

Identifying Electrically Assisted Steering Transfer Functions using a Modified FIR Filtering Approach

Kevin McLaughlin and Roger Lahoud

TRW Automotive

Miguel Bahena

Ford Motor Company

Copyright © 2009 SAE International

ABSTRACT

The transfer function required to control an electrically assisted steering gear is identified using a modified FIR filtering approach. The topology of the electrically assisted steering gear is described. The modified FIR filtering approach is explained with emphasis on the frequency domain implication of window choice when performing measurements. An example is provided using an analytic model of the steering gear so that the accuracy of the technique can be evaluated on a known system. The technique is applied to measure the transfer function of an actual electrically powered steering gear on a test bench and in a rolling vehicle.

INTRODUCTION

Predicting the dynamic response of an electrically powered steering gear is essential for controller design. Precise knowledge of input/output transfer functions allows optimization of the control system so that a positive steering experience can be provided to the driver. Phase information is especially critical in electric steering because the zero db crossover frequency is often higher than the natural frequency of the steering gear, meaning that phase compensation is required.

A challenge when measuring the transfer function is doing so without invading the closed system with cables and connections. Because the steering gear is in the engine compartment, the electronics are sealed. It is impossible to use external connections without

destructively damaging the natural sealing system of the gear. Having an on-board algorithm resident in the software of the steering gear electronics is extremely advantageous. Many methods exist to measure transfer functions but not all are amenable to this on-board requirement [1-2].

Stepped-sine wave excitation is a simple solution. A series of single sine measurements at the desired frequencies are used to excite the system. A waiting period is induced between the sine waves to allow the transients to decay. An advantage of this method is that the amplitude can be made a function of frequency, thereby minimizing the impacts of saturation near lightly damped modes. A major drawback of this method is the transients induced by the stepped waves: modes not of interest may be excited causing the system to experience nonlinearities due to hitting saturation limits.

Other well-known methods such as Chirps, Random Noise and Pseudo-Random Noise [1] can provide excellent results. However, these schemes are more complex with regard to processing the outputs. For an electrically assisted steering gear, these methods may require external systems connected to the electronics of the steering gear - a situation that must be avoided.

An alternate approach is presented where the excitation functions are coded on-board in the ECU. The computation of the gain and phase can also be coded in the ECU but for the present work, these computations are done off line. The proposed scheme modifies the

stepped-sine method. The excitation signals are sine waves multiplied by window functions, specifically a Blackman-Harris window. The flexibility in choosing the excitation frequencies and amplitudes is retained. The window functions are utilized to limit the frequency content of the commands and to reject noise in the measurements. The gain and phase of the output signal with respect to the input are computed as simple sums.

Transients in the system are avoided by using the window functions. It is shown that the application of the window function is equivalent to Finite Impulse Response filtering techniques. Software required is minimal and the technique is easily portable to other engineering systems.

Results for transfer function measurements are shown for an Electric Steering Gear on a test bench and in a rolling vehicle. The gear is a "belt-drive gear", consisting of a power rack-and-pinion gear and a motor coupled to the steering rack, with a belt and ballnut/ballscrew to provide steering assist. The simplified model of the system is a two-mass/two-spring system. It is shown that the measurements correlate well with the model.

DESCRIPTION OF AN ELECTRICALLY ASSISTED STEERING SYSTEM

A process is presented to identify the transfer function of an electrically assisted steering gear. A picture of an electrically assisted steering belt-drive gear is shown in Figure 1. The driver input torque at the handwheel is sensed with a torque sensor in the pinion. The torque is passed through a controller that computes the amount of assist required by the motor to turn the road wheels. The main transfer function of interest is the transfer function from motor command to sensed torque. All parameter values provided are for a specific application and can vary depending on the vehicle application.

From Figure 1, the steering wheel is connected via an intermediate shaft to the rack-and-pinion steering gear. The ratio of the rack-and-pinion is 50mm of rack travel for 1 revolution of the handwheel. The intermediate shaft connects to the steering gear via a torque sensor located in the pinion housing. The torque sensor has an internally compliant member termed a "torsion bar or T_{bar} " with a torque constant of 2 Nm/deg and a maximum travel of ± 5 degrees. The torsion bar travel is constrained with hard mechanical stops. The rack is connected via a ballnut/ballscrew/belt mechanism to the electric motor. The electric motor provides steering assist to the driver. The tie rods are connected to the steering arms and turn the vehicle road wheels.

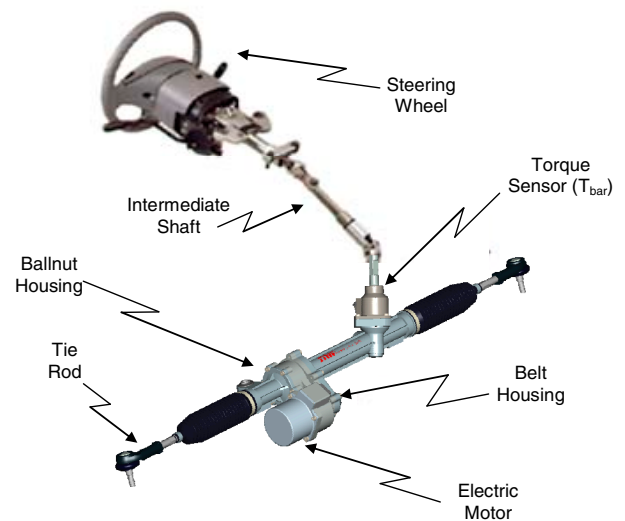


FIGURE 1: BELT-DRIVE EPS

An exploded view of the ballnut/ballscrew/belt mechanism is shown in Figure 2. As the rack moves back and forth the ballnut rotates. The ratio of the ballnut is 7mm of rack travel for 1 revolution of the ballnut. A large pulley is attached to the ballnut and via a belt to a small pulley on the motor rotor. The belt/pulley arrangement provides a 2.5 ratio, i.e. the motor rotates 2.5 times every time the ballnut rotates once.

The overall ratio from the motor to the handwheel is $2.5 \times (50/7) = 17.85$, i.e. for one revolution of the handwheel, the motor rotates 17.85 times. From a torque point of view, 1Nm of torque at the motor provides the equivalent of 17.85Nm of torque at the handwheel. The ballnut/belt amplifies the motor torque when viewed at the handwheel. Typical driver torques in power steering applications are under 4Nm yet typical assist load requirements can be as high as 100Nm referenced to the handwheel. The assist mechanism, by amplifying the motor torque via the belt/ballnut mechanism, provides such assist levels.

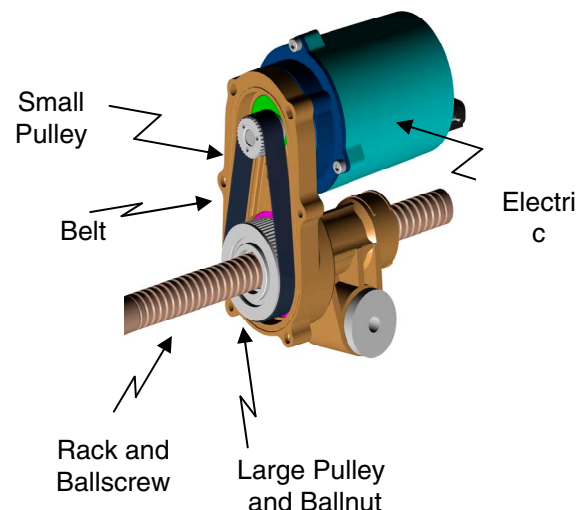


FIGURE 2: BALLNUT/BELT ARRANGEMENT

MODEL OF THE STEERING GEAR - Steering gears are typically modeled by lumping mass and compliance at different points [3]. In general, a two-mass, two-spring model captures the dynamics of interest to develop an adequate controller. A schematic of this model is shown in Figure 3.

The two-masses, one for the handwheel J_{HW} and one for the motor J_{motor} , are connected together by a spring K_{Tbar} representing the compliance between the handwheel and the motor. The compliance, K_{Tbar} , includes compliance due to the intermediate shaft, the torsion bar inside the torque sensor and compliance due to the ballnut/ballscrew/belt. This compliance is typically dominated by the torsion bar because it is very soft compared to the other items. The motor mass is connected to the ground by a spring representing the tire stiffness K_{tire} . Modeling the tire as a spring captures much of the dynamics required to study performance [4]. Both masses are attached to the ground with damping.

Typical parameters are shown in Table 1. As already discussed, the steering gear has various ratios due to the rack-and-pinion, ballnut/ballscrew and belt to provide amplification of the motor torque and to directly connect the driver to the road wheels. These ratios are not shown in this model for simplicity. All parameter values are referenced to the handwheel rotational frame. It is assumed that the frictions are viscous though coulomb friction is typically present in a steering gear.

Measuring the transfer function of the steering gear is addressed next. The model of Figure 2 is used to demonstrate the theoretical aspects of the system identification technique. It is desired to measure the transfer function from motor command to sensed torque. Sensed torque is the angular difference between the motor angle θ_{motor} and the handwheel angle θ_H times the torsion bar torque constant K_{Tbar} .

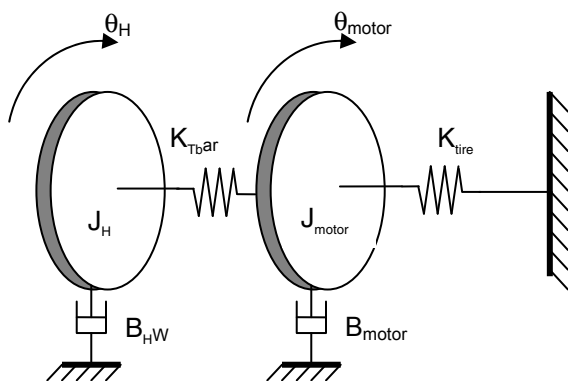


FIGURE 3: MODEL OF EPS STEERING GEAR

TABLE 1: MODEL PARAMETER VALUES

J_{HW}	0.04 kgm ²
J_{motor}	0.08 kgm ²
K_{tbar}	2 Nm/deg
K_{tire}	1 Nm/deg
B_{HW}	0.4 Nm*s/rad
B_{motor}	0.2 Nm*s/rad

THE APPROACH TO MEASURING THE TRANSFER FUNCTION

The proposed system identification approach uses a set of discrete sine waves each multiplied by a window function to measure the transfer functions of the steering gear. Windows can also be applied to the measured signals to reject noise. Windowing minimizes the frequency content of the excitation signal and also potential corruption of the measurement due to transients.

A general sine sweep method is described. Background information on windows, illustrating their advantages in signal processing, is presented. An example is shown that applies the modified sine sweep method to an analytic two-mass model. Finally, measurements of an electric steering gear are compared with analytical predictions.

SINE SWEEPS AND COMPUTING FREQUENCY CONTENT USING FOURIER INTEGRALS - Suppose there exists a time-domain signal $y(nT)$ where $n=1, 2, \dots, N$. To determine the frequency content at frequency ω the discrete Fourier integrals can be computed directly. First compute the sine and cosine integrals,

$$ns = \frac{1}{N} \sum_{n=1}^N y(nT) * \sin(2\pi\omega nT) \quad (1)$$

$$nc = \frac{1}{N} \sum_{n=1}^N y(nT) * \cos(2\pi\omega nT) \quad (2)$$

Typically, the number of points N is chosen to have an integral number of cycles. The magnitude M and the phase ϕ are computed as,

$$M = 2\sqrt{ns^2 + nc^2} \quad \phi = \tan^{-1}\left(\frac{ns}{nc}\right) \quad (3)$$