

Real-Time Powertrain Models of Hybrid Electric Vehicles

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ABSTRACT

Hybrid electric powertrains are developed for reducing the fuel consumption and consequently the emission of carbon dioxide. The fuel consumption of those systems depends significantly on the topology of the powertrain and the applied control strategy. For testing and improving the control strategy in terms of the overall system efficiency, Hardware-in-the-Loop (HIL) simulation can be applied. In order to operate the corresponding electronic control units in a virtual environment, it emulates the behavior of the actuator and sensor signals by a real-time simulation of the powertrain. Developing those powertrain models is a time-consuming task, due to high dynamics and high switching frequencies of the electrical subsystem and due to the large diversity of powertrain configurations. The electrical subsystem comprises the electric motor, inverter and battery whereas the different powertrain configurations are composed of the transmission, the combustion engine, the electric drives, etc.. Additionally, the correct power flow and losses within the powertrains components must be considered when HIL simulation is intended to be used for improving the control strategy.

In this contribution a method for modeling arbitrary combinations of clutches and various types of gears is presented which enables an efficient modeling of transmission systems. The obtained models are real-time capable and the correct power flows and losses are considered. The method is applied to a powertrain with a two mode power-split transmission, which is a typical example comprising electric drives as well as shifting clutches. Finally simulation results will be presented.

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INTRODUCTION

Electric (EV) and hybrid electric vehicles (HEV) are developed for improving the fuel consumption and reducing the emission of carbon dioxide. Today, HEVs are more common than EVs because of the cruising range and recharging/refueling time. In order to reduce the fuel consumption of HEVs the manufacturers try to improve the overall efficiency. It depends significantly on the powertrain's components, the configuration of the powertrain, and the overall control strategies implemented in electronic control units. For developing and testing electronic control units and improving the control strategies, Hardware-in-the-Loop simulation (HIL) can be applied. It requires a real-time model of the powertrain which is used to simulate the dynamic behavior of the powertrain and to emulate the sensor and actuator behavior of its components. The electronic control units are operated by means of a virtual plant. This virtual test environment lowers the costs and improves the test efficiency by enabling automated testing in a laboratory under repeatable conditions.

The overall scheme of a HIL simulation of a HEV is illustrated in [Figure 1](#), showing the real-time model and the most important electronic control units. The real-time model of the powertrain consists of several submodels, e.g. for the electric motor, the engine and the transmission, which are connected according to the topology of the powertrain [3]. Developing real-time models of hybrid electric powertrains is still an ambitious and time-consuming task, due to the high dynamics and high switching frequencies of the electrical subsystem (System A in [Figure 1](#)) on the one hand and the various configurations of the overall system and in particular the mechanical subsystem (System B in [Figure 1](#)) on the other hand. In order to improve the control strategy of HEV in terms of fuel/energy efficiency, it is also necessary to consider the correct power flows and losses within the powertrain's components.

This contribution is focused on modeling the mechanical subsystem comprising various transmission types. For the proposed HIL simulation, real-time capable models of the electric drive, the internal combustion engine and the battery are required, as well. However, these components are not objective of this contribution.

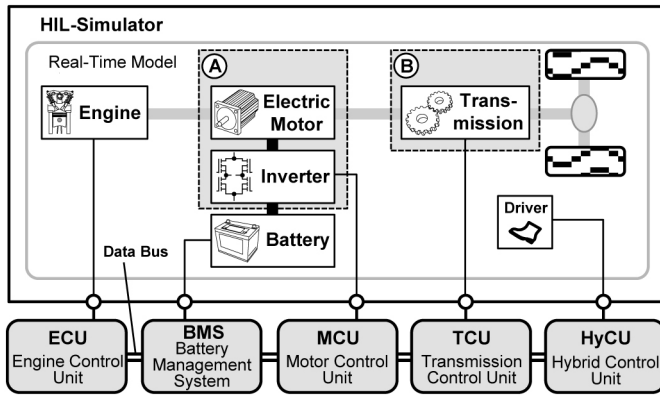


Figure 1. HIL simulation of a hybrid electric powertrain with the electric subsystem (A) and the mechanical subsystem (B).

The methods presented hereinafter enable an efficient modeling of arbitrary transmission configurations and the consideration of correct power flow and losses.

TRANSMISSION MODELS

In HEV, a multitude of different transmission types can be utilized. In power-split transmissions e.g., planetary gears are used to split up the mechanical power, generated by the combustion engine, into a mechanical and an electric part [3]. The gear ratio can be adjusted by changing the ratio of mechanical to electric power. In order to improve the efficiency of power-split transmissions, clutches can be used to switch between different power-split modes or modes with a fixed gear ratio. Other transmission types for HEV can be found in [5]. Generally, the transmission systems used in HEV and also modern automatic transmissions consists of complex combinations of planetary gears, spur gears, and clutches.

In this section, a modeling approach for the mechanical subsystem is introduced, which enables the modeling of various transmission systems, incorporating clutches and different types of gears. First modeling methods for gears and clutches are presented and thereafter a mechanical network analysis is briefly described which is used to obtain the overall transmission model. Finally, for achieving real-time capability, appropriate methods for modifying the system equations are presented.

Gears

A method for modelling arbitrary gear structures is presented in [6]. It includes simple spur gears as well as different types of planetary gears. Complex spur gear structures can be modeled by combining elementary spur gear subcomponents consisting of one gear pair, Figure 2.

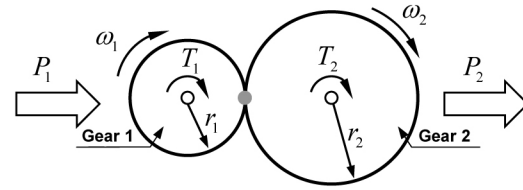


Figure 2. Elementary subcomponent of a spur gear.

The angular velocity and torque equation of a gear pair can be written as

$$\omega_2 = -\frac{\omega_1}{i} \quad \text{with } i = r_2/r_1 \quad (1)$$

and

$$T_2 = T_1 \cdot i \quad (2)$$

with the radii of the gears r_1 and r_2 . When considering power losses Eq. (2) is not valid, since it describes the ideal torque transmission, only. Modifying Eq. (2) yields,

$$T_2 = \begin{cases} i \cdot T_1 \cdot \eta_{21}, & P_1 > 0 \\ i \cdot T_1 \cdot 1/\eta_{12}, & P_1 < 0 \end{cases} = i \cdot T_1 \cdot \eta(T_1, \omega_1) \quad (3)$$

with the efficiency

$$\eta_{X,Y} = \frac{-P_X}{P_Y} = \frac{-T_X \cdot \omega_X}{T_Y \cdot \omega_Y} \quad (4)$$

where X and Y are placeholder for the gear 1 and 2. Since the definition of the efficiency depends on the direction of the power flow, it changes its value according to the sign of P_1 . The dependency on the sign of the mechanical power can also be considered as a dependency of the signs of T_1 and ω_1 . Therefore, the efficiency in Eq. (3) is written as a function of T_1 and ω_1 . This approach also allows considering friction of the gear pair and the bearings, since friction can be described by the load torque and the angular velocity of the gear [7]. Eq. (3) is modified according to

$$T_2 = i \cdot T_1 - i \cdot T_1 \cdot [1 - \eta(T_1, \omega_1)] = i \cdot T_1 - \Delta T(T_1, \omega_1) \quad (5)$$

where ΔT describes the torque losses. This enables us to consider losses by a torque source within the mechanical network analysis presented in the third subchapter. For reducing the calculation effort of the overall transmission model, the torque losses ΔT are considered by a map instead of a friction model.

Simple planetary gears, as well as more complex planetary gears (e.g. a Ravigneaux planetary gear set) can be modelled by combining elementary subcomponents which consist of one carrier and one gear pair [8]. Figure 3, shows a simple planetary gear composed of a ring-planet subcomponent and a sun-planet subcomponent sharing the same carrier and planets.

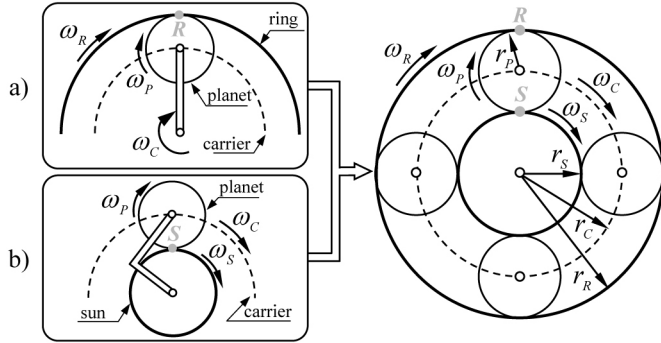


Figure 3. Elementary subcomponents of a simple planetary gear: a) ring-planet subcomponent and b) sun-planet subcomponent.

For each subcomponent an equation for the angular velocity can be obtained by utilizing the general plane motion of the point R and S, respectively. Applying this method to both subcomponent yields

$$\omega_p + (i_{RP} - 1) \cdot \omega_C - i_{RP} \cdot \omega_R = 0 \quad \text{with } i_{RP} = r_R / r_p \quad (6)$$

and

$$-\omega_S + (1 + i_{SP}) \cdot \omega_C - i_{SP} \cdot \omega_p = 0 \quad \text{with } i_{SP} = r_p / r_S \quad (7)$$

where the constant gear ratios i_{RP} and i_{SP} depend on the radii of the ring gear r_R , the sun gear r_S , and the planet r_p (number of teeth in real gears). The torque of the ring and the sun gear can be obtained by using the torque equilibrium of the gears and considering the torque losses analogously to the spur gears

$$T_R = -i_{RP} \cdot T_p + \Delta T_{RP}(T_p, \omega_p^*), \quad (8)$$

$$T_S = \frac{1}{i_{SP}} \cdot T_p - \Delta T_{SP}(T_p, \omega_p^*) \quad (9)$$

where the relative angular velocity ω_p^* can be obtained by

$$\omega_p^* = \omega_p - \omega_C. \quad (10)$$

Analogously to spur gears, the torque losses ΔT are considered by using appropriate maps, which describe ΔT in dependency of the relative angular velocity ω_p^* and the applied torque T_p . Each subcomponent has its own friction torque map ΔT concerning the corresponding gear pair. Usually, sliding friction causes the major part of the overall torque losses in transmissions [9], i.e. the major part of the torque losses can be expressed by means of a torque and an angular velocity. For considering other types of losses, such as windage losses, churning losses or rolling friction losses, dependencies on other values (e.g. angle and temperature) might occur whereby the overall transmission model becomes more complex and real-time capability will not be possible. The map ΔT can be obtained by measurements, an offline model or it can be calculated by using a constant efficiency value in combination with Eq. (3), (8) or (9).

For an efficient modelling of arbitrary gear structures the presented subcomponents are combined. The mechanical network analysis, presented in the third subchapter, is used to generate the overall system equations of the combined gear subcomponents.

Clutches

Clutches are important components, since they are used to switch between different gear ratios, i.e. the transmission ratio of the engine speed to the speed of the output shaft can be adjusted. Figure 4a shows a schematic diagram of a clutch [10]. When a normal force F is applied, a friction torque T_f occurs which is exploited to transmit torque from one shaft to the other. The behavior of a clutch depends significantly on the friction torque T_f

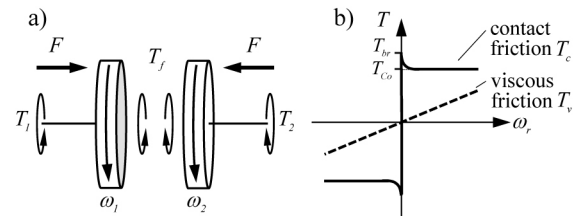


Figure 4. a) Schematic diagram of a clutch and b) general friction model.

In Figure 4b, a general friction characteristic is shown which combines two types of friction: contact friction T_c (combining static, stribeck and coulomb friction) and viscous friction T_v . The contact friction can be calculated according to

$$T_c = \mu(\omega_r) \cdot r_{eff} \cdot F \quad (11)$$

where $\mu(\omega_r)$ is the friction coefficient depending on the relative angular velocity

$$\omega_r = \omega_1 - \omega_2,$$



r_{eff} the effective radius of the clutch and F the applied normal force. Since the clutches in transmissions are usually actuated hydraulically, the normal force is a function of the hydraulic pressure applied by the oil pump and controlled by valves and electronics.

The normal force depends also on the design of the clutch. In [10] the calculation of the normal force for a multi-plate clutch is presented. When wet clutches are used, viscous friction

$$T_v = \nu \cdot \omega_r \quad (13)$$

occurs, where ν is depending on the viscosity of the lubricant and the geometry of the clutch. The overall friction torque T_f can be obtained by

$$T_f = T_c + T_v. \quad (14)$$

Using the shown friction characteristic for real-time simulation is challenging since it is nonlinear and not unique at zero angular velocity. An overview of different clutch models for real-time simulation is given in [11]. Most of those approaches approximate the ideal friction behavior by a continuous function.

Considering coulomb and viscous friction only and inverting the characteristic shown in Figure 4b, the new friction characteristic shown in Figure 5a is obtained. It describes the relative angular velocity ω_r in dependency of the friction torque T_f . If the transmitted friction torque T_f exceeds the applied coulomb friction limit T_{Co} , the clutch slips with the relative angular velocity

$$\omega_r = \frac{1}{\nu} \cdot T_v. \quad (15)$$

Otherwise, the relative angular velocity becomes zero. The coulomb friction

$$T_{Co} = \text{sign}(T_f) \cdot \mu \cdot r_{eff} \cdot F \quad (16)$$

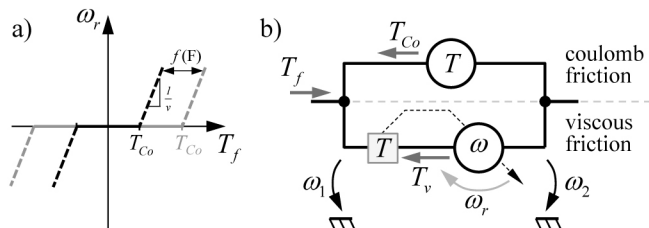


Figure 5. a) New characteristic of the friction torque b) network representation of a clutch.

is proportional to the normal force F , where μ is a constant friction coefficient. If no normal force is applied, the clutch is deactivated and the characteristic shown in Figure 5a will be equal to a continuous linear function with zero-crossing at zero torque. Otherwise, the linear function in the friction characteristic will shift proportional to the applied normal force.

Figure 5b shows the network representation of a clutch. The network representation is similar to electric circuits, where torques are equal to currents and angular velocities equal to voltages. For the clutch it consists of a torque source representing the coulomb friction T_{Co} and a torque controlled angular velocity source representing the relative angular velocity ω_r . These sources are connected to each other via two nodes where the torques are added according to Eq. (14).

Calculating the coulomb friction for the torque source requires the evaluation of the sign function in Eq. (16). Inserting Eq. (14) in Eq. (16) yields

$$T_{Co} = \text{sign}(T_f) \cdot \underbrace{\mu \cdot r_{eff} \cdot F}_{|T_{Co}|} = \text{sign}(T_{Co} + T_v) \cdot |T_{Co}|. \quad (17)$$

Equation (17) is implicit since T_{Co} appears on both sides of the equation. Replacing the coulomb friction torque by its absolute value multiplied by the algebraic sign $a_{T_{Co}}$ and rearranging the variables yields

$$0 = \text{sign}(a_{T_{Co}} \cdot |T_{Co}| + T_v) - a_{T_{Co}} \quad (18)$$

where the only unknown is the algebraic sign $a_{T_{Co}}$. Since in the presented approach the overall friction torque T_f becomes a state variable, the algebraic sign can be calculated directly according to

$$a_{T_{Co}} = \text{sign}(T_f). \quad (19)$$

When the overall model contains more than one clutch T_f is vectorial and Eq. (19) must be evaluated for each entry of the vector.

The relative angular velocity source in Figure 5b can be calculated by using Eq. (15)

$$\omega_r = \begin{cases} \frac{1}{\nu} \cdot T_v & , c_s = 0 \\ 0 & , c_s = 1 \end{cases} = F(c_s) \cdot T_v \quad (20)$$

where c_s denotes the clutch states and $F(c_s)$ is a switchable coefficient. If c_s is zero, the clutch is in the disengaged and slip state, respectively.