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Model reference control of conceptual clutched train substituted for vehicular friction clutch

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Abstract. In order to essentially reduce heat generation and frictional dissipation carried by friction clutch engagement, conceptual design of clutched train combined with hydrostatic braking system is proposed as a novel substitution for vehicular friction clutch. Potential collateral merits of clutched train may improve service life and control accuracy since less friction heat generated during synchronization process. Parameter of clutched train is obtained by Genetic Algorithm optimization aiming at axial space-saving and light weight. Control-oriented model of proposed concept is derived and used in Model Reference Control development. Based on optimum parameter of clutched train, simulation result has shown the functionality of clutched train on vehicle standing-start, and well-behaved Model Reference Control on smoothing clutched train synchronization process.

1 Introduction

Friction clutch is known as a key mechanical connection and disconnection between an engine and a transmission in a car. The basic function of clutch is supposed to transfer power from engine to vehicle with desired smoothly and efficiently. However, owing to it transmitting torque through friction, some issues particularly in terms of friction dissipation [1] and shift comfort [2-4] are inevitable. For example in case when significant heat generation during slipping phase, high nonlinear fluctuation of friction coefficient [3, 4] would make it difficult to give an accurate signal to actuator for the estimation of transmitted clutch torque.

As a branch of planetary gearset application, concept of clutched train with hydraulic system is proposed in this paper to substitute for friction clutch used in a car and mated to automated mechanical transmission to ensure smooth transfer of torque with a little heat generated by gear-meshed sliding friction [5]. Note that less heating is associated with improving mechanical life, as well as more accurate control due to little variation on friction coefficient. Clutched train utilizes the alternation between one degree-of-freedom (DOF) and two DOFs of rotary motion to determine the output instead of friction. The primary advantages of proposed concept lie in avoiding high non-linearity and significant frictional dissipation with engagement, and in return improving the control veracity in practice.

This paper would present the conceptual design and application of Model Reference Control (MRC) algorithm on clutched train synchronization process control. Simulation results show that the functionality of conceptual design is to connect and disconnect engine and transmission, and the designed MRC could work with conceptual design to reduce vehicle jerk at vehicle standing-start.

2 Conceptual design

As shown in Figure 1, the configuration of conceptual design is comprised of a symmetrical compound planetary gearset and a hydraulic system. The compound planetary gearset can be considered as a combination of single stage Planetary-Gearset-I (PGS-I) and Planetary-Gearset-II (PGS-II), which have the same parameters and share one arm. The PGS-I ring gear is fixed. The PGS-I sun gear is coupled with the driving shaft, which is driven by the output shaft of an engine. The PGS-II sun gear is coupled to a coaxial output shaft which drives an automatic transmission.

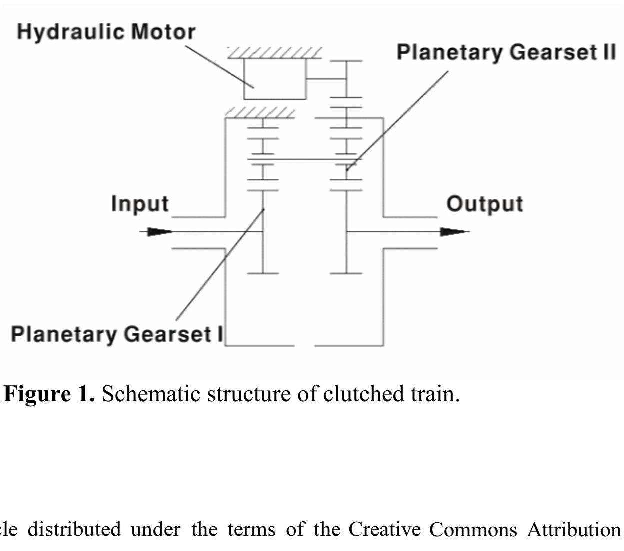

Figure 1. Schematic structure of clutched train.

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Figure 2 illustrates the synchronization process of clutched train with hydrostatic braking system from the opening to sticking phase. The mode-switch valve would hand off the DOF of rotational dynamics from two to one or from one to two via controlling the reversible motor-pump free or locked. In the opening and slipping mode, the mode switch is set to connect the reversible motor-pump discharge port to the pressure relief valve for regulating the operating pressure. In the sticking mode, the mode-switch valve is shifted to connect the discharge port to check valve to cut the discharge flow. As an expected consequence, it is supposed to have same capability as a conventional clutch in interrupting and transferring the power from engine to transmission. The torque of output shaft during slipping phase, which depends on the back pressure of reversible motor-pump, can be controlled via the pressure relief valve. Accumulator can be installed to harvest the fluid energy produced by reversible motor-pump for decelerating to zero instead of friction clutch converting into heat. In other words, it makes the reversible motor-pump act as a pump to recover the pent-up momentum-carrying energy. Note that after synchronization the PGS-II ring gear is held by the back pressure of reversible motor-pump without any energy supply in terms of keeping engagement. Once overload is happened on clutched train, pressure relief valve can ensure clutched train working on safe operation by setting maximum allowable pressure.

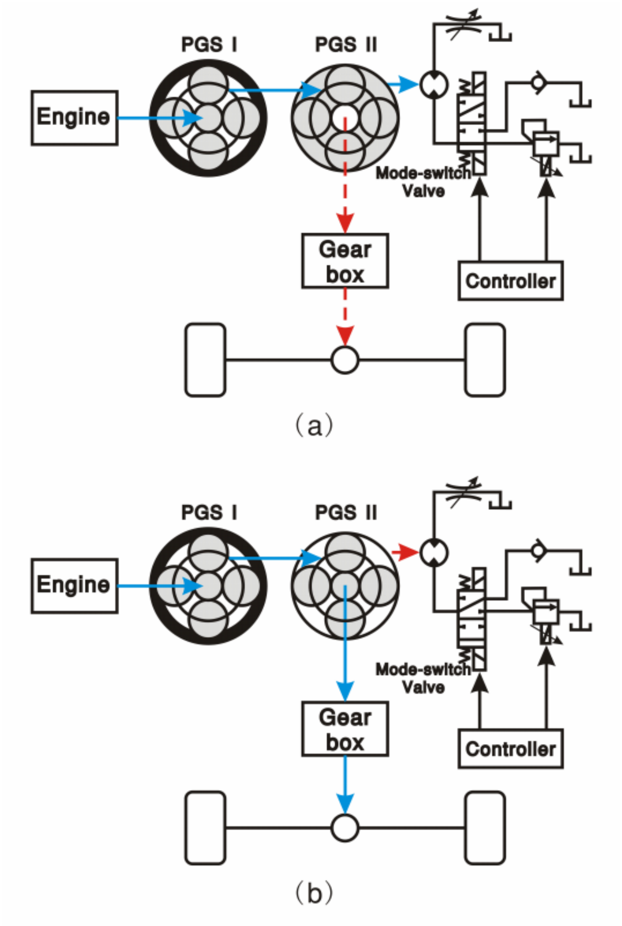

Figure 2. Power flow in driveline equipped with clutched train working in the (a) Opening phase; (b) Sticking phase (PGS-II ring gear locked).

3 Parameter optimization

In view of other merits about friction clutch like compact size and light weight, the parameters correlated to proposed gear train should be taken into account for compact size and light weight as well. Referring to literature[6], Genetic Algorithm (GA) was recognized as an effective and efficient technique to optimize weight design problem of gear train. Several reasonable gear train parameters are selected, and formulated for dimension optimization problem and constrained by the planetary gearset geometry limitations, mating conditions and load capacity, and solve the problem by using GA.

3.1 GA setting

In order to create better new individuals, GA would repeat selection, crossover and mutation until reaching the optimum solution to the problem. The main setting of GA includes:

i) Crossover: 0.85. Determines the probability that an offspring will be created by the crossover of two parents as opposed to a replica of one parent;
ii) Mutation: 0.1. Determines the probability of a spot mutation occurring to each allele in a new population;
iii) Population Size: 400. The population size determines the number of chromosomes that make up a generation.

3.2 Optimum parameter

The optimization result of gear train parameters is listed in Table 1 below. These optimum parameters are used to investigate the functionality and effects on vehicle driving comfort performance of clutched train via simulation.

Table 1. GA optimum parameter.

	Parameter	Value
z_S	Teeth number of sun gear	30
b	Gear face width (mm)	10
m	Gear module (mm)	2
K	Gear ratio between ring gear and sun gear	3

4 Dynamic modeling of clutched train

4.1 Kinematics of clutched train

In this study, the compound planetary gearset is divided into two single stage planetary gearsets which have the same parameters and share one arm. The relationship of angular velocity among sun gear, ring gear and arm in single stage planetary gearset can be determined by the known Willis rule: ‘the ratio of the relative (in regard to the arm) angular velocities is equal to their gear ratio’ [7].

For the sun gears, ring gears and arm with the PGS-I and PGS-II, these following equations are obtained:

$$\omega_{S1} + K \cdot \omega_{R1} - (1+K) \cdot \omega_A = 0 \tag{1a}$$

$$\omega_{S2} + K \cdot \omega_{R2} - (1+K) \cdot \omega_A = 0 \tag{1b}$$

$$\omega_{R1} = 0 \tag{1c}$$

where, ω_{S1} is the angular velocity of PGS-I sun gear; ω_{S2} is the angular velocity of PGS-II sun gear; ω_{R1} is the angular velocity of PGS-I ring gear; ω_{R2} is the angular velocity of PGS-II ring gear; ω_A is the angular velocity of arm; and K is the gear ratio between ring gear and sun gear, namely z_R/z_S. Since the parameters of PGS-I and PGS-II are totally same, they have the same gear ratio K.

By elimination of ω_A from Eq. (1a) and (1b), the following kinematics equation is obtained:

$$\omega_{R2} = \frac{\omega_{S1} - \omega_{S2}}{K} \tag{2}$$

Note that while the angular velocity of PGS-II ring gear ω_{R2} decelerating to zero via hydrostatic braking system, Eq. (2) demonstrates that ω_{S1}=ω_{S2}, namely clutched train synchronization.

4.2 Power flow of clutched train

Power losses in gear train (e.g. gear-meshed friction losses) usually is neglected to reasonably simplify the dynamic model on control orient with little consequence. In accordance with the law of energy preservation the sum of powers transmitted by sun gear, ring gear and arm in each single gear train must equal zero [7]:

$$M_{S1} \cdot \omega_{S1} + M_A \cdot \omega_A = 0 \tag{3a}$$

$$M_A \cdot \omega_A + M_{S2} \cdot \omega_{S2} + M_{R2} \cdot \omega_{R2} = 0 \tag{3b}$$

where M_{S1} is the torque of PGS-I sun gear; M_{S2} is the torque of PGS-II sun gear; M_{R1} is the torque of PGS-I ring gear; M_{R2} is the torque of PGS-II ring gear; M_A is the torque of arm.

From the condition of equilibrium of torques, the equations are noted [7]:

$$M_{S1} + M_{R1} + M_A = 0 \tag{4a}$$

$$M_{S2} + M_{R2} + M_A = 0 \tag{4b}$$

Based on Eq. (1a) to (1c), (3a) to (3b) and (4a) to (4b), the relationships of torques among sun gear, ring gear and arm of the PGS-I and PGS-II are derived as follows:

$$M_{S1} = M_{S2} = \frac{1}{K} M_{R2} \tag{5}$$

5 Control-oriented model of slip-stick phase

Since standing-start maneuver has been described as a typical manifestation of phenomenon on longitudinal oscillations excited by friction clutch engagement [8, 9], control algorithm for clutched train synchronization process would be vital important to vehicle driving comfort. Therefore, control-oriented model of clutched train would be built in this section.

5.1 Slipping phase

In light of Eq. (5), torque of sun gear of planetary gearset M_S herein is introduced and denoted as M_S=M_{S1}=M_{S2}. In the slipping phase, the driving shaft's angular velocity is not equal to the driven shaft's angular velocity, namely $\omega_e \neq \omega_r$. The vehicle driveline is considered as a lumped mass system. The simplified model of clutched train on controlled orient is set up and shown in Figure 3.

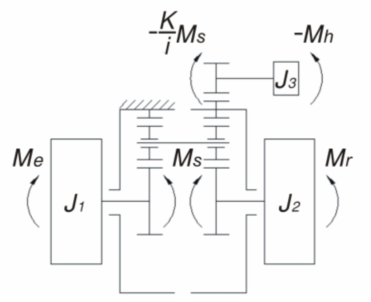

Figure 3. Simplified model of clutched train.

According to the Newton's Second Law, the simplified dynamics of clutched train in the slipping phase is given as follows:

$$J_1 \cdot \dot{\omega}_e = M_e - M_S \tag{6}$$

$$J_2 \cdot \dot{\omega}_r = M_S - M_r \tag{7}$$

$$J_3 \cdot \dot{\omega}_h = \frac{K}{\mu} M_S - \frac{\Delta p \cdot D_v}{2\pi} \eta_m \tag{8}$$

where J_1 is the total moment of inertial of the driving shaft, including the crankshaft, flywheel and PGS-I sun gear; J_2 is the total moment of inertial of the driven shaft, including PGS-II sun gear, driveline, tires and body; J_3 is the moment of inertial of reversible motor-pump shaft; ω_h is the angular velocity of the reversible motor-pump; M_e is engine output torque; M_r is vehicle resistant torque; μ is the gear ratio between the PGS-II ring gear and the gear coupled with reversible motor-pump output shaft, namely z_{R2}/z_h; Δp is the difference of pressure between the hydraulic motor inlet and outlet; D_v is displacement of hydraulic motor; and η_m is mechanical efficiency of hydraulic motor.

Considering the magnitude of J_3 is very small, Eq. (8) can be simplified as:

$$\frac{K}{\mu}M_S \approx \frac{\Delta p \cdot D_v}{2\pi}\eta_m \tag{9}$$

As observed from Eq. (6), (7) and (9), the value of pressure relief valve can be controlled to determine the angular velocity of PGS-II sun gear for less vehicle jerking.

5.2 Sticking phase

Until the angular velocity of PGS-II ring gear declines to zero under hydraulic resistance, according to Eq. (2) the driving shaft's angular velocity is equal to the driven shaft's angular velocity, namely ω_e=ω_r. Then the mode-switch valve will be shifted to sticking operation position. This will allow the PGS-II ring gear to be locked and keep ω_{R2}=0. When ω_e=ω_r=ω_m, the dynamic equation can be rewritten by combining Eq. (6) and (7):

$$(J_1+J_2)\dot{\omega}_m = M_e - M_r \tag{10}$$

6 Model reference control

MRC has been proposed by Chen et al. [8] in friction clutch engagement control to compensate discontinuity of slip-stick transition. However, different feedback control law (regulator) on accommodating clutched train synchronization process is presented. The architecture of MRC including reference model and plant is established in Figure 4.

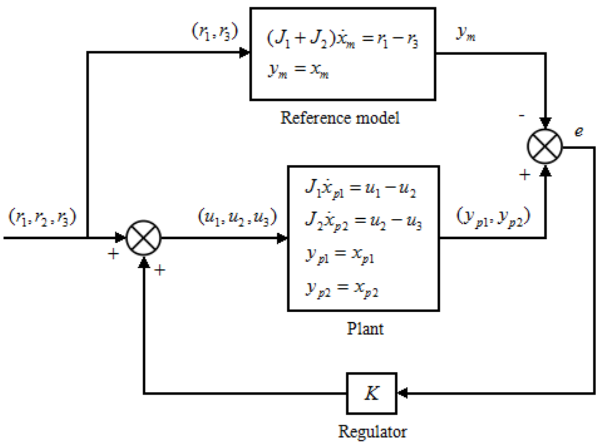

Figure 4. Architecture of MRC.

For the plant, the following variables are introduced and denoted as:

$$x_{p1} = \omega_e; x_{p2} = \omega_r;$$

$$u_1 = M_e; u_2 = M_S; u_3 = M_r$$

Substituting to Eq. (6) and (7), the state equations can be obtained:

$$J_1\dot{x}_{p1} = u_1 - u_2$$

$$J_2\dot{x}_{p2} = u_2 - u_3$$

$$y_{p1} = x_{p1}$$

$$y_{p2} = x_{p2}$$

For the reference model, the following variables are introduced and denoted as:

$$x_m = \omega_m$$

The output of the reference model are denoted as

$$y_m = x_m$$

The state equation is given according to Eq. (7):

$$(J_1+J_2)\dot{x}_m = r_1 - r_3$$

In the proposed MRC scheme, we define that:

i) $r_1(t)$, $r_2(t)$, $r_3(t)$ are reference inputs;
ii) **K** represents the gain for the feedback regulator;
iii) $e_1(t)$ and $e_2(t)$ are two outputs of interest from the closed-loop system, given as:

$$e_1 = y_{p1} - y_m$$

$$e_2 = y_{p2} - y_m$$

The proposed output feedback control algorithm is given as follows:

$$u_1 = r_1 + k_{11}e_1 + k_{12}e_2$$

$$u_2 = r_2 + k_{21}e_1 + k_{22}e_2$$

$$u_3 = r_3 + k_{31}e_1 + k_{32}e_2$$

Then time derivative of the error vector **e** is written as

$$\begin{pmatrix}\dot{e}_1\\ \dot{e}_2\end{pmatrix} = \begin{pmatrix}\dot{y}_{p1}-\dot{y}_m\\ \dot{y}_{p2}-\dot{y}_m\end{pmatrix} = \begin{pmatrix}\dfrac{u_1-u_2}{J_1}-\dfrac{r_1-r_3}{J_1+J_2}\\ \dfrac{u_2-u_3}{J_2}-\dfrac{r_1-r_3}{J_1+J_2}\end{pmatrix}$$

$$= \begin{pmatrix}\dfrac{(r_1+k_{11}e_1+k_{12}e_2)-(r_2+k_{21}e_1+k_{22}e_2)}{J_1}-\dfrac{r_1-r_3}{J_1+J_2}\\ \dfrac{(r_2+k_{21}e_1+k_{22}e_2)-(r_3+k_{31}e_1+k_{32}e_2)}{J_2}-\dfrac{r_1-r_3}{J_1+J_2}\end{pmatrix}$$

$$= \begin{pmatrix}\dfrac{k_{11}-k_{21}}{J_1} & \dfrac{k_{12}-k_{22}}{J_1}\\ \dfrac{k_{21}-k_{31}}{J_2} & \dfrac{k_{22}-k_{32}}{J_2}\end{pmatrix}\begin{pmatrix}e_1\\ e_2\end{pmatrix} + \begin{pmatrix}\dfrac{1}{J_1}-\dfrac{1}{J_1+J_2} & -\dfrac{1}{J_1} & \dfrac{1}{J_1+J_2}\\ -\dfrac{1}{J_1+J_2} & \dfrac{1}{J_2} & \dfrac{1}{J_1+J_2}-\dfrac{1}{J_2}\end{pmatrix}\begin{pmatrix}r_1\\ r_2\\ r_3\end{pmatrix}$$

$$= \mathbf{A}\begin{pmatrix}e_1\\ e_2\end{pmatrix} + \mathbf{B}$$

It is marked that:

$$\mathbf{A} = \begin{pmatrix}(\dfrac{k_{11}-k_{21}}{J_1}) & (\dfrac{k_{12}-k_{22}}{J_1})\\ (\dfrac{k_{21}-k_{31}}{J_2}) & (\dfrac{k_{22}-k_{32}}{J_2})\end{pmatrix}$$

$$\mathbf{B}(r_1,r_2,r_3)=\begin{pmatrix}(\frac{1}{J_1}-\frac{1}{J_1+J_2}) & (-\frac{1}{J_1}) & (\frac{1}{J_1+J_2}) \\ (-\frac{1}{J_1+J_2}) & (\frac{1}{J_2}) & (\frac{1}{J_1+J_2}-\frac{1}{J_2})\end{pmatrix}\begin{pmatrix}r_1\\ r_2\\ r_3\end{pmatrix}$$

Input u_1 physically stands for the engine output torque M_e, which is determined by the work load and can reflect the driver intention. Hence, u_1 is treated as a predefined input and need not be adjusted by MRC. Therefore,

$$k_{11}=0\,\&\,k_{12}=0$$

Similarly, input u_3 physically stands for the resistant torque M_r imposed on the driven shaft, which cannot be changed subjectively. Hence,

$$k_{31}=0\,\&\,k_{32}=0$$

So,

$$\mathbf{A}=\begin{pmatrix}(\frac{-k_{21}}{J_1}) & (\frac{-k_{22}}{J_1})\\ (\frac{k_{21}}{J_2}) & (\frac{k_{22}}{J_2})\end{pmatrix}$$

The critical variable is u_2. In order to achieve asymptotic stability, every eigenvalue of **A** should have a strictly negative real part in accordance with Lyapunov stability theory [10]. Then

$$|\lambda\mathbf{I}-\mathbf{A}|=0$$

$$\begin{vmatrix}(\lambda+\frac{k_{21}}{J_1}) & (\frac{k_{22}}{J_1})\\ (\frac{-k_{21}}{J_2}) & (\lambda-\frac{k_{22}}{J_2})\end{vmatrix}=0$$

$$\lambda^2+(\frac{k_{21}}{J_1}-\frac{k_{22}}{J_2})\lambda-\frac{2k_{21}k_{22}}{J_1J_2}=0$$

According to Vieta's formulas, thus k_{21} and k_{22} should satisfy:

$$\lambda_1+\lambda_2=\frac{k_{22}}{J_2}-\frac{k_{21}}{J_1}<0$$

$$\lambda_1\lambda_2=-\frac{2k_{21}k_{22}}{J_1J_2}>0$$

Regarding to simulation, k_{21} and k_{22} are set to 0.7 and -0.7, respectively.

Note that these three reference inputs r_1, r_2 and r_3 do not affect the system stability but will influence the output response **e**. The main function of the feed-forward regular is to avoid disturbance on the zero-input response, so that the exogenous term $\mathbf{B}(r_1,r_2,r_3)$ can reinforce the error convergence, the feed-forward variables r_1, r_2 and r_3 are therefore subject to the constraints:

$$(\frac{1}{J_1}-\frac{1}{J_1+J_2})r_1+(-\frac{1}{J_1})r_2+(\frac{1}{J_1+J_2})r_3=0$$

and

$$(-\frac{1}{J_1+J_2})r_1+(\frac{1}{J_2})r_2+(\frac{1}{J_1+J_2}-\frac{1}{J_2})r_3=0$$

As a consequence, r_2 can be calculated as:

$$r_2=\frac{J_2r_1+J_1r_3}{J_1+J_2}$$

The overall MRC algorithm can be summarized as:

$$u_2=\frac{J_2r_1+J_1r_3}{J_1+J_2}+k_{21}e_1+k_{22}e_2$$

As mentioned by Chen et al. [8], u_2 should be restricted under certain rate of increase k_f due to physical mechanism limitation, namely

$$u_2^*=k_ft$$

So, the torque of sun gear of planetary gearset M_S is determined by

$$\bar{u}_2=\min(u_2,u_2^*)$$

7 Simulation results

Co-simulation is carried out between ITI SimulationX (as plant) and Simulink/MATLAB (as controller) on the case of vehicle standing-start. The parameters of vehicle model are given in Table 2. The engine characteristics is shown in Figure 5.

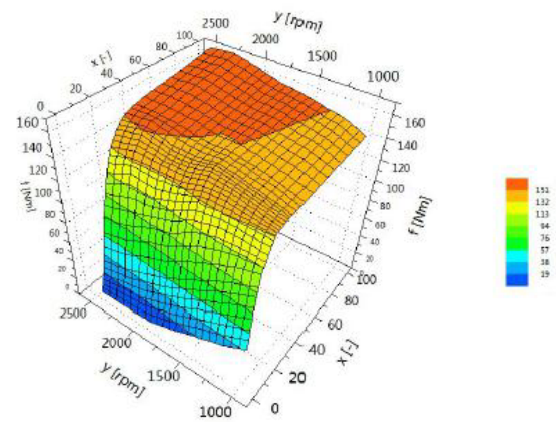

Figure 5. Engine characteristics.

The simulation results of clutched train working with MRC during slip-stick phase are shown in Figure 6, Figure 7 and Figure 8. As illustrated in Figure 6, clutched train can reach synchronization under MRC. Meanwhile, vehicle jerk observed in Figure 7 shows that it can be reduced to below 8.01 m/s^3 and 16.18 m/s^3 with less than 3 Hz and thus satisfy the requirement of driving comfort

[11]. It also indicates that the simplified control-oriented model omitting the moment of inertial of reversible motor-pump shaft has little effect on comfortable synchronization performance. However, as shown in Figure 8, the maximum of measured back pressure from reversible motor-pump are 38.89 MPa. In other words, rated pressure of hydraulic system will become the limitation of maximum transmitted torque in power train system.

Table 2. Parameters of vehicle model.

	Parameters	Value
J_1	Total moment of inertial of the driving shaft (kg·m^2)	1.01
J_2	Total moment of inertial of the driven shaft (kg·m^2)	0.97
J_3	Total moment of inertial of the reversible motor-pump shaft (kg·m^2)	0.01
m	Vehicle mass (kg)	1400
i_0	Gear ratio of reducer	4
i_1	Gear ratio of gearbox	3.32
R	Tire radius (m)	0.35
f	Rolling friction coefficient	0.01

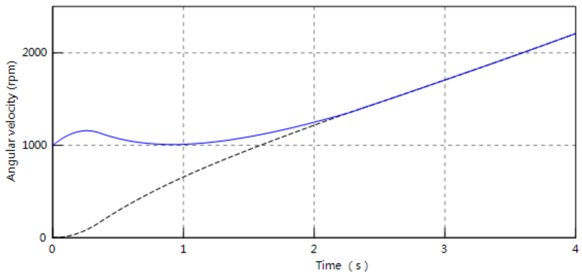

Figure 6. Angular velocity of driving and driven shaft at the slip-stick transition. (solid line: angular velocity of driving shaft, dashed line: angular velocity of driven shaft)

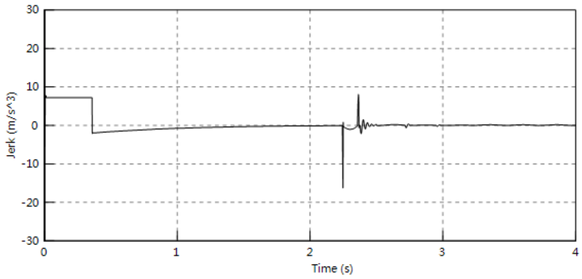

Figure 7. Derivation of vehicle acceleration, namely jerk.

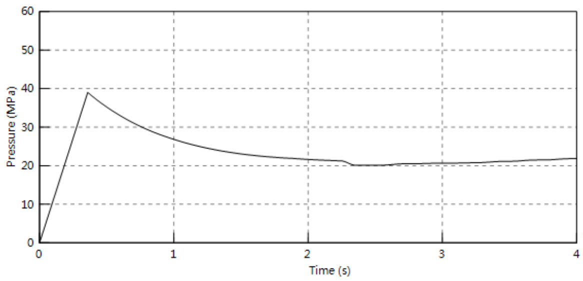

Figure 8. Pressure of hydraulic system.

8 Conclusion

Conceptual design of clutched train with hydrostatic braking system is provided in this paper. Control-oriented model and MRC on clutched train synchronization process are developed. Simulation results show that conceptual proposal has the desired function to connect and disconnect engine with transmission. In the period of clutched train slip-stick, vehicle jerk can be accommodated by MRC for comfortable driving experience of vehicle standing-start. However, the maximum transmitted torque of proposed concept will be primarily limited by the rated pressure of hydraulic system and result in limitation of relevant application in heavy vehicle.

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