Feasibility investigation on novel concept of clutched train system to substitute for vehicular friction clutch

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Abstract
Sliding friction is utilized in the vehicular friction clutch to provide some slippage for smooth shifting and standing-start which is associated with some undesired issues, such as wear and heating of plates, introduction of high non-linearity and significant frictional dissipation. In order to reduce heat generation and frictional dissipation during slipping phase, a novel concept of clutched train is proposed to substitute for friction clutch. The dynamics of clutched train is modeling and integrated into powertrain to estimate the effects of clutched train on longitudinal vehicle oscillations stimulated by the sudden change of accelerator pedal. Further, the engagement control of clutched train can be transplanted from the developed control algorithm of friction clutch due to their similar control-oriented models, and the influence on longitudinal vehicle oscillations aroused by clutched train engagement is investigated. Simulation results show that it can behave like the conventional clutch and satisfy the requirement of driving comfort with certain control algorithm. Finally, the frictional dissipation of clutched train engagement is evaluated and compared with friction clutch engagement.

Key words: Clutched train, Planetary gearset, Friction clutch substitution, Vehicle jerk, Driving comfort, Frictional dissipation

1. Introduction

Friction clutch is widely used in contemporary automotive field for connecting an engine and a transmission in a car because of its high efficiency and compact size. However, since it uses the principle of transmitting torque through friction, some issues particularly in terms of shift comfort are inevitable. Generally, in order to improve the driving comfort, engagement speed will be lowered, which unfortunately increases the frictional dissipation in the friction clutch. It would not only cause mass loss of clutch’s friction materials which augments linearly with increasing sliding distance (Vadiraj, 2010), but massive ensuing frictional heat generated during slipping phase also leads to the reduced service life of the clutch. In addition, high non-linearity (Petrun et al., 2012) and discontinuity (Chen et al., 2011) during slip-stick transition are introduced into clutch actuation and vehicular powertrain dynamics. Thereby, various complicated friction models and control algorithms were developed and served for driving comfort and clutch service life.

Previous research paper by Zhang et al. (2002) described the application of Adaptive Optimal Control to compromise in friction plate wear and driving comfort, while other research papers from Tanaka and Wada (1995) and Liu and He (2009) showed that Fuzzy Control provides skillful operation for friction clutch engagement. These controllers improve the driving comfort during the slipping phase. Meanwhile, other researchers claimed that Model Referenced Control reduces the jerking effect significantly during friction clutch slip-stick transition (Chen et al., 2011; Chen et al., 2012). However, in the aforementioned studies the friction coefficient between the clutch plates is considered to be constant over the whole range of relative sliding velocities, temperature, normal force, direction etc (Petrun et al., 2012), and hence makes it difficult to give an accurate signal to actuator for the estimation of the transmitted clutch torque while previous works by Marklund and Larsson (2007) indicated that the friction coefficient of the clutch plates has a noticeable variation that aroused by the different velocities and temperature. Moreover,
Cappetti et al. pointed out that cushion spring compression behavior as an important contribution to clutch engagement performance is also influenced by the rise temperature of frictional heat generation (2012).

Additionally, because of extra power requirement for actuation and slippage involving significant frictional dissipation as implied by previous studies of automatic clutch, planetary gearset is increasingly introduced in several kinds of clutchless hybrid vehicles equipped with dual input gearboxes (Sasaki 1998; Teshima et al., 2006; Yoon et al., 2013). On the other hand, the concept of clutchless has been recognized but rarely realized in the single input gearbox. For example, although hydraulic torque converter takes the place of friction clutch between the engine and the transmission, hydraulic automatic transmission and continuously variable transmission that using hydraulic torque converter still need friction clutch for disengagement to prevent the engine from stalling when the car comes to a stop.

A novel concept of clutched train with Hydrostatic Braking System (HBS) to replace friction clutch used in a car and mated to single input gearbox (automated manual transmission) is proposed in this paper. It is introduced for achieving smooth transfer of torque with less frictional heat generated during engagement. 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. Meanwhile, the proposed clutched train would provide a substantially consistent control-oriented model with the conventional friction clutch during engaging process. Therefore, most of control methods studied for conventional friction clutch might be transplanted.

This paper is mainly concerned with the impact of clutched train system substituted for friction clutch on vehicle longitudinal dynamics and the frictional dissipation during engagement. The dynamic model of clutched train is established and integrated into powertrain to investigate the substitution effects on vehicle longitudinal oscillations during the sudden accelerator pedal change and the clutched train engagement. The clutched train behavior and effects on driving comfort will be found via simulation methods. Based on data acquisition from simulation, frictional dissipation during the clutched train engagement can be evaluated.

2. System description

Fig. 1 shows the concept of clutched train with HBS to substitute for friction clutch in powertrain. The configuration of symmetrical compound planetary gearset can be considered as combination of single stage Planetary-Gearset-I (PGS-I) and Planetary-Gearset-II (PGS-II), which have the same gear parameters and share one arm. The PGS-I ring gear is fixed. The PGS-II ring gear is internal mesh with the planets and external mesh with a gear. 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.

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Fig. 1 Schematic of clutched train with HBS in powertrain.
Note: 1. PGS-I sun gear; 2. PGS-I planets; 3. PGS-I ring gear; 4. Arm; 5. PGS-II sun gear; 6. PGS-II planets; 7. PGS-II ring gear; 8. Gear coupled with reversible motor-pump output shaft; 9. Reversible motor-pump; 10. Pressure relief valve; 11. Oil tank; 12. Mode-switch valve; 13. Check valve.
The engagement process of clutched train can be described with the same way as friction clutch, including opening, slipping and sticking phase. 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 the check valve to cut the discharge flow. As an expected consequence, clutched train could work in the position of conventional clutch on interrupting and transferring the power from the engine to the 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. Instead of converting into heat during slipping, an accumulator can be installed to harvest the fluid energy produced by the reversible motor-pump for decelerating to zero. In other words, it would make the reversible motor-pump act as a pump to recover the pend-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 additional energy supply in terms of keeping fully engaged.

Specification of clutched train system is shown in Fig. 2 and listed in Table 1. These parameters are designed and selected to mate CamPro Engine S4PH (1.6 L).

Table 1. Parameters of clutched train system.

| Gear Parameters                           | Value   |
|------------------------------------------|---------|
| Teeth number                             | Sun 30  |
|                                          | Planet 30 |
|                                          | Ring 90 |
| Gear face width (mm)                     | 10      |
| Normal module (mm)                       | 2       |
| Pressure angle (deg)                     | 20      |
| Helix angle (deg)                        | 12.68   |
| Inertia (kg·m²/rad)                      | 0.03    |
|                                          | 0.0005  |
|                                          | 0.05    |
| Transverse contact ratio between planet and sun gear | 1.65 |
| Transverse contact ratio between planet and ring gear | 1.91 |

Reversible Motor-pump

| Parameters                           | Value   |
|--------------------------------------|---------|
| Displacement volume (cm³)            | 19      |
| Max continuous operating pressure (MPa) | 35    |
| Max intermittent operating pressure (MPa) | 42 (<5 sec) |
| Max continuous operating speed (rpm)  | 8100    |
| Max intermittent operating speed (rpm) | 8900   |

3. Dynamics of clutched train

3.1 Kinematics

The configuration of proposed clutched train consists of two single stage planetary gearsets with the same parameters and sharing one arm. According to the known Willis rule (Damir 2012), relationship of angular velocity among sun gear, ring gear and arm in single stage planetary gearset can be obtained:

\[ \omega_{S1} + K \cdot \omega_{R1} - (1 + K) \cdot \omega_{A} = 0 \]  
\[ \omega_{S2} + K \cdot \omega_{R2} - (1 + K) \cdot \omega_{A} = 0 \]  
\[ \omega_{R1} = 0 \] (1c)

where \( \omega_{S1} \) is the angular velocity of PGS-I sun gear; \( \omega_{S2} \) is the angular velocity of PGS-II sun gear; \( \omega_{R1} \) is the angular velocity of PGS-I ring gear; \( \omega_{R2} \) is the angular velocity of PGS-II ring gear; \( \omega_{A} \) is the angular velocity of arm; and \( K \) is the ratio of teeth number of ring gear \( z_R \) to teeth number of sun gear \( z_S \), namely \( z_R/z_S \). Since the gear parameters of PGS-I is coincide with PGS-II, same ratio \( K \) is assigned.
By elimination of $\omega_A$ from Eq. (1a) and (1b), the following kinematics equation is obtained:

$$\omega_{R2} = \frac{\omega_{S1} - \omega_{S2}}{K} \quad (2)$$

### 3.2 Power flow

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 (Damir 2012):

$$M_{S1} \cdot \omega_{S1} + M_A \cdot \omega_A + L_1 = 0$$

$$M_A \cdot \omega_A + M_{S2} \cdot \omega_{S2} + M_{R2} \cdot \omega_{R2} + L_2 = 0$$

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; and $L_i$ $(i=1, 2)$ is the power loss in single stage planetary gearset, which is defined as:

$$L_1 = -(1-\eta_1) \cdot M_{S1} \cdot \omega_{S1}$$

$$L_2 = -(1-\eta_2) \cdot M_A \cdot \omega_A$$

where $\eta_i$ $(i=1, 2)$ is efficiency of single stage gear train. Calculation of efficiency $\eta_i$ $(i=1, 2)$ is based on expressions (Eq. 17 (a) and (b)) in literature derived by Huang et al. (2014). Then we can obtain:

$$\eta_1 \cdot M_{S1} \cdot \omega_{S1} + M_A \cdot \omega_A = 0 \quad (3a)$$

$$\eta_2 \cdot M_A \cdot \omega_A + M_{S2} \cdot \omega_{S2} + M_{R2} \cdot \omega_{R2} = 0 \quad (3b)$$

At equilibrium of torques, the equations are noted (Damir 2012):

$$\eta_1 \cdot M_{S1} + M_{R1} + M_A = 0 \quad (4a)$$

$$M_{S2} + M_{R2} + \eta_2 \cdot M_A = 0 \quad (4b)$$

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:

$$M_{S1} : M_{R1} : M_A = 1 : (\eta_1 \cdot K) : [ -\eta_1 \cdot (1 + K) ] \quad (5a)$$

$$M_{S2} : M_{R2} : M_A = 1 : K : -\frac{(1+K)}{\eta_2} \quad (5b)$$

### 3.3 Gear mesh model

Given that mesh stiffness variation is the primary stimuli of gear vibration (Lin & Parker, 2002), gear mesh model is phrased in terms of time-varying mesh stiffness which is aroused by the change in the number of contact teeth pair and contact position of gear teeth. According to analytical studies by Tian et al. (2004) and Liang et al. (2013), time-varying mesh stiffness of planet-sun and planet-ring gears can be functions of the relative mesh mesh phases. The
corresponding formulas are programming in MATLAB and plotted in Fig. 3 and Fig. 4, respectively.

Fig. 3 Mesh stiffness of sun-planet gears when the ring gear is locked.

Fig. 4 Mesh stiffness of ring-planet gears when the ring gear is locked.

3.4 Dynamics

In respect of shuffle mode oscillation study, dynamic model of clutched train system considers the rotational motions of the gear bodies with time-varying gear mesh stiffness (refer to Section 3.3) and no backlash based upon the dynamic modeling methodology of single-planet gearset generalized by Inalpolat and Kahraman (2008). Assumption of no backlash that described in linear time-variant model has been well supported by experimental study on helical gears (Kubur et al., 2004). In order to obtain frequency domain with linear solution, overall time-varying gear mesh stiffness is suggested to be transformed to constant average gear mesh stiffness with periodic transmission error excitation (Inalpolat & Kahraman, 2008). Relevant important constituents on vibration response properties can be captured with a view to investigate the effect of clutched train on longitudinal vehicle dynamics. Fig. 5 illustrates undamped torsional model with assumptions of perfect lubrication condition. Based on dynamic model in literature (Inalpolat & Kahraman, 2008), equations of motion for PGS-I are given as:

\[ J_{S1} \cdot \ddot{\theta}_{S1} + \sum_{i=1}^{n1} k_{S1,P1i} \cdot r_{S1} \cdot \delta_{S1,P1i} = |M_e| - \sum_{i=1}^{n1} k_{S1,P1i} \cdot r_{S1} \cdot e_{S1,P1i}(t) \]

\[ J_{P1} \cdot \ddot{\theta}_{P1} + k_{S1,P1} \cdot r_{P1} \cdot \delta_{S1,P1i} - k_{R1,P1i} \cdot r_{P1} \cdot \delta_{R1,P1i} = -k_{S1,P1} \cdot r_{P1} \cdot e_{S1,P1i}(t) + k_{R1,P1i} \cdot r_{P1} \cdot e_{R1,P1i}(t) \]

\[ J_{R1} \cdot \ddot{\theta}_{R1} + \sum_{i=1}^{n1} k_{R1,P1i} \cdot r_{R1} \cdot \delta_{R1,P1i} + k_{R1,G} \cdot r_{R1} \cdot \delta_{R1,G} = -\sum_{i=1}^{n1} k_{R1,P1i} \cdot r_{R1} \cdot e_{R1,P1i}(t) \]

\[ J_A \cdot \ddot{\theta}_A + \sum_{i=1}^{n1} J_{P1} \cdot \ddot{\theta}_{P1i,A} \cdot \sum_{i=1}^{n1} [k_{S1,P1} \cdot r_{A,P1} \cdot \delta_{S1,P1i} + k_{R1,P1i} \cdot r_{A,P1} \cdot \delta_{R1,P1i}] = -|M_A| + \sum_{i=1}^{n1} [k_{S1,P1} \cdot r_{A,P1} \cdot e_{S1,P1i}(t) + k_{R1,P1i} \cdot r_{A,P1} \cdot e_{R1,P1i}(t)] \]

where engine output torque \( Me \) as the driving torque in the clutched train system is function in literature by Zhang et al. (2002); referring to Inalpolat & Kahraman’s study (2008), transmission error excitation \( e_{Sj,Pj} \) \((j=1,2)\) and \( e_{Rj,Pj} \) \((j=1,2)\) are defined in Fourier series form based on evaluation of overall effective mesh stiffness fluctuations; relative gear mesh displacements are defined as \( \delta_{Sj,Pj}=r_{Sj} (\theta_{Sj}-\theta_{A})+r_{Pj} (\theta_{Pj,e}-\theta_{A}) \), \( \delta_{Rj,Pj}=r_{Rj} (\theta_{Rj}-\theta_{A})+r_{Pj} (\theta_{Pj,e}-\theta_{A}) \) and \( \delta_{Rj,G}=r_{Rj} \theta_{Rj} \); relative displacement \( \theta_{Pj,e} \) of planet \( P_j \) \((j=1,2)\) to arm is defined as \( \theta_{Pj,e}=\theta_{Rj}-\theta_{A} \); total inertia \( J_A \) of the arm with \((n1+n2)\) planets is defined as \( J_A=J_1+n1J_{P1}+n2J_{P2}+n1n2m_{P1} (r_{S1}+r_{P1})+n2m_{P2} (r_{S2}+r_{P2}) \) where \( J_A \) is moment of inertia of the arm; \( m_{P} \) is mass of a planet; and \( n1 \) and \( n2 \) are number of planets of PGS-I and PGS-II, respectively.
Similarly, equations of motion for PGS-II are described as:

\[
J_A \ddot{\theta}_A + \sum_{i=1}^{n_2} J_{p2} \ddot{\theta}_{p2i,A} + \sum_{i=1}^{n_2} [k_{s2,p2} \cdot r_{A,p2} \cdot \delta_{s2,p2i} + k_{r2,p2} \cdot r_{A,p2} \cdot \delta_{r2,p2i}] = [M_A] - \sum_{i=1}^{n_2} [k_{s2,p2} \cdot r_{A,p2} \cdot e_{s2,p2i}(t) + k_{r2,p2} \cdot r_{A,p2} \cdot e_{r2,p2i}(t)]
\]

\[
J_{p2} \ddot{\theta}_{p2i} + k_{s2,p2} \cdot r_{p2} \cdot \delta_{s2,p2i} - k_{r2,p2} \cdot r_{p2} \cdot \delta_{r2,p2i} = -k_{s2,p2} \cdot r_{p2} \cdot e_{s2,p2i}(t) + k_{r2,p2} \cdot r_{p2} \cdot e_{r2,p2i}(t)
\]

\[
J_{s2} \ddot{\theta}_{s2} - \sum_{i=1}^{n_2} k_{s2,p2} \cdot r_{s2} \cdot \delta_{s2,p2i} = [M_s] + \sum_{i=1}^{n_2} k_{s2,p2} \cdot r_{s2} \cdot e_{s2,p2i}(t)
\]

\[
J_{r2} \ddot{\theta}_{r2} + \sum_{i=1}^{n_2} k_{r2,p2} \cdot r_{r2} \cdot \delta_{r2,p2i} = \sum_{i=1}^{n_2} k_{r2,p2} \cdot r_{r2} \cdot e_{r2,p2i}(t) - k_{r2,p2} \cdot r_{r2} \cdot e_{r2,p2i}(t)
\]

\[
J_p \ddot{\theta}_p + k_{r2,p} \cdot r_p \cdot \delta_{r2,p} + k_{M,G} \cdot r_M \cdot \delta_{M,G} = \frac{\Delta p \cdot D_p}{2 \pi \cdot \eta_m} - k_{r2,p} \cdot r_p \cdot e_{r2,p}(t)
\]

where relative gear mesh displacements are defined as \( \delta_{s2,p} = r_{s2} \cdot \theta_{s2} + r_p \cdot \theta_p \) and \( \delta_{M,G} = r_M \cdot \theta_M \); \( \Delta p \) is the pressure drop between the reversible motor-pump inlet and outlet ports; \( D_p \) is the displacement of reversible motor-pump; \( \eta_m \) is the mechanical efficiency of reversible motor-pump; and function of vehicle resistant torque \( M_r \) refers to literature by Zhang et al. (2002).
4. Longitudinal oscillations excited by tip-in and tip-out

Either sudden accelerator pedal change or clutched component engagement can easily stimulate longitudinal oscillations. Effect of clutched train system on longitudinal vehicle oscillations excited by the sudden change of accelerator pedal, so-called tip-in and tip-out, will be discussed in this session. As shown in Fig. 6, ITI SimulationX model of powertrain components mainly comprises rotating inertias connected by shaft flexibilities. Stiffness element mainly includes input shaft, tooth mesh and drive shaft. In order to estimate important shuffle mode natural frequencies of driveline, lowest resonance mode will be treated here in case when in low gears (Kiencke and Nielsen, 2005) and the clutched train is fully engaged. The representative data of driveline used in simulation is given by Hrovat and Tobler (see Table 1 in literature (Hrovat and Tobler, 1991)) in addition to clutched train system.

![Fig. 6 Torsional elastic model of powertrain.](image_url)

![Fig. 7 Engine torque and speed difference.](image_url)

![Fig. 8 Shuffle mode.](image_url)
Driveline oscillations behavior can be described by longitudinal vehicle acceleration or speed difference (Kiencke and Nielsen, 2005). Fig. 7 presents the speed difference while abrupt step on accelerator pedal and resulted in dramatic changes in engine torque. Based on speed difference, shuffle mode can be achieved by Fast Fourier Transformation. As observed in Fig. 8, shuffle mode natural frequency is about circa 2.1 Hz. Since the typical frequency range of driveline component design change is 2-10 Hz, clutched train system as substitution for friction clutch would not bring noticeable influence to driveline on longitudinal vehicle oscillations during synchronization.

5. Longitudinal oscillations excited by clutched train engagement

Standing-start maneuver has been described as a typical manifestation of the phenomenon on longitudinal oscillations excited by the friction clutch engagement (Chen et al., 2011). For this reason, the control-oriented model of clutched train is integrated with the developed control algorithms to investigate longitudinal oscillations excited by the clutched train engagement on the standing-start case particularly.

5.1 Control-oriented model of clutched train system

Derivation of control-oriented model of clutched train system is shown in Appendixes. Comparing the control-oriented dynamics expressions listed in Table 2, the proposed clutched train system is akin to conventional clutch. It implies that most of control methods used in friction clutch might be transplanted into clutched train system.

Table 2. Comparison of control-oriented models of clutched train system and friction clutch.

| Model  | Clutched train system (see Appendixes) | Friction clutch (Zhang et al., 2002; Chen et al., 2011) |
|--------|---------------------------------------|--------------------------------------------------------|
| Slipping | \[
\begin{aligned}
J_1 \cdot \dot{\omega}_e &= M_e - M_S \\
J_2 \cdot \dot{\omega}_r &= M_S - M_r
\end{aligned}
\] | \[
\begin{aligned}
J_1 \cdot \dot{\omega}_e &= M_e - M_C \\
J_2 \cdot \dot{\omega}_r &= M_C - M_r
\end{aligned}
\] |
| Sticking | \[
(J_1 + J_2) \dot{\omega}_m = M_e - M_r
\] | \[
(J_1 + J_2) \dot{\omega}_m = M_e - M_r
\] |
| Actuation | \[M_S = \text{abs}(\frac{\eta_s \cdot D_x \cdot \tau}{2\pi K} \Delta p) \cdot \text{sign}(\omega_e - \omega_s)\] | \[M_C = \text{abs}(\mu \cdot p_c \cdot A \cdot R) \cdot \text{sign}(\omega_e - \omega_c)\] |

5.2 Simulation

Co-simulation is implemented between ITI SimulationX (as plant shown in Fig. 9) and Simulink/MATLAB (as controller). Simulation results of clutched train running on slip-stick phase combined with Model Referenced Control (MRC), Constant Rate Control (CRC) and Fuzzy Control (FC) to diminish the impact on jerking, respectively, are demonstrated and discussed in the following sections.
When the signal input of pressure relief valve is given from controller, the angular velocity of PGS-II ring gear will decrease to zero gradually under the variational hydraulic resistance. The angular velocity of driven shaft is expected to approach to driving shaft with less jerking. Therefore, in this study, MRC, CRC and FC are chosen to supervise the engagement process and subdue the effect of engagement on driving comfort. The simulation results of proposed substitution under MRC, CRC and FC are shown in Fig. 10, Fig. 11 and Fig. 12, respectively.

![Fig. 10 Angular velocity of driving and driven shaft at the slip-stick transition.](image1)

![Fig. 11 Derivation of vehicle acceleration, namely jerk.](image2)

![Fig. 12 Pressure of hydraulic system.](image3)

Referring to the studies of Chen et al. (2011; 2012), feedback regulator was proposed in the architecture of MRC. The selected feedback gain \(k_{ij}\) (\(i=2; j=1, 2\)) are set to 0.7 and -0.7, respectively, in accordance with Lyapunov stability theory. As for CRC, the friction work, which was taken into account for preventing overwear of friction plates (Zhang et al., 2002), is removed from the evaluation of optimal trajectory on the proposed substitution engagement. With regard to FC, the rate of engagement depends on the amplitude and the change in engine working load, and the ratio of...
the angular velocity difference between the driving and the driven shaft to the angular velocity of the driving shaft (Tanaka and Wada, 1995; Liu and He, 2009).

As shown in Fig. 10, MRC, CRC and FC can be used in the clutched train engagement control to connect the engine and the transmission. But CRC spends less time to reach synchronization than two other controllers.

Fluctuations in Fig. 11 illustrate that clutched train has the same problem of discontinuity (Chen et al., 2011) like conventional clutch during slip-stick transition. Zhang et al. recommend that vehicle jerk can be comfortably accepted below 25.5 m/s\(^3\) when frequency is no higher than 3 Hz; or below 10 m/s\(^3\) (2007). Therefore, vehicle jerk from CRC and FC are beyond 25.5 m/s\(^3\) and resulted in uncomfortable driving experience, while MRC can reduce vehicle jerk to below 8.01 m/s\(^3\) and 16.18 m/s\(^3\) with less than 3 Hz and satisfy the requirement of driving comfort. It also indicates that the dynamic model omitting the moment of inertial of reversible motor-pump shaft (refer to Eq. (10)) has little effect on comfortable engagement performance.

As observed in Fig. 12, CRC needs less of rated pressure with the hydraulic system than MRC. While discontinuity of slip-stick transition induces pressure pulsations into the hydraulic system, MRC can contain the pressure pulsations.

6. Frictional dissipation during engagement

In accordance to data acquisition from simulation, frictional dissipation of gear mesh in the proposed clutched train engagement can be assessed (see Eq. (15) derived in study (Huang et al., 2014) and assume that individual gear efficiency between different gear engagement are same as 0.97). As for evaluation of frictional dissipation of slippage in the friction plates of conventional friction clutch, see literature (Chen et al., 2011). The aforementioned frictional dissipation of clutched train and friction clutch during slipping phase with MRC, CRC and FC are listed in Table 3 and depicted in Fig. 13. Comparing with friction clutch, the advantage of clutched train on less frictional heat generation during engagement can be validated.

| Control Algorithm       | Clutched Train System (97%) | Friction Clutch       |
|-------------------------|-----------------------------|-----------------------|
| Model Referenced Control| \(2.508 \times 10^3\) J      | \(1.285 \times 10^4\) J|
| Constant Rate Control   | \(2.099 \times 10^3\) J      | \(1.339 \times 10^4\) J|
| Fuzzy Control           | \(3.241 \times 10^3\) J      | \(2.554 \times 10^4\) J|

Fig. 13 Difference of frictional dissipation between clutched train (97%) and friction clutch engagement.

7. Conclusion

In this paper, a clutched train to substitute for a conventional friction clutch is presented, modeling and investigated via software simulation on the standing-start case. The simulation results indicate that:

1) Clutched train as substitution for friction clutch would not bring noticeable influence into powertrain on longitudinal vehicle oscillations during synchronization.
2) Proposed substitution can provide similar process to friction clutch for engagement and disengagement.
3) Several developed control algorithms on conventional clutch can be transplanted and applied into clutched train
employed in powertrain due to consistent control-oriented model.

4) Compared with Constant Rated Control, vehicle jerk on the standing-start case can be reduced to comfortably accepted level by Model Referenced Control, but higher requirement of rated pressure with the hydraulic system.

5) Model Referenced Control can contain pressure pulsation generation during slip-stick transition.

6) Frictional dissipation of clutched train is much less than friction clutch during engagement.

Considering the frequent use of clutch on account of traffic congestion, it would be significance for developing clutched train to recover the kinetic energy instead of being transformed into useless heat. However, this simulation work only provides preliminary investigation about feasibility of the proposed concept. Future work will encompass the functionality of prototype with further experimental study.

Appendices

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, namely \( \eta_1=\eta_2=1 \). Eq. (5a) and (5b) can be rewritten as:

\[
M_{s1} = M_{s2} = \frac{1}{K} M_{r2}
\]  

(6)

In light of Eq. (6) above, 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 powertrain is considered as a lumped mass system. The simplified model of clutched train on controlled orient is set up and shown in Fig. 14.

![Simplified model of clutched train](image)

Fig. 14 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:

\[
J_1 \cdot \ddot{\omega}_e = M_e - M_s
\]  

(7)

\[
J_2 \cdot \ddot{\omega}_r = M_s - M_r
\]  

(8)

\[
J_3 \cdot \ddot{\omega}_p = \frac{K}{\tau} M_s - \text{abs}(\frac{\Delta p \cdot D}{2\pi} \eta_m) \cdot \text{sign}(\omega_e - \omega_s)
\]  

(9)

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; \( \omega_p \) is the angular velocity of the reversible motor-pump; \( \tau \) is the gear ratio between the PGS-II ring gear and the gear coupled with reversible motor-pump output shaft, namely \( z_{R2}/z_h \). Considering the magnitude of \( J_3 \) is very small, the Eq. (9) can be simplified as:

\[
\frac{K}{\tau} M_s \approx \text{abs}(\frac{\Delta p \cdot D}{2\pi} \eta_m) \cdot \text{sign}(\omega_e - \omega_s)
\]  

(10)
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 $\omega_e = \omega_c$. 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 $\omega_{R2} = 0$. When $\omega_e = \omega_c = \omega_m$, the dynamic equation can be rewritten by combining Eq. (7) and (8) thus:

$$(J_1 + J_2)\ddot{\omega}_m = M_c - M_e,$$  \hspace{1cm} (11)

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