A Novel Twin Epicyclic Gear Train for Optimal Power Sharing Strategy in Two-Motors Driven Hybrid Vehicles

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***Abstract*—The paper presents a novel approach in power sharing between an internal combustion engine and two electrical motors (M1 and M2) for optimal drive of a hybrid vehicle. It is realized by employing the three Degrees of Freedom (DOF) twin epicyclic gear train with specially selected gear ratios. The first stage of the train provides either a high torque from the IC engine or moderate torque from the motor M1. The second stage provides moderate torque form the motor M2. The motors are capable of providing sufficient speed and torque to drive the vehicle along the horizontal or close to horizontal road surfaces. The closed-loop controller has been designed to provide a proper switching sequence between IC engine and the motors. When vehicle's torque demand increases beyond of the motors capability (driving up the hill) the controller engages the engine into the system. The designed continuous variable transmission system is able to minimize the power usage of the IC engine and thus to minimize the exhaust gas emission and save cost of petrol usage. Constant charging of the vehicle batteries by the IC engine helps to maintain charge level of the batteries at the nominal level most of the time.**

***Index Terms*—hybrid vehicle, twin epicyclic gear train, optimal power sharing**

I. Introduction

Hybrid electric vehicle technology has created awareness amongst automobile users, especially with regards towards the importance of green technology and a more environmentally-friendly vehicle. Hybrid electric vehicles are being commercialized worldwide by automotive companies such as Chrysler, Ford, General Motors (GM), Honda, Nissan, and Toyota with the growing demands. Over the years, different types of hybrids have been tested to investigate the best configuration that can achieve greater fuel economy. This includes the series hybrid, parallel hybrid, series-parallel hybrid, and the complex hybrid electric drive trains.

There were many attempts by the researches to optimize the power management and control strategies for hybrid electric vehicles. A series configured hybrid vehicle require control between the usage of the IC engine for charging and supplying electrical power for the motor to run as the internal combustion will not be driving the wheels directly [1], [2]. A parallel configured hybrid vehicle requires control of power loading ratio between the IC engine and the electrical motor as this configuration allow both of them via power train to drive the wheels [2], [3]. The series-parallel hybrid vehicle integrates both the series and parallel systems to take advantage of both configurations [2]. The complex hybrid system is derived from the series-parallel hybrid configuration in which the major difference is the bidirectional power flow in and from the electric motor/generator. The bidirectional power flow allows for a more versatile operating mode to suit different types of driving and load conditions.

Yong-Gi Kim [4] states that the power efficiency is accompanied by the converting process between the kinetic energy and the electrical energy due to the large energy flow through the electrical path when the motor generator serves as a transmission and thus prevents the increase in the capacity of the motor generator and poor fuel economy. S.-C. Tzeng *et al*. [5] showed that a parallel-type hybrid electric vehicle utilizing an energy-distribution mechanism and a dual energy-integration mechanism can achieve minimum fuel consumption and minimum emissions for the IC engine. M. Koichiro *et al*. [6] reviewed the Toyota Hybrid System (THS) and showed the improvements in power performance of THS which contains a power-split planetary gear system which combines the benefits of series and parallel vehicles by avoiding the disadvantages or a pure series or pure parallel hybrid with proper power management and control strategies. Kano *et al*. [7], in their patented works proposed a hybrid driving unit using two electric motors, a transmission, by using the power splitting planetary gear. J. Edwards [8] proposed the integral combination of an electric motor means with a transmission means in which the directions of rotation and the rotational speeds of the two inputs is controlled to provide the mechanical output at any desired rotational speed with peak power output, depending on the control parameters, thus providing an ideal infinite speed device.

The development of power splitting transmissions is a crucial breakthrough to facilitate power sharing of hybrid electric vehicle. Ren Q. *et al*. [9] have reviewed different types of power split transmission and its power

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management and control strategies for automobile manufacturers such as Toyota, GM Allison, BMW, and Bosch. Toyota Hybrid System (THS) employs a single mode power split transmission which have five different modes of operation which are electric drive, normal drive, power boost, battery charge, and negative split. GM Allison, Advance Hybrid System (AHS) dual mode hybrid transmission system using two epicyclic gear units has two modes of operation namely the low range and high range whereby the switching of the two modes is accomplished by means of different clutches settings. The crucial aspect of this design is the ability to switch between these two modes smoothly. Dual mode planetary gear hybrid powertrain [10], [11] can produce high transmission efficiency over almost the entire speed ratio range according to the detailed calculation using the three-parameter efficiency function. The input split unit had the advantage of simple control in pure electric vehicle mode and the largest battery charging power in regenerative braking mode whereas a compound split had the advantage of using smaller motors in low to medium speed range [12].

Controller design for hybrid vehicles [13] describes different approaches in control algorithm and techniques such as Rule-based (RB) control, Rule-based plus Fuzzy Logic, Rule-based plus Neural Network (NN), and Dynamic Programming. The power management control system in a hybrid vehicle system is a very important issue as it involves mechanical and electrical power flow which is also correlated to vehicles drivability. The dual mode power split transmission for hybrid electric vehicle [14] uses a rule based control strategy for the energy distribution between the battery and the fuel tank. The control system may shut off the IC engine if the engine is running at inefficient regions or run the IC engine at a more efficient region higher power setting and use the excess power for recharging storage devices. The battery charging control assures that the batter state of charge (SOC) remains within the preset upper and lower bounds as it causes efficient battery operation and prevents the battery from depletion or damage.

II. Torque and Speed Sharing by Means of Twin Epicyclic Gear Train

The proposed hybrid vehicle transmission is shown in Fig. 1. It is essentially a system that has multiple power inputs and outputs. The inputs are connected to the IC engine and two DC motor shafts (M1 and M2), and the output is used to drive the wheels. An additional motor/generator MG is used to start the IC engine and charge the battery while the engine is running.

The twin epicyclic gear train consists of two power transmission stages. The first stage of the transmission is used to provide a smooth and continuous power outflow from the IC engine and motor M1. It has two inputs: one is connected to the IC engine shaft via clutch CL and the second one is connected to the shaft of motor M1. The output of first stage of transmission supplies the IC engine or motor M1 power to the input of second stage of transmission. The IC engine in this design is actuated by the controller only when the load on the vehicle is quite heavy (i.e. driving up the hill) and the motor M1 is actuated when the vehicle load is not heavy (starts driving a vehicle on a horizontal road).

The second stage of drive train comprises of two inputs, one is connected to the output of first stage of transmission and the second input is connected to the shaft of M2 motor. The output of the train is connected to wheels of the vehicle. The motor M2 in this stage is actuated together with motor M1 to provide maximum vehicle speed on freeways.

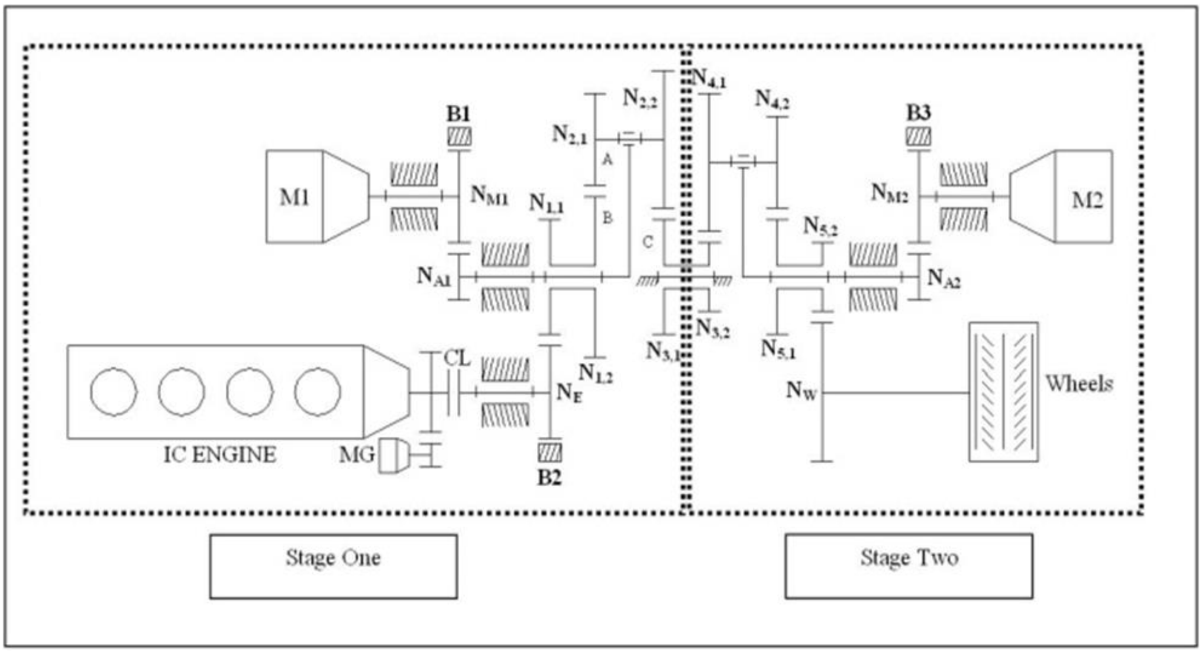

Figure1. Schematic diagram of hybrid vehicle transmission system

The drive train in Fig. 1 can alternatively operate in two different modes, namely Motors and Engine. It depends on the actuation status of three hydraulic brakes B1, B2, B3 and clutch CL. Both modes together will provide a complex hybrid drive system for a vehicle that utilizes fuel power along with electrical energy of the battery.

In Motors mode B2 is activated, whereas B1, B3 are not activated. In this mode, the IC engine is not engaged via clutch CL and its input into the transmission is fully blocked. The drive train provides only two inputs connected to M1, M2 and the output connected to wheels. The velocity ratios in the transmissions are such selected that M1 and M2 will equally provide the required torque and speed for a vehicle wheels. That, in general, will meet the requirement of Continuous Variable Transmission (CVT) for a vehicle with smooth transition from zero to a maximum driving speed on a close to horizontal road surface. The balanced use of two driving motors M1 and M2 should provide an optimal torque and speed sharing while driving the vehicle at various speed conditions. In fact, this arrangement of simultaneous use of two motors can provide same energy consumption, as well as, sufficient vehicle speed and acceleration as in traditional electrical vehicles in the market but without employing the manual transmission. The Motors mode also implies the use of a single motor M1 or M2 by activating B2 or B1, respectively. It may then provide only half of the maximum vehicle speed and can be used as an emergency measure in case of either motor failure.

In Engine mode, B1 and B3 are activated to block the input to the motors M1 and M2. The B2 is not activated. As a result, the IC engine alone will directly drive the wheels with high torque but slow speed due to the selected velocity ratios. This mode can be used as an additional to the electric mode of driving in case of emergency drive with no electrical power available (example, due to the battery problems). If necessary, an additional automatic gearbox can be introduced to vary driving speed of the vehicle in this (IC engine only) mode.

The purpose of this paper is to introduce the design concept of the hybrid power transmission that is based on the twin epicyclic gear train with three power inputs.

III. Kinematic Analysis of the Epicyclic Gear Train

Fig. 1 shows that the epicyclic gear of the first stage of the transmission has two power inputs from M1 and IC engine and one output via the sun gear 3.

The velocity ratios of simple gear trains in this stage of transmission are as follows:

$$\left(\frac{\omega_{arm_1}}{\omega_{M_1}}\right) = -\frac{N_{M_1}}{N_{a_1}} = -VR_{M_1,a_1}\ ; \quad \left(VR_{M_1,a_1} = \frac{N_{M_1}}{N_{a_1}}\right) \qquad (1)$$

$$\left(\frac{\omega_1}{\omega_E}\right) = -\frac{N_E}{N_1} = -VR_{E,1}\ ; \quad \left(VR_{E,1} = \frac{N_E}{N_1}\right) \qquad (2)$$

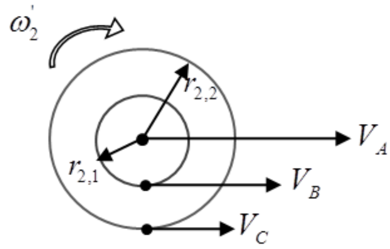

Figure 2. Velocities distribution on the compound gear 2

Fig. 2 shows the distribution of velocity vectors at the contact points of the satellite gear 2. VArm is the velocity of gear center, VB is the velocity at the point of contact B with the sun gear 1, and VC is the velocity at the point of contact C with the sun gear 3 (Fig. 1). r2,1 and r2,2 are the pitch radii of two gears of the compound satellite gear 2. It is obvious from the figure that relative to its own axis the rotational speed of the compound gear ω2' can be defined as:

$$\omega_{2'} = \frac{V_A - V_B}{r_{2,1}} = \frac{V_A - V_C}{r_{2,2}} \qquad (3)$$

Taking into the consideration that $V_B = \omega_1 \cdot r_{1,2}$;

$$V_C = \omega_3 \cdot r_{3,1}; \qquad r_{1,2} + r_{2,1} = r_{3,1} + r_{2,2};$$

$$V_{arm} = \omega_{arm}\left(r_{1,2} + r_{2,1}\right) = \omega_{arm}\left(r_{3,1} + r_{2,2}\right)$$

Formula (3) can be modified into

$$\omega_3 = -\left(\omega_E \cdot \left(VR_{E,1} \cdot VR_{1,3}\right) + \omega_M \cdot VR_{M,a} \cdot \left(1 - VR_{1,3}\right)\right) \qquad (4)$$

In (4) we assume $VR_{1,3} = \dfrac{N_{1,2} \cdot N_{2,2}}{N_{2,1} \cdot N_{3,1}}$.

From Fig. 1 the velocity ratios of simple gear trains in the second stage of the power are as follows:

$$\frac{\omega_{arm_2}}{\omega_{M_2}} = -\frac{N_{m_2}}{N_{A_2}} = -VR_{M_2,A_2}\ ; \qquad VR_{M_2,A_2} = \frac{N_{M_2}}{N_{A_2}}$$

$$\frac{\omega_W}{\omega_5} = -\frac{N_{5,2}}{N_W} = -VR_{W,5}\ ; \qquad VR_{W,5} = \frac{N_{5,2}}{N_W}$$

The velocity equation for this stage of twin epicyclic gear train is derived similar to one of the first stage of transmission (4), i.e.

$$\omega_W \cdot VR_{W,5} = -\omega_3 \cdot VR_{3,5} + \omega_{M_2} \cdot VR_{M_2,A_2} \cdot \left(1 - VR_{3,5}\right) \qquad (5)$$

In (5) we assume $VR_{3,5} = \dfrac{N_{3,2}}{N_{4,1}} \cdot \dfrac{N_{4,2}}{N_{5,1}}$.

For the simplicity of derivation we may accept that $VR_{W,5} = 1$ and thus $N_{5,2} = N_W$ in (5) because it affects equally speed ratios for ω_3 and ω_{M2}.

Substituting (4) into (5) yields the final expression for ω_W:

$$\omega_W = \omega_E \cdot VR_{E,1} \cdot VR_{1,3} \cdot VR_{3,5} + \omega_{M_1} \cdot VR_{M_1 A_1} \cdot (1 - VR_{1,3}) \cdot VR_{3,5} + \\ + \omega_{M_2} \cdot VR_{M_2,A_2} \cdot \left(1 - VR_{3,5}\right) \qquad (6)$$

IV. Force Analysis of the Epicyclic Gear Train and Vehicle Mechanics

The torque equation for the first stage epicyclic transmission can be derived from the static force and torque balance equations for the compound gear 2.

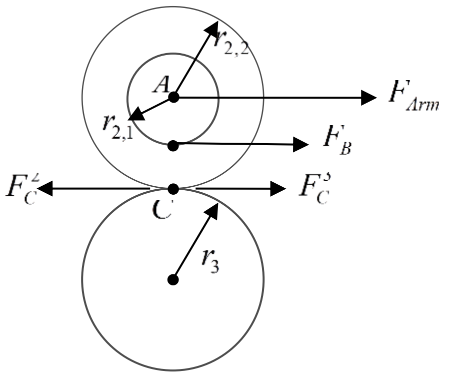

Figure 3. Forces distribution of compound gear 2

Fig. 3 shows the distribution of contact forces at the gear 2. FArm is the force applied by the arm on gear center, FB is the force applied by gear 1 on gear 2, FC2and FC3 are the forces applied by gear 3 on gear 2 and gear 2 on gear 3, respectively. From the free body diagram (Fig. 3) of the compound gear 2 we can obtain the following force balance equation:

$-F_c{}^2 = F_{arm} + F_B$. Since $-F_c{}^2 = F_c^3 = F_c$, then

$$F_c = F_{arm} + F_B \tag{7}$$

The torque balance equation for the compound gear 2 (net torque with respect to center A) can be derived as follows

$$F_c \cdot r_{2,2} = F_B \cdot r_{2,1} \tag{8}$$

Since the system is a two-degrees-of-freedom one, then both equations (7) and (8) should satisfy simultaneously. Excluding F_B from both equations and taking into the consideration that

$F_c = \dfrac{T_3}{r_{3,1}}$ and $F_{arm} = \dfrac{T_{arm}}{r_{1,2} + r_{2,1}} = \dfrac{T_{arm}}{r_{2,2} + r_{3,1}}$, the following formula for torque T_3 can be derived

$$T_3 = T_{arm} \cdot \left(\frac{1}{1 - \dfrac{N_{1,2} \cdot N_{2,2}}{N_{2,1} \cdot N_{3,1}}} \right) \tag{9}$$

Formula (8) can be rewritten in terms of number of teeth as follows

$$T_3 = \frac{T_1}{\dfrac{N_{1,2} \cdot N_{2,2}}{N_{2,1} \cdot N_{3,1}}} \tag{10}$$

The total torque on the shaft of compound gear 3 is the sum of both equations (9) and (10).

$$T_3 = -\left[\frac{T_E}{VR_{E,1} \cdot VR_{1,3}} + \frac{T_{M_1}}{VR_{M_1,a_1} \cdot \left(1 - VR_{1,3}\right)} \right] \tag{11}$$

In (11) we consider that $VR_{1,3} = \dfrac{N_{1,2} - N_{2,2}}{N_{2,1} \cdot N_{3,1}}$;

$$T_1 = -T_E \cdot \frac{N_{1,2}}{N_E} \text{ and } T_{arm} = -T_{M_1} \cdot \frac{N_a}{N_{M_1}}$$

The torque equation for second stage of the twin epicyclic transmission (Fig. 1) can be derived similarly to the first stage of transmission and finally can be expressed as:

$$T_W = -\frac{T_3}{VR_{3,5}} + \frac{T_{M_2}}{VR_{M_2,A_2} \cdot \left(1 - VR_{3,5}\right)} \tag{12}$$

Substituting (11) into (12) yields the final expression for T_W

$$T_W = \left[\begin{array}{l} \dfrac{T_E}{VR_{E,1} \cdot VR_{1,3} \cdot VR_{3,5}} + \dfrac{T_{M_1}}{VR_{M_1,A_1} \cdot \left(1 - VR_{1,3}\right) \cdot VR_{3,5}} + \\ + \dfrac{T_{M_2}}{VR_{M_2 A_2} \cdot (1 - VR_{3,5})} \end{array} \right] \tag{13}$$

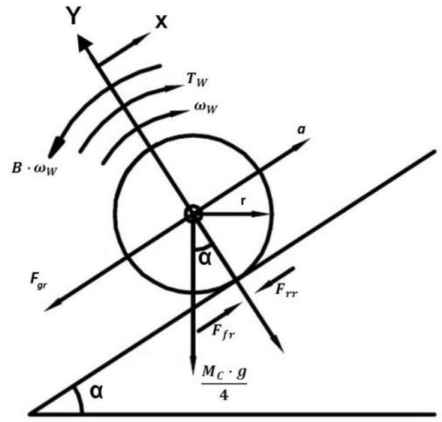

Figure 4. Vehicle single driving wheel on a ramp road

Fig. 4 shows the single driving wheel of a vehicle on a ramp road with all the forces that resist the driving. The resistance can be generated by aerodynamic drag, rolling resistance (coefficient fr), grading resistance due to the ramp road (slope angle α), vehicle inertia forces (vehicle mass MC), wheel shafts viscous friction (coefficient B). In this paper we assume that due to the optimal aerodynamic shape of the vehicle the related drag can be neglected. On the other hand, the driving is fully supported by sufficient traction between the tires and road surfaces (peaking coefficient μP). Fig. 4 also shows the driving torque applied on wheels TD, angular velocity of wheels ω_W, shaft frictional torques $B \cdot \omega_W$, vehicle acceleration a and the wheels radius r. In Fig. 4 the grading resistance force Fgr and rolling resistance force Frr for all four wheels can be combined into the road resistance force Frd and expressed as

$$F_{rd} = M_C(f_r \cos\alpha + \sin\alpha)$$

The general equation of dynamic motion of the car with mass Mc along X-axis (Fig. 4) can be derived based on the Newton's second law of motion and net forces applied on driving wheels.

$$\sum F = \frac{2T_M}{r \cdot VR_M} - M_c \cdot g(\sin\alpha + \cos\alpha \cdot f_r) - \\ - \frac{B \cdot \omega_W}{r} \quad (14)$$

In (14) we consider that $\sum F = M_c \cdot a$; $T_W = {2T_M}/{VR_M}$, and accordance with (13) :

$$VR_M = VR_{M_1A_1} \cdot (1 - VR_{1,3}) \cdot VR_{3,5} = VR_{M_2,A_2} \cdot (1 - VR_{3,5}) \quad (15)$$

In order for the motors to at least accelerate the vehicle (a >0) the following inequality should hold true:

$$\frac{2T_M}{r \cdot VR_M} > M_c \cdot g(\sin\alpha + \cos\alpha \cdot f_r) \quad (16)$$

For simplicity it is assumed zero velocity at the commencement of motion and the negligible value of viscous friction forces in (14). Then the maximum acceleration a_{max} of the vehicle driven by the selected motors and transmission ratios on the horizontal road surface (α =0) can be calculated from (14) as follows:

$$a_{max} = \frac{1}{M_C}\left(\frac{2T_M}{r \cdot VR_M} - M_C \cdot g \cdot f_r\right) \quad (17)$$

The slope of the road with angle α will add the load on the wheels which should be supported by each motor individually to maintain the acceleration along the road. It is because both motor inputs to the epicyclic gear train are independent mechanically and could be back driven individually by the rolling down the slope vehicle due to the growing grading resistance. Then, assuming a single motor T_M driving the vehicle and the equality sign in expression (16) the maximum slope angle α_M for the single motor driven vehicle just before it decelerate can be approximated as follows:

$$\alpha_M < \sin^{-1}\left(\frac{T_M}{r \cdot VR_M \cdot M_C \cdot g}\right) \quad (18)$$

The rolling resistance as a factor of the cosine component of the expression has been ignored as it has small effect on the final value.

V. Selection of Velocity Rations for Optimal Powe Sharing between IC Engine and Electriv Motors

The criteria for selection of velocity ratios for the transmission are based on the idea that the power generated by the IC engine at the first input of the first stage of power train should be more than enough to compensate a possible power required to drive vehicle up a hill with maximum slope angle αmax. Subsequently the torque generated by the motor M1 at the second input of the transmission or motor M2 that is coupled with the second input of the second stage of power train should be sufficiently enough to cope with the vehicle moderate torque demand while driving along horizontal or close to horizontal road surface with maximum angle from (18).

The selection of velocity ratios for the proposed twin epicyclic gear transmission (Fig. 1) starts with selection of maximum possible vehicle linear speed V and corresponding wheels rotational speed ωW , where V= ωW r. Then the motor related subcomponents of the gear ratios in the expression (6) or (13) can be calculated based on equal share of final speed ωW between two identical motors (ωM = ωM1= ωM2) as described by (15). It can be expressed as follows:

$$VR_M = \frac{\frac{\omega_W}{2}}{\omega_M} = \frac{V}{2\omega_M \cdot r} \quad (19)$$

If for the sake of possible similarity of the gears in the transmission we assume that $VR_{M_1A_1} = VR_{M_2A_2} = 1$ then from (15) $VR_{1,3}$ and $VR_{3,5}$ can be defined from the following simple equations:

$$1 - VR_{3,5} = VR_M$$

$$(1 - VR_{1,3}) \cdot VR_{3,5} = VR_M$$

Or it can be rewritten as

$$VR_{3,5} = 1 - VR_M$$

$$VR_{1,3} = 1 - \frac{VR_M}{VR_{3,5}} \quad (20)$$

The engine related subcomponent of the gear ratios in the expression (6) or (13) can be selected based on the torque capability of the engine to provide enough acceleration up the hill with maximum slope angle α_{max} from Fig. 3, i.e. $\tan\alpha_{max} < \frac{\mu_P}{2} - f_r$. Then velocity ratio factor for the engine related subcomponent can be calculated from (13) as follows:

$$VR_E = VR_{E,1} \cdot VR_{1,3} \cdot VR_{3,5} = \frac{\left({T_E}/{2}\right)}{T_W} \quad (21)$$

T_E is a nominal torque developed by the engine at nominal\optimal engine shaft angular speed. Formula (21) assumes half of the nominal torque T_E at the commencement of the vehicle driving when engine idles and has not reached its nominal speed. Then $VR_{E,1}$ can be defines as follows:

$$VR_{E,1} = \frac{VR_E}{VR_{1,3} \cdot VR_{3,5}} \quad (22)$$

For the selection of number of teeth in the velocity ratios of twin epicyclic gear trains $VR_{1,3}$ and $VR_{3,5}$ the following derivation has been proposed:

$$VR_{1,3} = \frac{N_{1,2} \cdot N_{2,2}}{N_{2,1} \cdot N_{3,1}}; \; VR_{3,5} = \frac{N_{3,2}}{N_{4,1}} \cdot \frac{N_{4,2}}{N_{5,1}} \quad (23)$$

The embedded geometrical constrains in the structure of epicyclic gear trains are as follows:

$$N_{1,2} + N_{2,1} = N_{2,2} + N_{3,1};$$

$$N_{3,2} + N_{4,1} = N_{4,1} + N_{5,1} \quad (24)$$

In order to minimize the variation of gear sizes in the transmission we can accept the following equalities:

$$N_{1,2} = N_{2,2};\ N_{2,1} = N_{3,1}$$

$$N_{3,2} = N_{4,2};\ N_{4,1} = N_{5,1} \tag{25}$$

As a result, the single gear ratios within the epicyclic trains can be defined from (22) as follow:

$$\frac{N_{1,2}}{N_{2,1}} = \frac{N_{2,2}}{N_{3,1}} = \sqrt{VR_{1,3}} \tag{26}$$

$$\frac{N_{3,2}}{N_{4,1}} = \frac{N_{4,2}}{N_{5,1}} = \sqrt{VR_{3,5}} \tag{27}$$

VI. Control Logic for Switching Power Sources

The control logic for actuation of the three energy sources IC engine and motors M1, M2 in Fig. 1 is provided by actuation of three brakes B1, B2, B3. The switching between the motors M1, M2 and IC engine depends on the readings from the vehicle tilt angle α sensor. Vehicle is driven most of the time by motors M1 and M2 on horizontal or close to horizontal road. For these road conditions B2 is activated, B1 and B3 are not. When the tilt sensor detects that the road slop exceeds the predetermined value αM (18) then the controller switches the power supply from M1 and M2 to IC engine by activating B2 and deactivating B1 and B3. The deactivation of B1 and B3 takes place with some time delay tmin in order to provide a smooth transition of power line from one source to another. IC engine can run the vehicle at the speed limited by the gear ratio (20) and the road slope conditions $\tan\alpha_{max} < \frac{\mu_P}{2} - f_r$. Conversely, when the vehicle runs at $\alpha \leq \alpha_M$ the controller switches the power line from IC engine back to motors M1 and M2. The maximum desired speed of the vehicle is reached when both motors M1 and M2 run at their nominal speeds and torques.

The driving habit of the driver is constantly monitored by the wiper motion of a potentiometer that is rigidly connected to the accelerometer pedal of the vehicle. The potentiometer then controls the amount of voltage required to run motors M1, M2 and the motor that controls the opening of the IC engine throttle. In order to accommodate the intention of the driver to accelerate or decelerate fast enough the controller takes into the account not only the absolute position of the wiper but the rate of its motion as well. It can be implemented by PD control strategy in feed-forward loop of the controller subsystem. The developed control system is able to automatically trace the required vehicle driving conditions and make right and fast decisions on switching the power sources from one to another.

VII. Conclusion

The concept of power sharing between IC engine and two motors has been discussed in this paper. The use of three Degree of Freedom (3 DOF) twin epicyclic gear train provides an excellent condition to implement this idea in the most effective way. The paper describes the role of motors M1 and M2 as a source of shared electrical energy to drive smoothly a vehicle from minimum to maximum speed with moderate torques demand. Instead the IC engine remains in the vehicle as a source of energy to charge the batteries as well as to run the vehicle in case of substantial increase in torque demand (driving up the hill) or emergency drive conditions when there is a trouble with electrical power suppler subsystem. When using the system with a set of specially allocated brakes, the transmission can be employed in two distinct modes of operation: Motors and Engine. The introduced strategy for selection of individual velocity ratios VR within the transmission is an essential step in achieving all the objectives of power sharing while driving a vehicle. The introduction of two electric motors to share the vehicle load at distinct driving conditions as well as the use of petrol IC engine only to cope with heavy load or emergency conditions during the drive ultimately leads to fuel savings and pollution reduction.

References

[1] C. C. Chan, "The state of the art of electric and hybrid vehicles," *Proceedings of IEEE*, vol. 90, no. 2, pp. 247-275, February 2002.

[2] E. Mehrdad, G. Yimin, and E. Ali, *Modern Electric, Hybrid Electric, and Fuel Cell Vehicles—Fundaments, Theory, and Design*, 2nd ed, CRC Press, 2010.

[3] H. Iqbal, *Electric and Hybrid Vehicles: Design Fundamentals*, Boca Raton, Florida: CRC Press LLC, 2003

[4] K. Yong, "Power train for hybrid electricvehicle," United States Patent, Patent No. US7,478,690 B2, 2009.

[5] S. C. Tzeng, K. D. Huang, and C. C. Chen, "Optimization of the dual energy-integration mechanism in a parallel-type hybrid vehicle," *Applied Energy*, vol. 80, no. 3, pp. 225-245, February 2005.

[6] M. Koichiro, *et al*., *Development of New-Generation Hybrid System THS-II – Drastic Improvment of Power Performance and Fuel Economy*, Toyota Motor Corp., SAE Technical Paper Series, 2004.

[7] K. Seigo, *et al*., "Hybrid driving unit and vehicle carrying the same," United States Patent, Patent No. US7,426,971 B2, 2008.

[8] J. Edwards, "Dual-input infinite-speed integral motor and transmission device," United States Patent, Patent No. 5,067,932, 1991.

[9] Q. Ren, D. A. Crolla, and A. Wheatley, "Power management and Control strategies for a hybrid vehicle with a dual mode power split transmission," in *Proc. 5th IFAC Symposium on Advances in Automotive Control*, Part 1, Seascape Resort, USA, 2007, vol. 5, pp. 111-118.

[10] K. Ahn, S. Cho, W. Lim, Y. Park, and J. M. Lee, "Performance analysis and parametric design of the dual-mode planetary gear hybrid powertrain," *Proc. IMechE, Part D*: *Journal of Automobile Engineering*, vol. 220, no. 11, pp. 1601-1614, November, 2006.

[11] H. Sunghwa, C. Woulsun, A. Sunhhyun, K. Yongjoo, and K. Hyunsoo, "Mode shift control for a dual-mode power-split-type hybrid electric vehicle," *Proc. IMechE, Part D: Journal of Automotive Engineering*, vol. 228, no. 10, pp. 1217-1231, September 2014.

[12] J. Kim., Y. Kang, T. Kim, B. Min, and H. Kim, "Design of power split transmission: Design of dual mode power split transmission," *International Journal of Automotive Technology*, vol. 11, no. 4, pp. 565-571, 2010.

[13] D. A. Crolla, Q. Ren, S. ElDemerdash, and F. Yu, "Controller design for hybrid vehicles – state of art review," in *Proc. IEEE Vehicle Power and Propulsion Conference*, Harbin, 2008, pp. 1-6.

[14] B. Mashidi and A. M. Emadi, "Dual-mode power-split transmission for hybrid electric vehicle," *IEEE Trans. on Vehicular Technology*, vol. 59, no. 7, pp. 3223-3232, September 2010.

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