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Frictional Dissipation and Mechanical Efficiency Analysis of Clutched Train Engagement

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ABSTRACT

Background: Clutched train is a rarely mentioned application of epicyclic gear train on alternation of friction clutch. Objective: In this paper we analyze the power flow in clutched train and apply the concept of virtual power to derive expressions for assessing frictional dissipation during engaging process and mechanical efficiency over clutched train synchronization . Results: Analysis result indicates that parameter of gear train is related to frictional dissipation and mechanical efficiency. Conclusion: These expression and analysis result would be instrumental in clutched train study involving frictional dissipation assessment and mechanical efficiency improvement.

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INTRODUCTION

Epicyclic gear train has been introduced and studied in hybrid car's transmission with its power-split function (Sasaki, 1998; Liu and Peng, 2008), and also proposed in several kinds of clutchless hybrid vehicle system equipped with dual input gearboxes (Teshima *et al*., 2006; Yoon *et al*., 2013). As a branch of epicyclic gear train applications, the concept of clutched train by braking one member is expected as alternation of friction clutch in automotive power train. Less heat generated on clutched train engagement is believed to improve service life as well as accurate control, while friction clutch using sliding friction to smooth the engaging process has significant heat generation which would make it difficult to give an accurate signal to actuator for the estimation of the transmitted clutch torque owing to obviously variant friction coefficient.

In order to proof the advantage of clutched train, the frictional dissipation aroused by gear-meshed sliding friction during engagement should be assessed. And parameter of epicyclic gear train linking to less heat may be found. Even though several comprehensive methodologies (Macmillan, 1961; Yu & Beachley, 1985; Pennestrì & Valentini, 2003; Chen & Liang, 2011) have been used in analyzing many different developed epicyclic systems, either structure or power loss on clutched train is rarely mentioned and discussed. Here we apply the concept of virtual power proposed by Chen & Angeles (2006) to derive analytical expression of frictional dissipation carried by gearmeshed sliding friction on clutched train. Given the mechanical efficiency of epicyclic system is usually much lower than a simple gear train (Pennestrì & Freudenstein, 1993), the analysis of mechanical efficiency over clutched train synchronization should be taken into account for improvement.

In this paper a novel pattern of clutched train and corresponded kinematics are presented. The approach involving power flow and virtual power flow analysis is conducted to study power loss on clutched train. We focus on the frictional dissipation during the period of clutched train engaging process and mechanical efficiency while the clutched train synchronization. Then gear train parameter is discussed with frictional dissipation reduction and mechanical efficiency improvement in accordance with derived expressions.

2. System Description:

As shown in Figure 1, the clutched train consists of a compound epicyclic system and a hydraulic system. In compound epicyclic system two carriers of two simple epicyclic gear trains with same parameters are connected to share shaft. The ring gear of Epicyclic Gear Train I is fixed. The sun gear of Epicyclic Gear Train I is coupled with the driving shaft, which is driven by the output shaft of an engine. The sun gear of Epicyclic Gear Train II is coupled to a coaxial output shaft which drives an automatic transmission.

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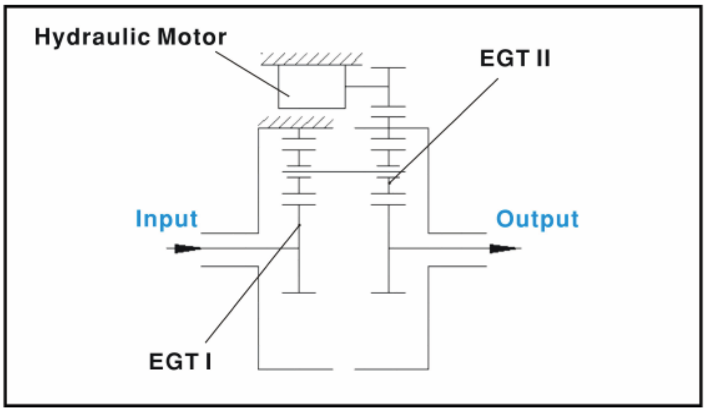

Fig. 1: Schematic of clutched train with hydraulic system.
EGT: Epicyclic Gear Train.

Figure 2 illustrates the power flow in driveline during clutched train engaging process from the opening to sticking phase. The DOF (degree of freedom) of rotational dynamics can be turned from two to one or from one to two via mode-switch valve to brake or release the rotator in reversible motor-pump. In analogous opening and slipping mode, the mode switch is set to connect the reversible motor-pump discharge port to pressure relief valve. The back pressure value of reversible motor-pump controlled by pressure relief valve is supposed to determine the output torque of clutched train. While in analogous sticking mode the mode-switch valve is shifted to check valve to block the discharge port. The increasing back pressure will brake the ring gear of Epicyclic Gear Train II and realize clutched train synchronization. As a result, clutched train would behave as a conventional clutch with functions of interrupting and transferring the power from engine to transmission. Note that during analogous slipping phase the rotational dynamics is two DOFs, while it is one DOF in analogous sticking phase. In other word, the system we analyzing on frictional dissipation during engaging process is two DOFs, and on mechanical efficiency over synchronization is one DOF.

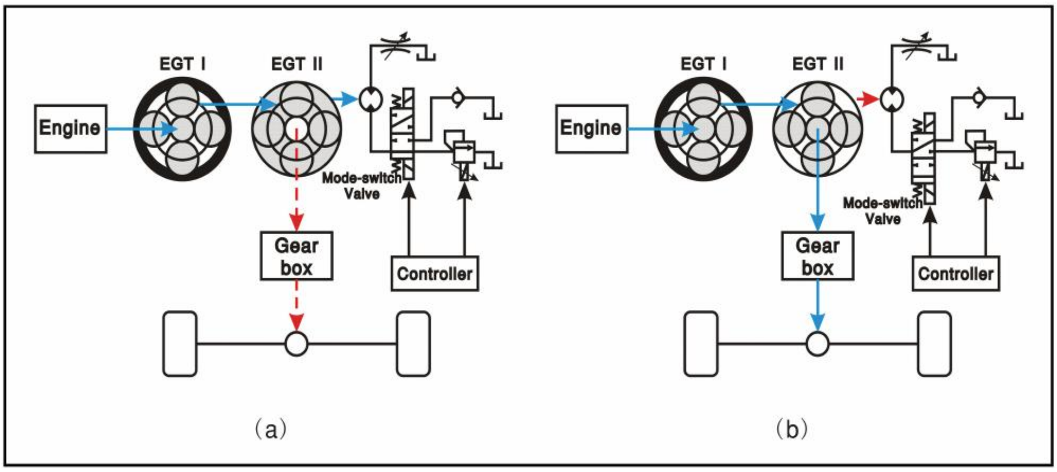

Fig. 2: Power flows in driveline with clutched train system in the (a) Opening phase and (b) Sticking phase (ring gear locked).
EGT: Epicyclic Gear Train.

3. Kinematics Analysis:

In kinematics analysis, the compound epicyclic gear train can be regarded as two simple epicyclic gear trains, which have the same parameters and share one carrier. The relationship of angular velocity among sun gear, ring gear and carrier of epicyclic gear train can be determined by the known Willis rule:
"The ratio of the relative (in regard to the carrier) angular velocities is equal to their gear ratio."
(Jelaska, 2012: 334)

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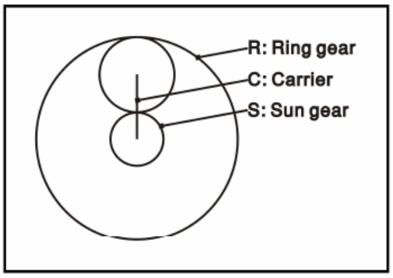

Fig. 3: Epicyclic gear train.

For the sun gears, ring gears and carrier with the Epicyclic Gear Train I and Epicyclic Gear Train II as shown in Figure 3, these following equations are obtained:

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

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

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

where ω_{S1} is the angular velocity of sun gear of Epicyclic Gear Train I; ω_{S2} is the angular velocity of sun gear of Epicyclic Gear Train II; ω_{R1} is the angular velocity of ring gear of Epicyclic Gear Train I; ω_{R2} is the angular velocity of ring gear of Epicyclic Gear Train II; ω_C is the angular velocity of carrier; and K is the gear ratio between ring gear and sun gear, namely z_R/z_S. Because the parameters of Epicyclic Gear Train I and Epicyclic Gear Train II are totally same, they have the same gear ratio K.

By elimination of ω_C from equations (1a) and (1b), the following kinematics equation is obtained:

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

Equation (2) indicates that input shaft and output shaft would synchronize since the angular velocity of ring gear of Epicyclic Gear Train II decreases to zero.

Here two speed ratio are defined upon kinematics relations in equations (1a) to (1c):

$$k_1 = \frac{\omega_C}{\omega_{S1}} = \frac{1}{1+K} \tag{3a}$$

$$k_2 = \frac{\omega_C}{\omega_{S2}} = \frac{1+K \cdot n}{1+K} \tag{3b}$$

where n is ratio between angular velocity of ring gear and angular velocity of sun gear with the Epicyclic Gear Train II, namely ω_{R2}/ω_{S2}.

4. Power Flow Analysis:

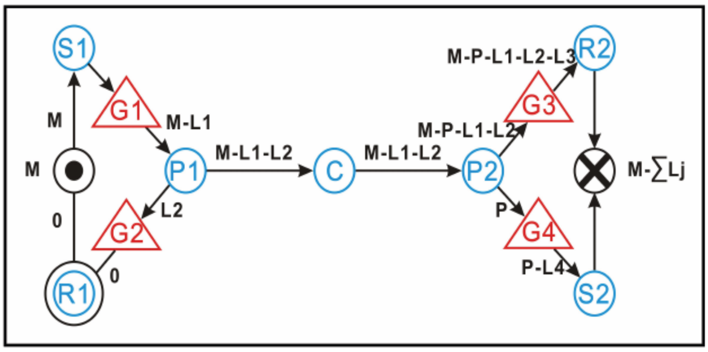

Fig. 4: Power flows of the clutched train with power losses.

Power flow from one link to another indicated by arrow are shown in Figure 4, respectively. As claimed in Chen & Angeles's study (2006) only gear-meshed sliding friction G_j (j=1, 2, 3, 4) is considered as contribution to power losses (neglecting oil churning and friction in shaft support bearings). *M* and *P* are input power and split power, respectively:

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$$M = T_{S1} \cdot \omega_{S1} \tag{4a}$$

$$P - L_4 = T_{S2} \cdot \omega_{S2} \tag{4b}$$

According to the law of energy preservation the sum of powers transmitted by sun gear, ring gear and carrier must equal zero:

$$T_{S1} \cdot \omega_{S1} + T_C \cdot \omega_C + \sum_{j=1}^{2} L_j = 0 \tag{5a}$$

$$T_C \cdot \omega_C + T_{S2} \cdot \omega_{S2} + T_{R2} \cdot \omega_{R2} + \sum_{j=3}^{4} L_j = 0 \tag{5b}$$

where T_{S1} is the torque of sun gear of Epicyclic Gear Train I; T_{S2} is the torque of sun gear of Epicyclic Gear Train II; T_{R1} is the torque of ring gear of Epicyclic Gear Train I; T_{R2} is the torque of ring gear of Epicyclic Gear Train II; T_C is the torque of carrier; L_j is power loss, which through Epicyclic Gear Train I and II are, respectively, given by

$$\sum_{j=1}^{2} L_j = -(1-\eta_I) \cdot T_{S1} \cdot \omega_{S1} \tag{6a}$$

$$\sum_{j=3}^{4} L_j = -(1-\eta_{II}) \cdot T_C \cdot \omega_C \tag{6b}$$

where η_i (i=I, II) is efficiency of single stage gear train.

$$\eta_I \cdot T_{S1} \cdot \omega_{S1} + T_C \cdot \omega_C = 0 \tag{7a}$$

$$\eta_{II} \cdot T_C \cdot \omega_C + T_{S2} \cdot \omega_{S2} + T_{R2} \cdot \omega_{R2} = 0 \tag{7b}$$

From the condition of equilibrium of torques, the equations are noted (Jelaska, 2012):

$$T_{S1} + T_{R1} + T_C = 0 \tag{8a}$$

$$T_{S2} + T_{R2} + T_C = 0 \tag{8b}$$

From equations (1a) to (1c), (3a) to (3b), (7a) to (7b) and (8a) to (8b), the relationships of torques among sun gear, ring gear and carrier of the Epicyclic Gear Train I and Epicyclic Gear Train II are derived as follows:

$$T_{S1} : T_{R1} : T_C = 1 : (\eta_I \cdot K) : [-\eta_I \cdot (1+K)] \tag{9a}$$

$$T_{S2} : T_{R2} : T_C = 1 : \frac{\eta_{II}(1+K) - 1 - n \cdot K}{1-n} : -\frac{\eta_{II}(1+K) - n(1+K)}{1-n} \tag{9b}$$

So,

$$\frac{T_{S2}}{T_{S1}} = \frac{\eta_I (1-n)}{\eta_{II} - n} \tag{10}$$

5. Frictional Dissipation During Engagement:

As stated in Chen & Angeles's study (2006) virtual power is defined as the power measured by an observer standing on a rotating frame with a constant angular velocity. The power quantities indicated in Figure 5 are virtual power. Note that the carrier is grounded. In the same system the virtual power and corresponding split power in Figure 5 is *V* and *V'*, respectively.

As claimed by Chen & Angeles (2006) an observer is supposed to stand on an arbitrary frame with a constant angular velocity ωA. The virtual ratios $\alpha 1$ and $\alpha 2$ passing through Links S1 and S2, respectively, are given as:

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$$\alpha_1 = \frac{\omega_{S1} - \omega_C}{\omega_{S1}} = 1 - k_1 = \frac{V}{M} \tag{11a}$$

$$\alpha_2 = \frac{\omega_{S2} - \omega_C}{\omega_{S2}} = 1 - k_2 = \frac{-V' - L_4}{P - L_4} \tag{11b}$$

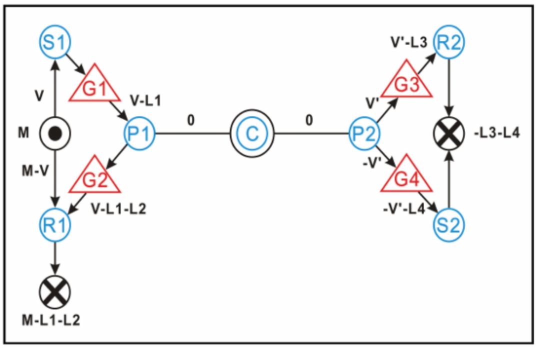

Fig. 5: Virtual power flow of the clutched train with power losses.

According to equations (11a) and (11b), we have

$$V = (1 - k_1)M \tag{12a}$$

$$V' = -(1 - k_2)(P - L_4) - L_4 \tag{12b}$$

Loss factor, as a proportional ratio of power losses to input power, is assumed as a constant (Chen & Angeles, 2006). It can be measured on carrier and be evaluated by an observer standing on the corresponding carrier. Thus the power losses due to gear-meshed sliding friction at G_j (j=1, 2, 3, 4) are given by

$$L_1 = \lambda_1 V \tag{13a}$$

$$L_2 = \lambda_2 (V - L_1) \tag{13b}$$

$$L_3 = \lambda_3 V' \tag{13c}$$

$$L_4 = \lambda_4 (V' - L_3) \tag{13d}$$

where λ_j (j=1, 2, 3, 4) are loss factors of G^j (j=1, 2, 3, 4), respectively.

Chen & Angeles point out that the power losses are invariant with respect to the observing frames (2006). So the expressions of power losses can be obtained in accordance with equations (3a) to (3b), (12a) to (12b), and (13a) to (13d):

$$\sum_{j=1}^{2} L_j = \frac{(1 - \eta_1 \cdot \eta_2)K}{1 + K} \cdot M \tag{14a}$$

$$\sum_{j=3}^{4} L_j = \frac{(1 - \eta_3 \cdot \eta_4)(n - 1)K}{1 + K + \eta_3 (1 - \eta_4)(1 + n \cdot K)} \cdot P \tag{14b}$$

$$\sum_{j=1}^{4} L_j = \frac{(1 - \eta_1 \cdot \eta_2)K}{1 + K} \cdot M + \frac{(1 - \eta_3 \cdot \eta_4)(n - 1)K}{1 + K + \eta_3 (1 - \eta_4)(1 + n \cdot K)} \cdot P \tag{14c}$$

where η_j (j=1, 2, 3, 4) is the local efficiency at the single gear mesh in Epicyclic Gear Train I and Epicyclic Gear Train II.

So the expression of total frictional dissipation during the period of clutched train engaging process can be described as:

$$E = \int_0^{t_0} (\sum_{j=1}^{4} L_j) dt = \int_0^{t_0} (\frac{(1 - \eta_1 \cdot \eta_2)K}{1 + K} \cdot M + \frac{(1 - \eta_3 \cdot \eta_4)(n - 1)K}{1 + K + \eta_3 (1 - \eta_4)(1 + n \cdot K)} \cdot P) dt \tag{15}$$

where t_0 is engagement time. Note that in practice the variant *M*, *P*, *n* can be calculated based on real-time data acquisition.

As observed from the above expression, the appropriate parameter K choosing for epicyclic system under different engine characteristic, power train and control algorithm would have positive impact on less heat generation during analogous slipping phase.

6. Mechanical Efficiency Over Synchronization:

Mechanical efficiency of clutched train during analogous sticking phase is discussed in this section. According to equations (2) and (10), the total mechanical efficiency during clutched train synchronization can be derived as:

$$\eta = \frac{M_{S2} \cdot \omega_{S2}}{M_{S1} \cdot \omega_{S1}} = \frac{\eta_I (1-n)}{\eta_{II}} (1 + n \cdot K) \tag{16}$$

It is marked that

$$\eta_I = \frac{M - \sum_{j=1}^{2} L_j}{M} = \frac{1 + \eta_1 \cdot \eta_2 \cdot K}{1 + K} \tag{17a}$$

$$\eta_{II} = \frac{M - \sum_{j=1}^{4} L_j}{M - \sum_{j=1}^{2} L_j} = \frac{[1 + K + \eta_3 (1 - \eta_4)(1 + K)](1 + n \cdot K)}{[1 + K + \eta_3 (1 - \eta_4)(1 + K)](1 + n \cdot K) - (1 - \eta_3 \cdot \eta_4) K} \tag{17b}$$

While clutched train is synchronization (ring gear of Epicyclic Gear Train II is braking), the ratio n of angular velocity of ring gear ω_{R2} to angular velocity of sun gear ω_{S2} becomes zero. If we assume the individual gear efficiency η_j (j=1, 2, 3, 4) can be obtained by measuring, then the relationship between total mechanical efficiency η and gear ratio K would be held for mechanical efficiency improvement. For example when the individual gear efficiency η_j (j=1, 2, 3, 4) between different gear engagement are same as 0.95, the total mechanical efficiency of clutched train would decline with increasing gear ratio K as shown in Figure 6.

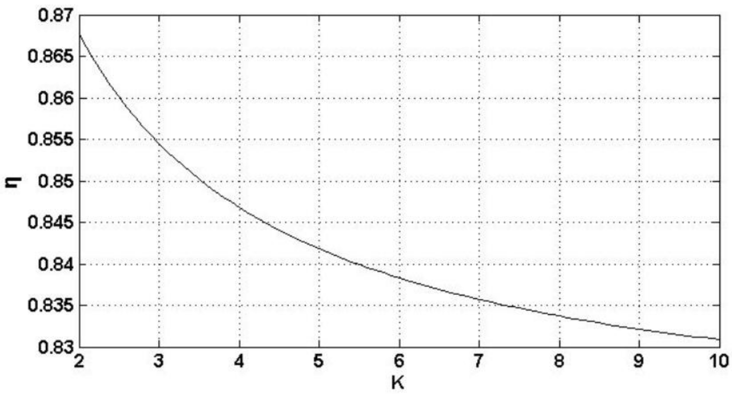

Fig. 6: The total mechanical efficiency η versus gear ratio K over clutched train synchronization.

7. Conclusion:

We apply the concept of virtual power to find the expression of frictional dissipation on clutched train during engagement. Also, the expression for computing the mechanical efficiency, while clutched train is synchronization, is given. The relationship between the mechanical efficiency and gear train parameter is discussed. The analysis shows that mechanical efficiency over clutched train synchronization would be improved with small gear ratio K of ring gear to sun gear.

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