Bond Graph Modeling of Automotive Transmissions and Drivelines

Joško Deur*, Vladimir Ivanović, Francis Assadian**, Ming Kuang***, Eric H. Tseng***, Davor Hrovat***

*University of Zagreb, Faculty of Mechanical Engineering, Ivana Lučića 5, 10002 Zagreb, Croatia (Tel: +358-6168-372; e-mail: josko.deur@fsb.com, vladimir.ivanovic@fsb.hr).
** Cranfield University, School of Engineering, Department of Automotive Engineering, Bedfordshire MK43 0AL, UK, (email: f.assadian@cranfield.ac.uk).
*** Ford Research and Innovation Center, 2101 Village Rd., Dearborn 48121, USA, (e-mail: mkuang@ford.com, htseng@ford.com, dhrovat@ford.com).

Abstract: The paper presents an overview of the bond graph models of advanced automotive transmission and driveline systems. This includes one-mode and two-mode series-parallel hybrid electric vehicle transmissions, a continuous variable transmission, active-limited slip and torque vectoring differentials in 2WD and 4WD configurations, and electromechanical actuator-based wet and dry clutch actuation systems. It is illustrated that the bond graph method can be effectively used to gain valuable insights about the system dynamics structure and behavior.

Keywords: Transmissions, drivelines, hybrid vehicles, clutches, bond graph modeling, analysis.

1. INTRODUCTION

In the last two decades there have been substantial advancements in design and application of automatic transmissions and drivelines, including new designs such as hybrid electric vehicle (HEV) transmissions, dual clutch transmissions (DCT), continuously variable transmissions (CVT), and torque vectoring differentials (TVD) and four-wheel drive (4WD) systems.

Among different HEV transmission solutions, the series-parallel configurations have found significant applications in either one-mode variant (e.g. Toyota Prius and Ford Escape; (Miller, 2006; Kuang and Hrovat, 2003)) or two-mode configuration (GM SUV/truck applications; (Holmes and Schmidt, 2002)). These transmissions are sometime called e-CVTs, as they attempt to realize the CVT effect of keeping the engine in its optimal/efficient speed region by mean of use of electrical machines. The torque vectoring differentials can provide the left/right or front/rear torque transfer, thus significantly improving the vehicle traction control and vehicle dynamics control performance (Sawase and Sano 1999). The TVD functionality can be incorporated into a 4WD system, as proposed in (Honda Motor Co., 2005). The DCTs combine the comfort of conventional torque converter automatic transmissions and the fuel economy, the sporty, responsive and connected drive feel of manual transmissions (Matthes, 2005).

Along with the new automatic transmission and driveline designs, there have been significant efforts in improving and re-designing the common transmission actuation elements - the friction clutches, either conventional wet or more recent dry clutches. In particular, the emphasis has been on developing modular and more efficient electromechanical actuation systems for the clutches (Wheals et al., 2007; Wagner et al., 2009).

The new transmission and driveline solutions have often imposed new requirements on the quality of power train control, which typically strongly depends on the fidelity of related transmission and driveline dynamics models. In related previous publications (Hrovat and Tobler, 1991; Hrovat et al., 2000), it has been demonstrated that the bond graph method can be an effective tool for unified, modular, and minimum-realization physical modeling of power train component and systems, and their insightful analyses. Examples included an efficient, generic model of automatic transmission (AT) shifts for both layshaft and planetary ATs, and an effective and insightful model of drivetrain shuffle mode dynamics. In the meantime, this work has been extended with modeling, analysis, and control of the aforementioned, more recent transmission and driveline systems. This paper presents an overview of the bond graph models of these systems, including modeling of clutches and their actuators. Due to the paper size constraints, the model equations, detailed analysis results, and validation results are omitted, but they can be found in the authors’ references cited.

2. HYBRID ELECTRIC VEHICLE TRANSMISSIONS

Fig. 1 shows kinematic schemes of the considered one-mode and two mode series-parallel HEV transmissions (1MHT and 2MHT). The 1MHT pairs an internal combustion engine (ICE) with an electric machine M/G1 through a planetary gear (PG) used as a power split device. The M/G1 machine is used as a generator or a motor to keep the engine in the
optimal speed operating region. The second electrical machine M/G2 is connected to the transmission output shaft, and is normally used as a motor or a generator to meet the driver’s torque demand at the wheels during propulsion and braking.

The 2MHT introduces an additional PG and a pair of clutches (F2 and F3; F1 is normally locked), in order to keep the engine at the optimal point in a wider range of vehicle velocities and boost the transmission output torque at higher velocities. In the first, low-mid velocity mode (F2 = OFF, F3 = ON) the 2MHT has a similar structure as the 1MHT PG, because the M/G2 machine is only coupled to differential through the PG2 with locked ring gear. In the second, high-velocity mode (F2 = ON, F3 = OFF), clutch F2 connects the two PGs and makes the M/G2 machine speed dependent on the M/G1 speed, in addition to the output speed $\omega_{cd}$.

Fig. 2 shows the bond graph models of 1MHT and 2MHT kinematics for the electric variator mode, where the two electric machines are assumed to be connected to each other through the inverters and a common dc link between the inverters (there is no battery). The models are derived and the below bond graph-based (graphical) power flow analysis is presented in more detail in (Cipek et al., 2010, 2012). The dynamic models of these transmissions are presented in (Deur et al., 2010).

The 1MHT in Fig. 2a operates in four characteristic regimes: (i) series-like low-velocity mode where the majority of power is transferred through electric variator path giving a lot of torque $\tau_{mg2}$ at small output speed $\omega_{d2}$ (note: the mechanical path power $\tau \times \omega_d$ is small at low $\omega_d = \omega_{d1}$); (ii) parallel-like mid-velocity mode where both electrical and mechanical power paths are active; (iii) purely mechanical mode when the M/G1 speed and power drop to zero due to the rising output speed $\omega_d$ and the constant/optimal engine speed (≈ 2400 rpm); and (iv) negative recirculation of electric power due to $\omega_{mg1} < 0$ and $\tau_{mg1} = \tau / (h+1) > 0$ including “amplification” of mechanical path power (note that M/G1 is then motoring and M/G2 becomes a generator).

Comparison of Figs. 2a and 2b confirms that the 2MHT behavior in Mode 1 (when TF:h2 path is omitted) is similar to that of the 1MHT. The appearance of the additional mechanical path over the second PG element TF:h2 in Mode 2 (high velocities) results in the following advantages: (i) instead of being proportional to the growing vehicle velocity (as in the 1MHT), the speed $\omega_{mg2}$ does fall from the mode changing point (more M/G2 torque can be obtained); and (ii) the M/G2 torque can support M/G1 torque in allowing higher engine output torque. This results in a torque boost, a wider range of optimal engine operation, and a reduced electric recirculation (Cipek et al., 2010, 2012).

Fig. 3 shows the 1MHT model for the full hybrid mode, with an ideal battery assumed (no power losses). The behavior of such a transmission is similar to the basic electric variator transmission, with the following main differences: (i) at low velocities the M/G1 power can be increased to fill the battery and extend the optimal engine operation to lower velocities; (ii) in the mid velocity range the battery can provide additional power for supplying the M/G2 motor for a torque boost (for the same reason, there is no purely mechanical mode); and (iii) in the high-velocity range, the M/G1 is...
motoring like in the variator mode ($P_{ng1} < 0$), but there is no real electric power recirculation since the M/G2 is motoring, as well, based on the battery power supply.

3. CONTINUOUSLY VARIABLE TRANSMISSION

The main parts of the CVT include two V-shape pulleys connected by a metal belt, as illustrated in Fig. 4a. Power is transmitted by compression of the individual segments of the metal belt rather than tension as is done with rubber belts. To change the transmission ratio (the ratio between the secondary pulley radius to the primary pulley radius), the movable halves of each pulley are axially adjusted under hydraulic control.

The geometrical quantities of the pulley are shown in Fig. 4b. In terms of these quantities, before constructing the bond graph of the pulley/belt, the kinematic relationship between the pulley radius, $R$, and pulley angle, $\alpha$, has to be derived. Then by differentiating this equation, a relationship between the belt radial velocity, $v_R$, and the axial pulley velocity, $v_P$, is obtained. In order to account for frictional losses between the belt and the pulley, the kinematic relationship between the relative speed of the belt with respect to the pulley is derived. The above kinematic relationships are used to construct the bond graph fragment of the pulley/belt, as illustrated in Fig. 5a. In this bond graph, a coulomb friction model is utilized. However, if desired, this model can be expanded to include more sophisticated friction models such as a model based on elastohydrodynamic theory. The bond graph in Fig. 5a contains an inertia element ($Lm_{bel}$) which accounts for the mass of the pulleys and the belt. Furthermore, for simplicity, we ignored the dynamics associated with the belt.

The setup for the hydraulic control of the pulleys is known as the master-slave control. The primary pulley (master) supply pressure adjusts the pulley ratio, and the tension in the belt is kept large enough by the secondary pulley (slave) supply pressure for the power transmission. The control objective is satisfied by controlling the supply pressure to the primary pulley through a primary valve, and controlling the secondary pressure through a simple pressure reducing valve. The bond graph fragment for the hydraulic control system is illustrated in Fig. 5b. This bond graph model includes the dynamics of the spool, hydraulic line, and primary and secondary pistons. The primary valve is used in an open loop control configuration. Therefore, the plunger stroke is assumed to be prescribed and not controlled, and the primary valve pressure feedback signal is ignored.

4. ACTIVE DIFFERENTIALS

There are two basic types of active differentials: (a) Active Limited Slip Differentials (ALSD) and (b) Torque Vectoring Differentials (TVD). Figs. 6a and 6b show kinematic schemes of the typical ALSD and a common type of TVD. The corresponding kinematic bond graph models are shown in Figs. 6c and 6d (Deur et al., 2008a). When the clutch $F$ is disengaged ($\tau_f = 0$), the path from $\omega_1$ to $\omega_2$ junction points 1 in Fig. 6c is omitted, and the ALSD reduces to the passive differential where the output (wheel) torques are equal ($\tau_1 = \tau_2$) for generally unequal wheel speeds $\omega_1$ and $\omega_2$. Engaging
the clutch generates the clutch friction torque $\tau_c$, whose sign corresponds to the sign of clutch slip speed $\omega_f = \omega_c - \omega_2 = (\omega_1 - \omega_2) / 2$ Thus, for e.g. $\omega_3 < \omega_1$, the clutch takes the torque $\tau > 0$ from the rotating case (or effectively from the faster wheel 1) and bring it to the slower wheel 2. If $\omega_1 < \omega_2$, $\tau < 0$ holds, the torque is again transferred from the faster wheel (this time W2) to the slower wheel (W1). Based on the model in Fig. 6c, in both cases the faster-to-slower wheel torque transfer is given by $(\tau_2 - \tau_1) / 2 = \tau / 2$.

The TVD (Fig. 6b) includes two clutches, F1 and F2, and additional gearing with effective gear ratios $h_1 < 1$ and $h_2 > 1$, respectively. The model (Fig. 6d) includes two torque transfer paths over the two clutches. Clutch F2 always transfers the torque to W2, and this can be done even if $\omega_2 > \omega_1$, provided that $\omega_2 > 0 \Rightarrow \tau_2 > 0$, which gives $\omega_2 < h_2(2-h_2)^{-1}\omega_1$, i.e. $\omega_2 < 1.286\omega_1$ for $h_2 = 1.125$. Similarly, clutch F1 always transfers the torque to W1, provided that $\omega_1 > 0 \Rightarrow \tau_1 > 0$, which yields $\omega_1 < h_1^{-1}(2-h_1)\omega_2 = 1.286\omega_2$ for $h_1 = 0.875$. Based on the model in Fig. 6d, the torque transfer is found to be $(\tau_2 - \tau_1) / 2 = (\tau_2 - \tau_1) / 2$. Bond graph modeling of several other TVD configurations is presented in (Deur et al., 2008a) and (Milutinović and Deur, 2009).

Fig. 7b shows the bond graph model of a 4WD TVD vehicle, which combines the passive front differential and a rear TVD shown by the kinematic scheme in Fig. 7a. When operating in the FWD+TVD configuration, the central coupling device speed ratio is somewhat larger than one ($g > 1$), thus speeding up the rear axle and allowing for the torque transfer to W1 by engaging clutch F1 and to W2 by engaging clutch F2. In the 4WD configuration, the central coupling gear ratio is switched to $g = 1$ (by appropriate clutches, not shown in Fig. 7), and both rear TVD clutches are locked (rear TVD operates in a similar manner as a locked ALSD).

The model in Fig. 7b includes the dynamic effects of wheel and gear inertia. The causality rules show that for the common torque input causality the model is of fourth order, with the state variables associated with the wheel speeds. Similar dynamic models can be developed for the active differentials in Fig. 6 (second-order models), and they can be extended with halfshaft compliance effects (Hrovat et al., 2000). These dynamic models are given, in both bond graph and mathematical forms, in (Deur et al., 2008b) and (Deur et al., 2010).

5. ELECTROMECHANICAL CLUTCH ACTUATOR

Fig. 8 shows the kinematic scheme and corresponding bond graph for an active differential wet clutch electromechanical actuator (Ivanović et al, 2011a). A geared permanent magnet dc motor is used to engage the clutch through the ball-ramp mechanism that converts the motor torque $M_m$ into the clutch normal force $F_n$ by squeezing out the oil and compressing the clutch plates. The ball-ramp mechanism consists of the input and output disk with oppositely arranged grooves with defined slope (ramp), and balls placed in the grooves. The input disk is driven by the motor. The output disk is
connected to the pressure plate and its rotation is constrained by a spring-damper system fixed to the differential housing allowing for linear and small rotational motion during the clutch disengagement. A preloaded coned-disk return spring $k_{rs}$ provides passive clutch opening and proper mechanism operation. In the bond graph model (Fig. 8b), the geared motor model is standard. The ball-ramp system is modeled by the transformer $TF:ibr^0$, associated with transformation of the input disk torque $M_{br}$ to the ball-ramp force $F_{br}$, the elastic element $k_{br}$ reflecting the ball axial compliance, the friction due to ball deformation $R_{fr.br}$ and bearings $R_{fr.nb}$, the nonlinear return spring $k_{rs}$, and the limiter $k_{lim}$. Note that the bearing friction is cross-influenced by clutch rotational dynamics through the differential case speed $\omega_{dc}$ through the friction in the needle bearings $R:R_{nb,1,2}$. The fluid film dynamics and the clutch axial compliance define the pressure plate force $F_{pp}$. The first principal model includes the fluid squeeze dynamics ($c_{fluid}$, $k_{roughness}$) and the clutch compression dynamics ($c_{cl}$, $k_{cl}$). It can be effectively simplified by experimentally determined effective damping $c_{cl.e}$ and nonlinear compliance $k_{cl.e}$ with included clutch freeplay zone.

Fig. 8 also shows how the actuator dynamics can be coupled with the rotational and thermal dynamics submodels resulting in the full clutch model. The rotational model is given in its simplified form and can be easily extended, as shown in (Ivanović, 2011a). The thermal model has a lumped-parameter form that describes the thermal power flows/heat transfers between the clutch plates $T_c$, the differential case $T_{dc}$, and the oil $T_{oil}$.

As an alternative to the ball-ramp system, the clutches can be actuated by electromechanical lever-based actuators. Fig. 9 shows the kinematic scheme and corresponding bond graph of such an actuator for a dry dual clutch application (Ivanović et al, 2011b). The lever is constrained by the fulcrum roller
and the leaf spring allowing for linear displacement ($x_l$) and rotation around the fulcrum $\alpha_l$. As such it balances torques associated with the engagement bearing force $F_b$ and the force of initially preloaded linear energy source spring $F_s$. The force $F_b$ corresponds to the reactive clutch force related to the clutch normal force, as explained in (Ivanović et al., 2012). This force is controlled by simultaneously varying the lever arms ($l_1$ vs. $l_2$) and pushing the lever relative to the clutch by means of the fulcrum roller driven by a brushless dc motor ball-screw actuator. The required motor torque $M_m$ is equal to the sum of the motor inertial torque ($J_m$), the lever reactive force $F_l$, and the friction in the screw-drive and the fulcrum rollers reflected to the motor shaft $R_{fr-c}$. In the model, the fulcrum roller speed $v_f$ and the lever lift speed $v_l$ are coupled through the modulated transformer MTF in the fulcrum dynamics with module related to the lever-fulcrum roller contact angle. The lever rotation model reflects the torque balance applied to the lever with three modulated transformers MTF associated with the lever arms variation. The system compliance is reflected through the effective lever compliance $C_{l1}^{-1}$.

6. CONCLUSION

The bond graph method has been found to have many advantages in modeling and analysis of automotive power trains. They include: (i) direct representation of a complex transmission/driveline mechanical system with a graph that can be converted into static or dynamic model equations in a straightforward way; (ii) modularity in terms that the developed submodels of the system elements (e.g. planetary gears) and components (e.g. clutches) can be readily interfaced into a more complex system model; (iii) derivation of minimum-realization dynamic models for computationally efficient computer simulations, based on application of straightforward causality rules that "eliminate" dependent variables which often appear in transmission mechanical subsystem; (iv) simplicity of modeling in the multi-physical domain, as demonstrated on the example of wet-clutch active differential model describing the mechanical, electrical, fluid, and thermal effects; and (v) a unique and illustrative way of conducting graphical power flow analyses including relations between effort (torque) and flow (speed) variables, which was particularly effective in the case of HEV transmissions.

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