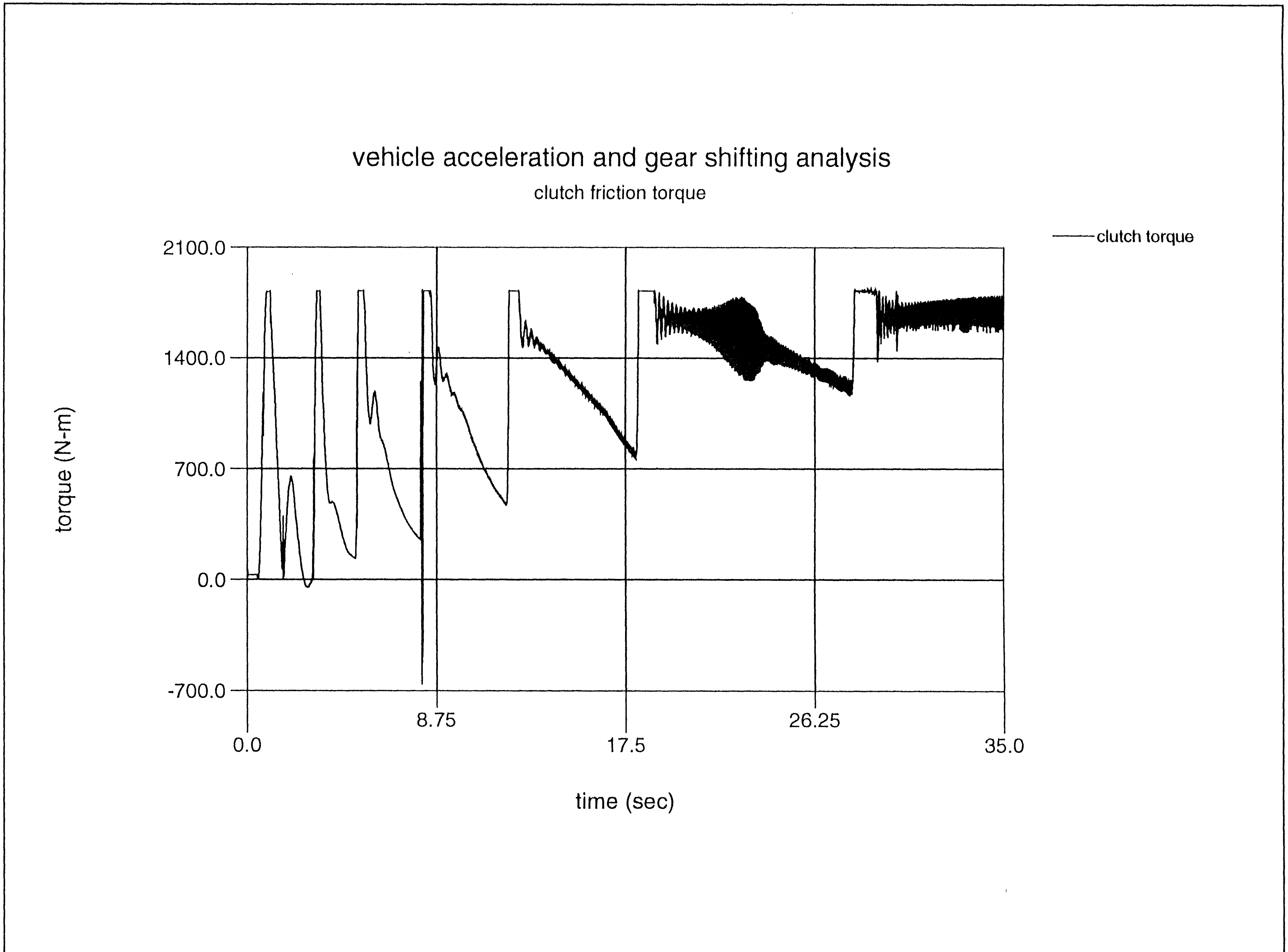

vehicle acceleration and gear shifting analysis
clutch friction torque
clutch torque
2100.0
1400.0
700.0
0.0
-700.0
torque (N-m)
0.0
8.75
17.5
26.25
35.0
time (sec)

fig. 6