

Kinematic Study of the GM Front-Wheel Drive Two-Mode Transmission and the Toyota Hybrid System THS-II Transmission

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ABSTRACT

General Motors has recently developed a front-wheel drive version of its two planetary two-mode transmission (2-MT) for a hybrid-electric vehicle powertrain [1]. This newer transmission includes two planetary gears with two transfer clutches and two braking clutches. With activation of designated pairs of these four clutches, four fixed-gear ratios between the transmission's input shaft and output shaft are obtained. In addition, activation of specific individual clutches gives two modes of operation whereby the IC engine speed is decoupled from the vehicle velocity thus providing an electrical continuously variable transmission (ECVT). This present paper extends the power-split analysis in [2] by deriving a safe-operating region (SOR) in the plane of IC engine speed vs. vehicle velocity for the four fixed-gear and two ECVT modes. This SOR is bounded by the speed limitations of the 2-MT components. Similar results are presented for the Toyota Hybrid System II (THS-II) transmission. Also developed is the vehicle velocity limit for both the 2-MT and the THS-II transmissions under battery electric vehicle (BEV) operation.

INTRODUCTION

The front-wheel drive version of GM's 2-MT includes two electric machines; two planetaries, one a compound planetary, and the other a simple planetary; two transfer clutches, and two braking clutches. References [1], and [3], [4], [5], [6] give detailed descriptions of this latest version of the 2-MT. As discussed in [2], the activation of specific pairs of these four clutches provides four fixed ratios between the input and output shafts of the 2-MT. This fixed-gear-ratio operation is similar to a four-speed automatic transmission excluding the torque converter. In addition, activation of

specific single clutches provides two operating modes whereby the IC engine speed on the transmission input shaft is decoupled from the vehicle velocity. This decoupling allows for the IC engine to idle without a torque converter. These two modes achieve electrical continuously variable transmission (ECVT) operation by splitting the IC engine output power into two paths to the transmission output shaft. One path maintains the IC engine output power in its original mechanical power form. The second path involves a double energy conversion whereby the IC engine's output mechanical power is converted to electrical power and this electrical power is reconverted back to mechanical power. Thus ECVT operation is achieved at the expense of this mechanical-to-electrical-to-mechanical double energy conversion of a portion of the IC engine output power.

Reference [2] develops a dimensionless variable, termed a *separation factor* that determines the fraction of the IC engine's output power that takes each of these two paths through the transmission. This separation factor is formulated for each of the two ECVT modes. For these two modes the separation factors involve the ratio of the vehicle velocity to the engine speed, similar to the *speed ratio* defined in [9]. Since ECVT operation allows for a decoupling of IC engine speed and vehicle velocity, the value of each of these two separation factors can be adjusted by controlling the engine speed for a given vehicle velocity. Thus in both modes the split of IC engine power between the two paths can be controlled. In general, there is a tradeoff that involves the fact that transmitting 100% of the IC engine power over the mechanical path, yielding the highest transmission efficiency by avoiding the double energy conversion, results in higher less efficient IC engine speeds. So the 2-MT efficiency must be balanced against the IC engine efficiency in the two ECVT modes.

A second paper presents an analysis of the Toyota Hybrid System II (THS-II) transmission used in the second edition of the Prius and in other hybrid-electric vehicles [7]. Similar to the analysis of the 2-MT, a dimensionless separation factor is formulated for the THS-II. However, the THS-II has only a single power-split ECVT mode. As is shown in [2], switching to the second mode in the 2-MT avoids inefficient transmission operation at higher vehicle velocities. Thus in addition to providing four fixed-gear ratios, the 2-MT is shown to avoid these higher vehicle velocity inefficiencies that are a fundamental property of the single-mode THS-II transmission.

In summary, these previous two papers [2] and [7] present an analysis of the energy flow in the THS-II and the 2-MT. This present paper extends these analyses to include the kinematics of these power-split transmissions whereby the speed of all the planetary gears and electric machines is considered. As an example of the need for these speed limits, consider the case of BEV operation with the IC engine off thus constraining its input shaft to be at zero rpm. The kinematic analysis below shows that there exists a rather low upper bound on the vehicle velocity. This bound is caused by a speed limitation on the planetary pinion gears in the input planetary for the 2-MT, and by the maximum allowable speed for the electric machine G/A (known as the generator) for the THS-II.

STRUCTURE AND ANALYSIS OF THE TWO PLANETARIES IN THE 2-MT

STRUCTURE

Figure 1, from [2] and stick diagram of Fig. 3 in [1], shows the structure of the front-wheel drive 2-MT with the two transfer clutches $CL2$ and $CL4$, and the two braking clutches $CL1$ and $CL3$. The first planetary-1 is a compound planetary with two sets of pinion gears mounted on a single carrier. The second simple planetary-2 has a single set of pinion gears between the sun and ring gears. The electric machines designated as G/A and M/B have their terminal powers designated positive when the machines operate as a generator (symbol G/A) and motor (symbol M/B), respectively. The ring, sun, and carrier gears are designated as R , S , and C , respectively, with the added number indicating the planetary.

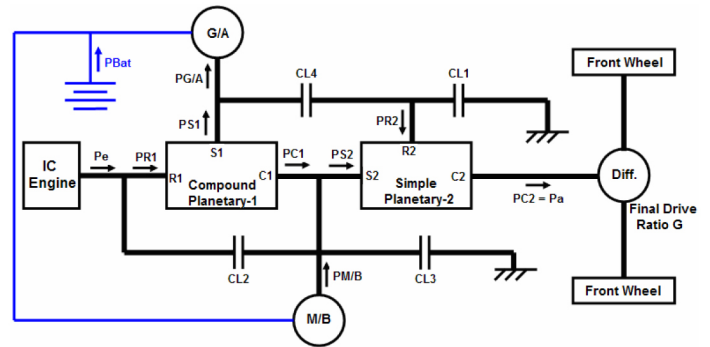


Figure 1. Schematic layout of the GM Two-Mode Transmission.

IMPOSED SPEED CONSTRAINT EQUATIONS FOR PLANETARIES-1 AND -2

Figure 2 shows the structure of the simple planetary-2 used in the front-wheel drive version of the 2-MT. This simple planetary has a single set of three pinion gears between the sun and ring gears. The carrier joins the axes of these three pinion gears and rotates at a speed of ω_C , as shown in Fig. 2.

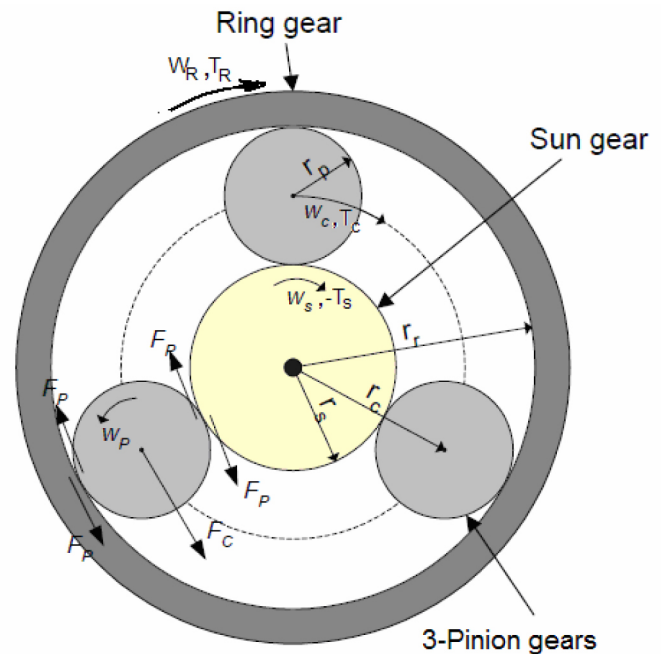


Figure 2. Structure of the simple planetary-2 that is shown in Fig. 1. (Ref. [8])

Figure 3 shows the structure of the compound planetary-1 in Fig. 1. Note that there are two sets of coupled pinion gears between the sun and ring gears. Both sets of pinion gears are mounted on a single carrier.

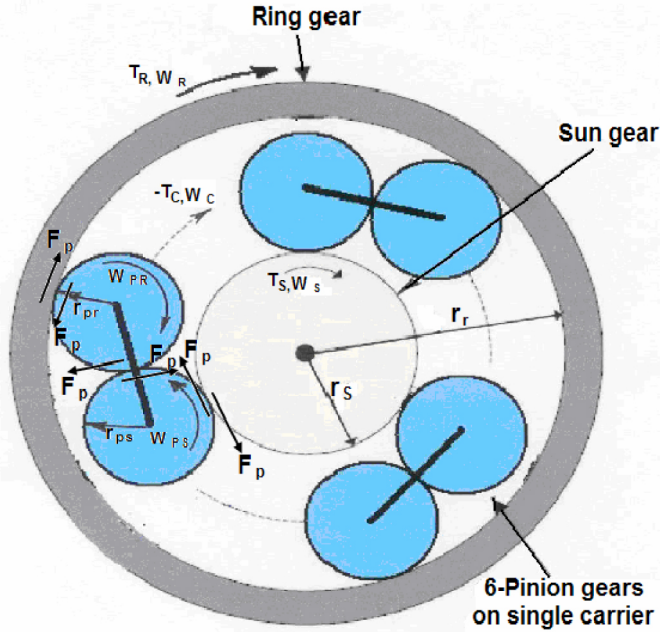


Figure 3. Structure of the compound planetary-1 shown in Fig. 1.

Appendix A presents an analysis of these two planetaries deriving the following speed constraints for the sun, carrier, and ring gears that have external shafts. The resulting equations involve the characteristic ratios defined by

$$\text{Compound Planetary-1: } \rho_1 = \frac{N_{S1}}{N_{R1}} \quad (1a)$$

$$\text{Simple Planetary -2: } \rho_2 = \frac{N_{S2}}{N_{R2}} \quad (1b)$$

where N_S and N_R are the number of gear teeth on the sun and ring gears for each planetary, respectively. The subscripts 1 and 2 refer to planetaries-1 and -2 in Fig. 1. The ratio of the number of teeth on the sun and ring gears is equal to the ratio of the gear radii used in Appendix A.

The speed constraint equation for the three external shafts (sun, carrier, and ring) for each of the two planetaries, as derived in Appendix A, Eqs. (A-4) and (A-13), are given by

Compound Planetary-1:

$$(1 - \rho_1) \cdot \omega_{C1} + \rho_1 \cdot \omega_{S1} - \omega_{R1} = 0 \quad (2a)$$

Simple Planetary-2:

$$(1 + \rho_2) \cdot \omega_{C2} - \rho_2 \cdot \omega_{S2} - \omega_{R2} = 0 \quad (2b)$$

where ω is the speed of the planetary shafts as indicated by the respective subscripts. Note that if two external shaft speeds for either planetary are specified, then the third shaft speed is determined.

In addition to these constraint equations for the three external planetary shafts, the speed equations for the pinion gears are also derived in Appendix A. These speed equations for the pinion gears are important as their angular speeds can exceed design limits even though the three external shafts have admissible angular speed values. Appendix A derives the pinion gear angular speed equations for the compound planetary-1 inner and outer set of pinion gears assuming differing number of teeth for each set of three pinion gears. These pinion gear speeds are as follows

$$\omega_{P1S} = \frac{N_{S1}}{N_{P1S}} \cdot (\omega_{S1} - \omega_{C1}) \quad (3a)$$

$$\omega_{P1R} = \frac{N_{R1}}{N_{P1R}} \cdot (\omega_{R1} - \omega_{C1}) \quad (3b)$$

$$\omega_{P2} = \frac{N_{S2}}{N_{P2}} \cdot (\omega_{S2} - \omega_{C2}) \quad (3c)$$

The inner set of pinion gears for planetary-1, contacting the sun gear, have S as a subscript, with R as a subscript for the outer set contacting the ring gear.

PLANETARY GEAR ANGULAR SPEEDS FOR THE 2-MT OPERATING IN THE FOUR FIXED-GEAR RATIOS

Table 1, (repeated from [2]), gives the clutch-pair activations for the clutches in Fig. 1 to obtain the four fixed-gear ratios for the 2-MT. The constraints imposed by these activations are also given in Table 1 as well as the ratio $\omega_{R1} / \omega_{C2} = (\text{Input speed} / \text{Output speed})$. The numerical values in the last column use the parameter values for the front-wheel drive 2-MT given in the Parameter Values section. If the vehicle velocity is specified, then the angular speed of all planetary gears can be determined in each of the four fixed-gear ratios.

Now using these constraints in Table 1, and the shaft speed constraints in Eqs. (2a) and (2b), Table 2 is developed giving

Table 1. Summary of clutch activation and transmission ratios for the four fixed-ratio configurations.

Gear	Clutch	Constraint	Clutch	Constraint	ω_{R1}/ω_{C2}	ω_{R1}/ω_{C2}
1	CL1	$\omega_{R2} = 0$	CL2	$\omega_{R1} = \omega_{C1} = \omega_{S2}$	$(1 + \rho_2)/\rho_2$	3.243/1
2	CL1	$\omega_{R2} = 0$	CL4	$\omega_{S1} = \omega_{R2} = 0$	$(1 - \rho_1) \cdot (1 + \rho_2)/\rho_2$	1.871/1
3	CL4	$\omega_{S1} = \omega_{R2}$	CL2	$\omega_{R1} = \omega_{C1} = \omega_{S2}$	1/1	1/1
4	CL4	$\omega_{S1} = \omega_{R2}$	CL3	$\omega_{C1} = \omega_{S2} = 0$	$\rho_1 \cdot (1 + \rho_2)$	0.612/1

Table 2. Angular speed of gears in the two planetaries in the 2-MT divided by the vehicle velocity for all four fixed-gear ratios.

Gear	ω_{R1}/V	ω_{S1}/V	$\omega_{C1}/V = \omega_{S2}/V$	ω_{P1S}/V	ω_{R2}/V	ω_{C2}/V	ω_{P2}/V
1	$\frac{(1 + \rho_2)}{\rho_2} \cdot \frac{G}{R}$	$\frac{(1 + \rho_2)}{\rho_2} \cdot \frac{G}{R}$	$\frac{(1 + \rho_2)}{\rho_2} \cdot \frac{G}{R}$	0	0	$\frac{G}{R}$	$\frac{N_{S2}}{N_{P2}} \cdot \frac{1}{\rho_2} \cdot \frac{G}{R}$
2	$\frac{(1 - \rho_1)(1 + \rho_2)}{\rho_2} \cdot \frac{G}{R}$	0	$\frac{(1 + \rho_2)}{\rho_2} \cdot \frac{G}{R}$	$-\frac{N_{S1}}{N_{P1S}} \cdot \frac{(1 + \rho_2)}{\rho_2} \cdot \frac{G}{R}$	0	$\frac{G}{R}$	$\frac{N_{S2}}{N_{P2}} \cdot \frac{1}{\rho_2} \cdot \frac{G}{R}$
3	$\frac{G}{R}$	$\frac{G}{R}$	$\frac{G}{R}$	0	$\frac{G}{R}$	$\frac{G}{R}$	0
4	$\rho_1 \cdot (1 + \rho_2) \cdot \frac{G}{R}$	$(1 + \rho_2) \cdot \frac{G}{R}$	0	$\frac{N_{S1}}{N_{P1S}} \cdot (1 + \rho_2) \cdot \frac{G}{R}$	$(1 + \rho_2) \cdot \frac{G}{R}$	$\frac{G}{R}$	$-\frac{N_{S2}}{N_{P2}} \cdot \frac{G}{R}$

the angular speed of all gears in the two planetaries divided by the vehicle velocity. The angular speed of the outer set of pinion gears in planetary-1, $\omega_{P1R} = (N_{P1S}/N_{P1R}) \cdot \omega_{P1S}$, is not included, as it can be readily calculated from ω_{P1S} . With $N_{P1R} > N_{P1S}$, we have $\omega_{P1R} < \omega_{P1S}$, so ω_{P1S} is the limiting pinion gear speed. In Table 2, V is the vehicle velocity, G is the final drive ratio, and R is the driven wheel rolling radius.

PLANETARY GEAR ANGULAR SPEEDS FOR THE 2-MT OPERATING IN MODES-1 AND -2

The previous section specifies the specific pair of the four clutches that need to be activated to give the four fixed-gear ratios between the input shaft on planetary-1's ring gear and the output shaft on the carrier gear of planetary-2. As shown in Table 2, the angular speeds of all of the 2-MT gears are specified as a function of the vehicle velocity. The 2-MT in addition to these four fixed ratios can operate in two additional modes designated as Modes-1 and -2. These two additional modes are achieved by the operation of a single clutch; CL1 for Mode-1 and CL4 for Mode-2, respectively. The key feature in both Modes-1 and -2, is that the IC engine speed is decoupled from the vehicle velocity, thus giving ECVT operation for the transmission. This then means that the speed of certain transmission gears can now be a function

of both the angular speed of the input shaft (i.e. the IC engine speed), and the speed of the output shaft (i.e. directly proportional to the vehicle velocity).

A detailed analysis of Modes-1 and -2, yields expressions for the angular speeds for all gears in the 2-MT two planetaries as a function of the IC engine speed ω_e , and the vehicle velocity, V . The results of this analysis are presented in Table 3 for Mode-1 and in Table 4 for Mode-2.

RANGE LIMITS ON SHAFT ANGULAR SPEEDS AND VEHICLE VELOCITY

Operation of the 2-MT must consider a set of angular speed limits and a vehicle velocity limit, summarized as follows:

$$\text{Engine speed: } \omega_e = \omega_{R1} \in \{0, [\omega_{e\text{-idle}}, \omega_{eM}]\} \quad (4a)$$

$$G/A \text{ speed: } \omega_{G/A} = \omega_{S1} \in [-\omega_{S1M}, \omega_{S1M}] \quad (4b)$$

$$M/B \text{ speed: } \omega_{M/B} = \omega_{C1} = \omega_{S2} \in [-\omega_{C1M}, \omega_{C1M}] \quad (4c)$$

Table 3. 2-MT gear speeds in Mode-1 as a function of the vehicle velocity V and IC engine speed ω_e

Planetary -	Ring ω_R	Sun ω_S	Carrier ω_C	Pinions ($\omega_{P1S}, \omega_{P2}$)
1	ω_e	$\frac{1}{\rho_1} \cdot \omega_e - \frac{(1-\rho_1) \cdot (1+\rho_2)}{\rho_1 \cdot \rho_2} \cdot \frac{G}{R} \cdot V$	$\frac{(1+\rho_2)}{\rho_2} \cdot \frac{G}{R} \cdot V$	$\frac{N_{S1}}{N_{P1S}} \cdot \frac{1}{\rho_1} \cdot [\omega_e - \frac{(1+\rho_2)}{\rho_2} \cdot \frac{G}{R} \cdot V]$
2	0	$\frac{(1+\rho_2)}{\rho_2} \cdot \frac{G}{R} \cdot V$	$\frac{G}{R} \cdot V$	$\frac{N_{R2}}{N_{P2}} \cdot \frac{G}{R} \cdot V$

Table 4. 2-MT gear speeds in Mode-2 as a function of the vehicle velocity V and IC engine speed ω_e

Planetary-	Ring ω_R	Sun ω_S	Carrier ω_C	Pinions ($\omega_{P1S}, \omega_{P2}$)
1	ω_e	$\frac{(1-\rho_1) \cdot (1+\rho_2) \cdot \frac{G}{R} \cdot V - \rho_2 \cdot \omega_e}{(1-\rho_1 - \rho_1 \cdot \rho_2)}$	$\frac{\omega_e - \rho_1 \cdot (1+\rho_2) \cdot \frac{G}{R} \cdot V}{(1-\rho_1 - \rho_1 \cdot \rho_2)}$	$\frac{N_{S1}}{N_{P1S}} \cdot [\frac{(1+\rho_2)}{(1-\rho_1 - \rho_1 \cdot \rho_2)} \cdot (\frac{G}{R} \cdot V - \omega_e)]$
2	ω_{S1}	$\frac{\omega_e - \rho_1 \cdot (1+\rho_2) \cdot \frac{G}{R} \cdot V}{(1-\rho_1 - \rho_1 \cdot \rho_2)}$	$\frac{G}{R} \cdot V$	$\frac{N_{R2}}{N_{P2}} \cdot [\frac{\rho_2}{(1-\rho_1 - \rho_1 \cdot \rho_2)} \cdot (\omega_e - \frac{G}{R} \cdot V)]$

$$\text{Pinion speed: } \omega_{P1S}, \omega_{P1R}, \omega_{P2} \in [-\omega_{PM}, \omega_{PM}] \quad (4d)$$

$$\text{Vehicle velocity: } V \in [0, V_M] \quad (4e)$$

Limit (4a) defines the maximum allowable IC engine speed excluding the range $(0, \omega_{e-idle})^1$. Limits (4b) and (4c) define the maximum allowable speeds for the two electric machines, designated as G/A and M/B in Fig. 1. Note that these speed limits can be positive or negative with respect to the reference directions in Figs. 2 and 3. Limits (4d) define the admissible speed range for the pinion gears in the two planetaries. Finally (4e) defines a vehicle velocity limit, here taken to be in the forward direction. Reverse vehicle velocities involve only electric machine M/B and are not considered in this present paper.

In general, the IC engine speed and the vehicle velocity must be constrained such that all five ranges in (4) are satisfied. This satisfaction must be true in all four fixed-gear ratios and in the two ECVT modes. With operation in each of the four fixed gears, the IC engine speed is in a fixed ratio to the vehicle velocity. In addition, as shown in Table 2, all planetary shaft speeds are also specified in each fixed-gear ratio for a given vehicle velocity. Thus, for the four fixed-gear ratios, there exists a maximum vehicle velocity that will simultaneously satisfy all five range constraints defined in (4).

For the two ECVT modes the various planetary shaft speeds can be a function of both the IC engine speed and the vehicle velocity, as is shown in Tables 3 and 4. Thus there exists a region in the plane of IC engine speed ω_e vs. vehicle velocity V where all five range limits in (4) are simultaneously satisfied. The next Section illustrates the methodology needed to determine the limits on IC engine speed and vehicle velocity for all six operating modes for the 2-MT.

OPERATIONAL LIMITS FOR THE FOUR FIXED-GEAR RATIOS

Table 2 gives expressions for the ratio of the angular speed of all planetary gears to the vehicle velocity for each of the four fixed-gear operating modes. Now by setting the angular speeds to their limiting values, as given by the range limits in (4a), (4b), (4c), (4d), (4e), the corresponding vehicle velocity can be determined. Selecting the smallest vehicle velocity from these calculations gives the *active* angular speed or vehicle velocity constraint (i.e. active refers to the particular limit from (4a), (4b), (4c), (4d), (4e) that includes all other limits). Using the numerical values in the Parameter Values section, the following Table 5 is constructed.

The results in Table 5 are summarized as follows:

FIXED-GEAR 1

The active constraint is the IC engine red-line angular speed of 6000 rpm. The vehicle velocity at this IC engine speed is 50.35 mph. All other 2-MT gears are within their operational

¹Note that both the IC engine speeds $\omega_e = 0$ and $\omega_e = \omega_{e-idle}$ are included in the admissible IC engine speed range in (4a).