ARAŞTIRMA MAKALESİ

DYNAMIC ANALYSIS OF VEHICLE POWER TRANSMISSION SYSTEMS

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TAŞIT GÜÇ AKTARMA ORGANLARININ DİNAMİK ANALİZİ

Özet

Bu çalışmada, taşıt güç aktarma organları, non-lineer elemanlar da dikkate alınarak modellenmiş ve simulasyonu gerçekleştirilmiştir. Simulasyon sonucunda, şaft ve kavrama titreşimleri zaman bazında incelenmiş ve taşıtın hız limitleri elde edilmiştir. Ayrıca, şaft ve kavrama titreşimlerinin frekans cevabı analizi de incelenmiştir. Modelleme esnasında, yuvarlanma, hava ve eğim gibi non-lineer seyir dirençleri de dahil edilmiştir. Çalışmanın sonunda, düşük viteslerde oluşan ileri-geri titreşimlerin simulasyonu gerçekleştirilerek sonuçlar yorumlanmıştır.

Abstract

In this study, a motor vehicle power train system has been modelled and simulated including also non-linear parts. As a result of simulation, shaft and clutch vibrations have been observed in time domain and speed limits of the vehicle has been obtained. Besides, frequency response analysis of shaft and clutch vibrations has been studied. During modelling, all the ride resistances such as rolling, air and slope which are non-linear have been included. At the end of the study, the longitudinal oscillations of the vehicle which happen when operating in lower gears have been simulated and results have been interpreted.

1. Introduction

Vehicle vibration models are the focus of the studies about vehicle vibrations recently (Nalecz, et al. 1988)(Hemingway, et al. 1985). These models are composed of vehicle body sprung mass, suspension system elements bonding vehicle body mass to the axles and wheels which are springs and dampers. During the design process, three criteria are important; ride comfort, working limits of suspension length and dynamic wheel pressure. On the other hand, displacement and acceleration of vehicle body become important while studying ride comfort (Michelberger, et al. 1987). Working limit of suspension length is defined as relative displacement between sprung and unsprung mass (Wilson, et al. 1986)(Burton, et al. 1995). Relative displacement between road and axle affect the dynamic wheel pressure and this value depends on the wheel properties, road irregularity and axle vibration amplitude (Dahlberg, 1980) (Nalecz, et al. 1988).

The numbers of the studies concerning the body bounce and pitch motion of the vehicle are many in literature (Yağız, et al. 1997). But the undesired longitudinal vibrations parallel to the road axis during the start of the vehicle are significant and needs to be analysed in order to reach a comfortable ride during the start particularly at the first or second shift. Since the principle source of this vibration mode which happens at 2-5 Hz is the torque produced by the vehicle motor and it is transmitted to the vehicle body through the power transmission systems, it becomes necessary to model the vehicle including the engine, transmission components and ride resistance.

2. Power Transmission System

The power transmission system of a vehicle is composed of the engine, clutch, transmission, differential, right and left axles and wheels as presented in Figure 1. The proposed model is enough to analyse the back and forth oscillations witnessed by most of the drivers during the start and parking manoeuvres.

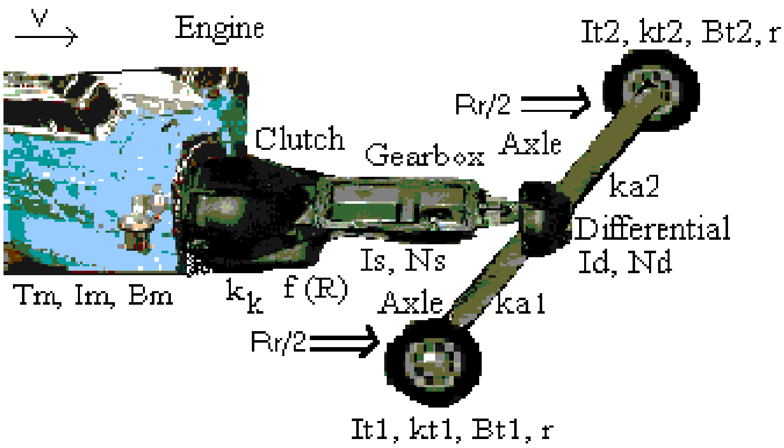

Figure 1. The Model of the Transmission System.

In this model:

M : Mass of the Vehicle,	R_r : Ride Resistance.
T_m : Engine Torque,	I_m : Engine Inertia,
f(R) : Clutch Dry Friction,	I_s : Transmission Inertia,
N_s : Gear Ratio,	I_d : Differential Inertia,
N_d : Differential Ratio,	I_t : Wheel Inertia,

k_{a1}, k_{a2} : Right and Left Axle Stiffnesses, k_t : Wheel Stiffness,

B_m : Engine Viscous Friction, k_k : Clutch Spring Stiffness,

B_t : Wheel Damping, r : Wheel Radius,

The Physical Model of the system has been shown in Figure 2. In this figure all elements and variables are rotational except vehicle mass and motion.

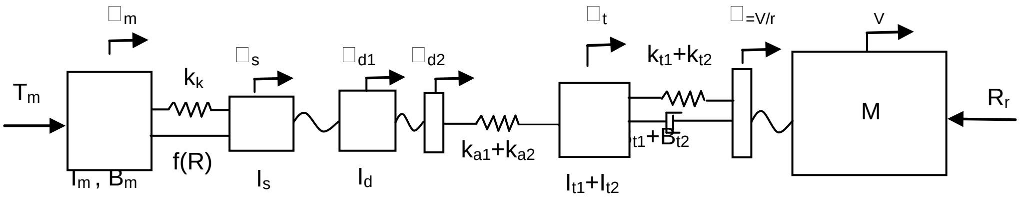

Figure 2. Physical Model.

Ride Resistance is the sum of the Wheel Rolling Resistance, Air Resistance and Slope Resistance (Gillespie, 1992). Inertia Resistances are also included in the model.

Rolling Resistance happens because of the road and deformation of the wheels and the energy dissipated as a result of the impact and defined as:

$$R_{rolling} = f_r\, W \tag{1}$$

where f_r is the rolling resistance coefficient and W is the weight of the vehicle. Air resistance depends on different factors such as the front surface profile area and structure of the vehicle, the dimension of the vehicle, all the surfaces of the vehicle in contact with the air. Assuming this resistance proportional to the area of the front of the vehicle and square of the speed of the vehicle is proved to be correct for all practical purposes. This quadratic relation makes air resistance non-linear and very effective at high gears (Ünlüsoy, 1986);

$$R_{air} = 0.609\, C_d\, A_f\, V^2 \tag{2}$$

C_d represents the aerodynamic drug coefficient, A_f is the front profile area of the vehicle in [m^2] and V [m/s] is the speed of the vehicle. Slope resistance is the result of the slope of the road. If the angular slope of the road is θ ;

$$R_{slope} = W \sin\theta \tag{3}$$

As a result ;

$$R_r = R_{rolling} + R_{air} + R_{slope} \tag{4}$$

Equations of motion of the vehicle is obtained by using Lagrange Equations and given below :

$$I_m \ddot{\theta}_m + B_m \dot{\theta}_m + f(R)(\dot{\theta}_m - \dot{\theta}_s) + k_k(\theta_m - \theta_s) = T_m \tag{5}$$

$$(I_s + \frac{I_d}{N_s^2})\ddot{\theta}_s - f(R)(\dot{\theta}_m - \dot{\theta}_s) + k_k(\theta_s - \theta_m) + \frac{(k_{a1} + k_{a2})}{N_s N_d}(\frac{\theta_s}{N_s N_d} - \theta_t) = 0 \tag{6}$$

$$(I_{t1} + I_{t2})\ddot{\theta}_t + (B_{t1} + B_{t2})(\dot{\theta}_t - \frac{\dot{x}}{r}) + (k_{t1} + k_{t2})(\theta_t - \frac{x}{r}) + (k_{a1} + k_{a2})(\theta_t - \frac{\theta_s}{N_s N_d}) = 0 \tag{7}$$

$$M\ddot{x} + \frac{(B_{t1} + B_{t2})}{r}(\frac{\dot{x}}{r} - \dot{\theta}_t) + \frac{(k_{t1} + k_{t2})}{r}(\frac{x}{r} - \theta_t) = -f_r Mg - 0.047 C_d A_f \dot{x}^2 - Mg\sin\theta \tag{8}$$

where x is the displacement of the vehicle, θ_m is the angular displacement of the engine, θ_s is the angular displacement of the gear, θ_t is the angular displacement of the wheel and $f(R)(\dot{\theta}_m - \dot{\theta}_s)$ represents the torsional effect of dry friction at the clutch. The Dry Friction Model is non-linear and given in Figure 3.

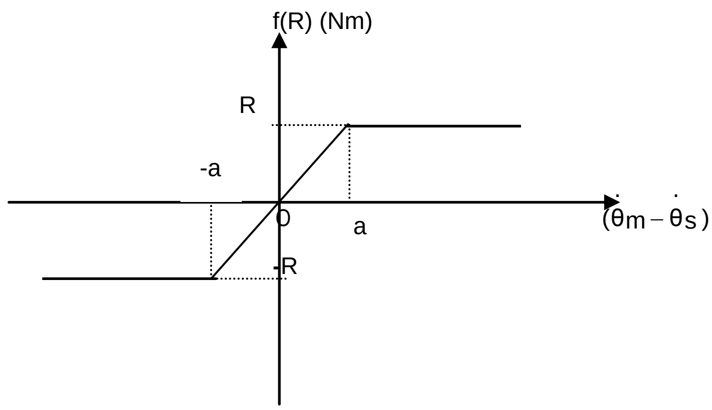

Figure 3. Dry Friction Model (Parameters are given in Appendix).

3. Simulation

Since the start of the vehicle is critical for longitudinal vibrations, the vehicle parameters are selected for the first shift and a constant 40 N.m engine torque is applied. The results are obtained for zero grade road though model is applicable for roads having slope. Under these conditions angular displacement at the clutch is plotted in Figure 4.a and angular displacement at the axles are given in Figure 4.b. These oscillations result in back and forth vibration of the vehicle causing an uncomfortable ride during the start and low shift manoeuvres. Infact the angular speed of the engine is also oscillatory as shown in Figure 4.c. When realizing simulation Matlab with Simulink is used and Runge

Kutta is preferred as numerical integration method. The simulation results are agree with the conclusions in the literature (Hrovat and Tobler, 1991).

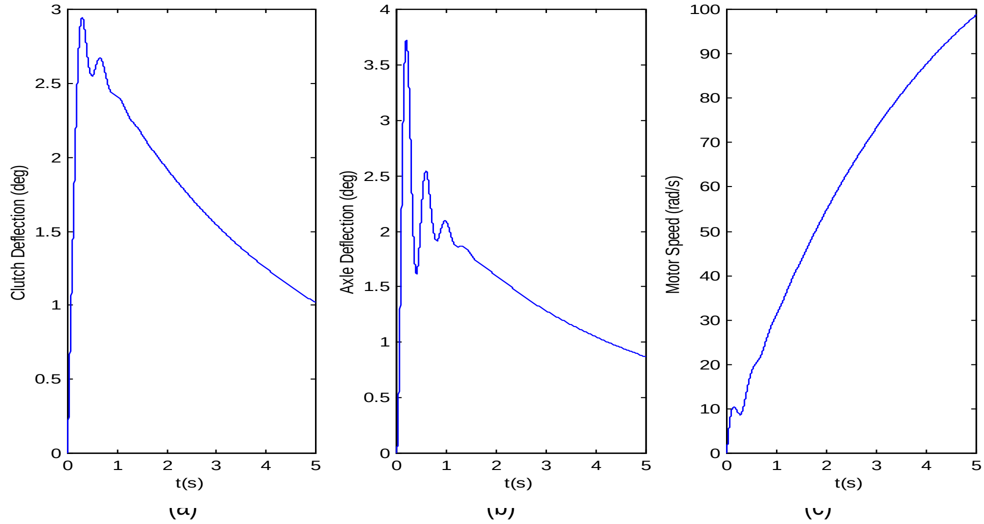

Figure 4.a) Angular Displacement of the Clutch. b) Angular Displacement of the Axle. c) Engine Shaft Angular Speed.

Figures 5.a , 5.b and 5.c show the frequency responses of the clutch deflection, axle deflection and vehicle speed for the linearized vehicle model. These plots give the main resonance frequency of the undesired back and forth vibrations.

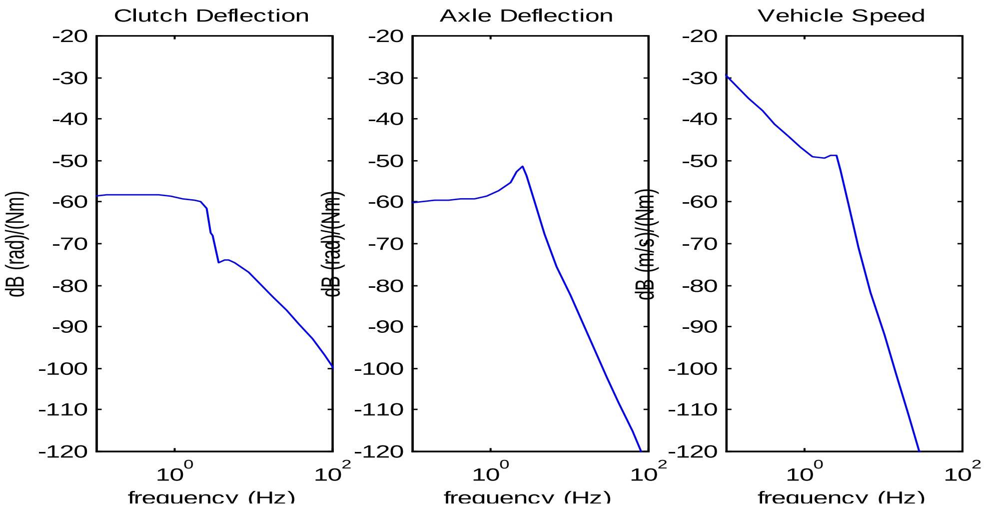

Figure 5.a, b, c) Frequency Responses of the Clutch Deflection, Axle Deflection and Vehicle Speed.

System has a resonance around 2.5 Hz as expected. Also, it is possible to obtain the speed performance of the vehicle. For the first shift and zero grade road, the simulation results are given in Figure 6.

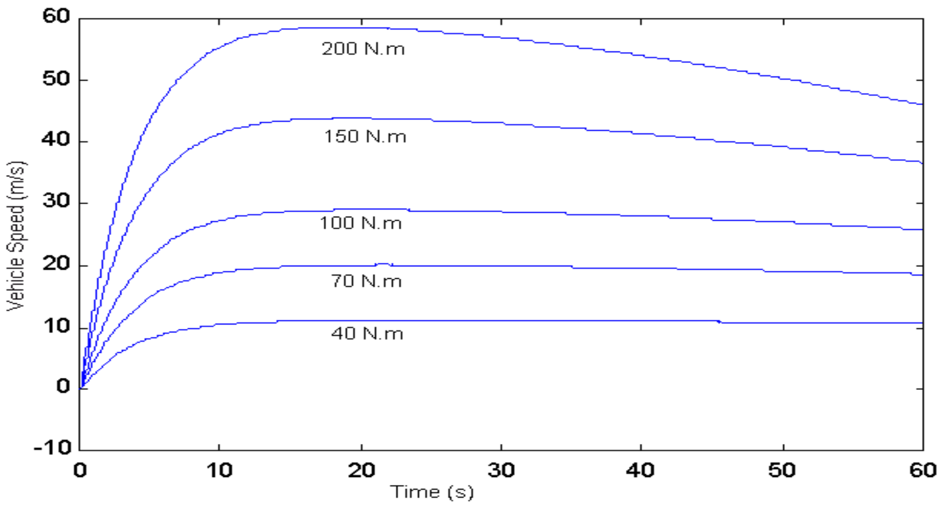

Figure 6. Speed Performance of the Vehicle at Different Engine Torques.

4. Conclusion

The flexible elements of power tra0nsmission systems causes undesired back and forth oscillations during the ride at first, second and rear shifts. This causes uncomfortable manoeuvres. In this study, the dynamics of this vibration has been studied on a non-linear power transmission model. Limited viscous band type rotational dry friction effect is also included to the model whereas all non-linear effects of the ride resistance were included. Back and forth oscillations are observed on transmission elements and vehicle engine speed at the time domain. On the linearized transmission model, frequency response of the system is obtained and resonance value of the back and forth vibrations are observed. Besides the dynamic model used is capable of giving the speed performance of the vehicle. This study is the first step in controlling the undesired back and forth vibrations and improving the ride comfort.

APPENDIX

Parameters

a :	3 rad/s	R :	200 Nm
T_m :	40 Nm	B_m :	0.271 Nms/rad
N_s :	3 (1st shift)	B_{t1} :	678.5 Nms/rad
N_d :	4	B_{t2} :	678.5 Nms/rad

I_m	:	0.136 kgm^2	k_k	:	680 Nm/rad
I_d	:	0.137 kgm^2	k_{t1}	:	679 Nm/rad
I_{t1}	:	0.5433 kgm^2	k_{t2}	:	679 Nm/rad
I_{t2}	:	0.5433 kgm^2	k_{a1}	:	4072 Nm/rad
I_s	:	0.147 kgm^2	k_{a2}	:	4072 Nm/rad
M	:	1400 kg	r	:	0.2745 m

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