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EPICYCLIC POWER SPLIT TRANSMISSIONS FOR HYBRID ELECTRIC VEHICLES FUEL CONSUMPTION,VEHICLE PERFORMANCE AND DRIVING AGGRESSIVENESS

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Ahmed Elmarakbi [α], Qinglian Ren [σ] & Rob Trimble [ρ]

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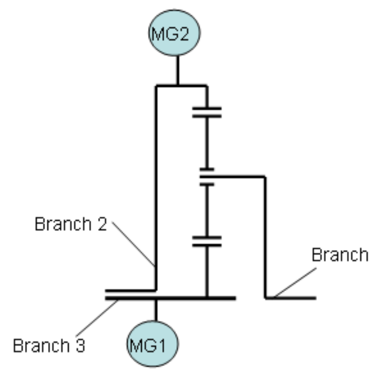
I. Introduction

Continuously Variable Transmissions (CVTs) have been around for many years and the cost-benefit issues relating to CVTs are well understood. The potential advantages are improved performance, economy and emissions or more importantly an improved compromise between them. Their disadvantages have been cost, complexity, noise and driving refinement. Only over the past five years or so has the development of CVTs reached a stage at which they are beginning to be genuinely competitive with the alternatives, e.g. conventional, torque converter automatics and automated manual gearboxes, such as the twin clutch VW DSG system.

An important type of CVT is the E-CVT (Electronically-Controlled Continuously Variable Transmission), a good example of which is Toyota Hybrid System (THS). It combines the characteristics of an electric drive and a continuously variable Transmission, using motor generator units in addition to toothed gears (Toyota Prius Transmission, 2006; Miller and Miller, 2005). In a THS system (see figure 1), one of the motor generators (MG2) is mounted on the driveshaft, and thus couples torque into or out of the driveshaft. The second motor generator (MG1) is connected with the sun gear and used to change the sun gear speed. Because MG2 is connected with the driveshaft, it cannot change speed and torque freely. Hence there are three power input/output branches in the system: the engine, MG1, the output, MG2. Because the speed of the output shaft is decided by the speed of the vehicle, there is some limitation on the control strategy to achieve optimum performance.



Branch 1- engine input shaft; Branch 2 – output shaft, also connected with MG2; Branch 3 – shaft connected with MG1

Fig. 1 : Three branch system

In a four branch transmission system, which is presented in this paper, neither of the motor generator units is mounted on the driveshaft or on the engine input shaft, which gives more freedom and benefits to the system (see figure 2).

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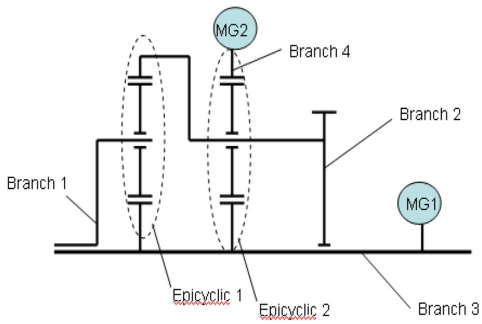

Branch 1- engine input shaft; Branch 2 – output shaft; Branch 3 - connected with MG1; Branch 4 – connected with MG2

Fig. 2 : Four branch system

One motor/generator is connected to the sun gear, and the other motor/generator is connected with a ring gear. So there are four branches of power input/output: the engine, the output shaft, and two motor generator units, MG1 and MG2.

This type of four branch transmission system has been described recently by Moeller (2006) who proposed that it offers advantages in many automotive applications. However, its usage in a hybrid electric vehicle driveline will be studied here.

A matrix method is used here to analyse the planetary transmission system, as introduced by Tian and Lu (1997). The key point of this method is to generate matrices to represent all the planetary train elements and other auxiliary components of the transmission.

II. Mathematical Modeling

The modeling of the hybrid electric vehicle performance is carried out using the QSS Toolkit (Guzzella and Amstutz, 2005). This is a quasistatic simulation package based on a collection of Simulink blocks and the appropriate parameter files that can be run in any Matlab/Simulink environment. The traditional ICE vehicle model itself is straightforward and is shown in figure 3.

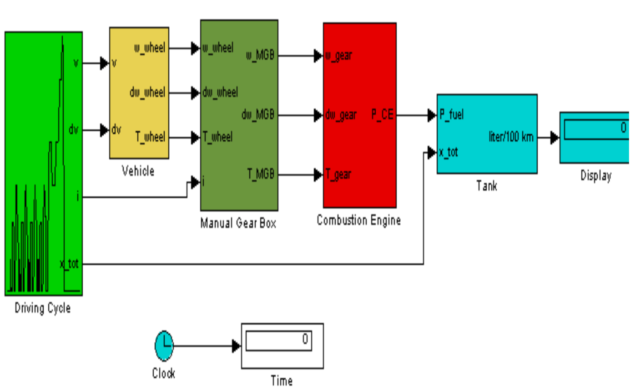

Fig. 3 : Overview of the conventional ICE vehicle model

There are 5 sub-systems: the driving cycle subsystem, vehicle subsystem, the gearbox subsystem, the combustion engine subsystem, and the fuel tank subsystem. The data for the engine and gearbox are taken from generic data in the QSS package. The function of the engine subsystem is to compute the fuel consumption from a consumption map, according to the torque and the rotational speed demand. The gearbox has 5 gears ranging from 3.84 to 0.63. The differential gear ratio is 3.95. The other vehicle data are shown in table 1. It is not intended to represent any specific vehicle – but rather to act as a generic vehicle platform to focus attention on the differences obtainable from the three vehicle models.

Table 1 : Vehicle Parameters (Miller, 2004)

Vehicle curb weight	1257 kg
Drag coefficient, C_d	0.29
Frontal Area	2.23 m^2
Tyre radius	0.292 m
Final drive	3.95:1

The models for the two hybrid electrical vehicles are built based on the same baseline vehicle, with the same vehicle parameters, but different transmissions, as shown in figure 4. The input to the model is one of the standard driving cycles – the NEDC and USA FTP-75 cycles are used extensively in this work – and the solution procedure is based on stepping through the driving cycle at typically one second steps, calculating the equilibrium condition and then collecting all the data for plotting at the end of the cycle. Thus, the focus of attention is on the overall efficiency of the engine and motor generator units and the major issue of whether it is possible to improve overall energy usage by operating the whole system at or near to the best efficiency points.

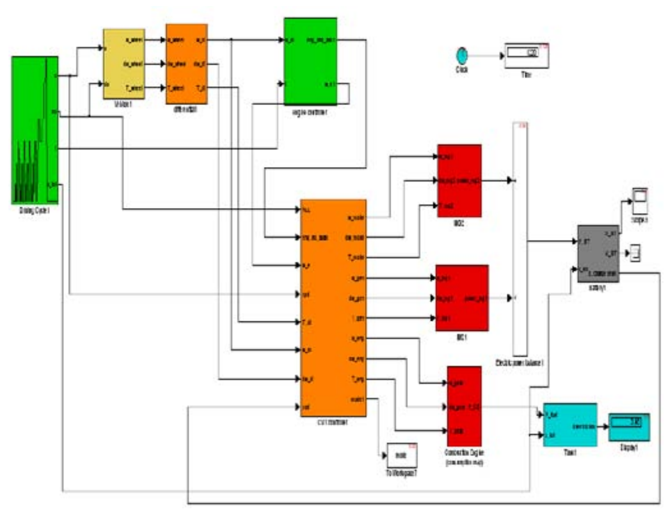

Fig. 4 : Overview of the hybrid vehicle model

III. Control Of Hybrid Vehicle Drivelines

The overall control of a hybrid drive train incorporating a power splitting transmission has proved to be an enormous challenge (Ren, and D. Crolla, 2007). This has already been shown to be the case for the first generation, single mode systems (Inoue et al., 2000), but looking at the complexity of the emerging dual mode systems (Cho et al., 2006), it is set to become even more challenging.

The overall control strategy can be summarized in broad terms as:

- Maximizing fuel economy
- Minimizing emissions
- Maintaining battery state of charge (SOC)
- Providing good performance and drivability

Central to these control objectives is the careful management of the power flows through the electrical and mechanical parts. Optimizing overall efficiency depends on a difficult compromise between operating the engine and transmission simultaneously around their most efficient conditions. The strategy for controlling the engine has received an enormous amount of attention, since this is an important issue irrespective of what transmission is used. Much emphasis has, not surprisingly, been placed on minimizing the brake specific fuel consumption (BSFC).

An overview of the high level, supervisory controller is shown in figure 5. Usually, the system components, e.g. engine, motor, generator etc. will also have their own local feedback loops and these are not considered here. Whatever the details of the actual controller, the approach used here is to combine it with a MATLAB model (e.g. figure 6) of the vehicle to compare the performance of alternative control strategies.

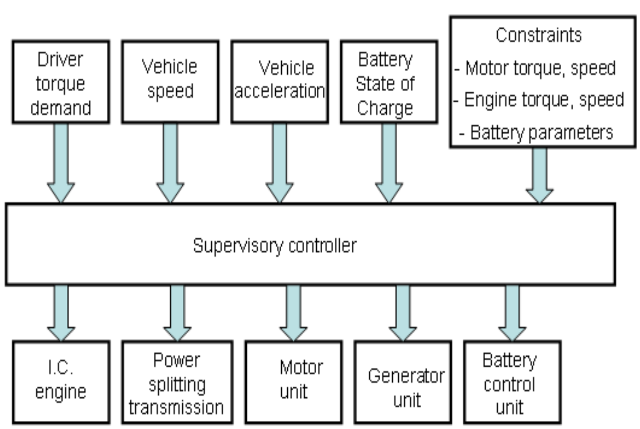

Fig. 5 : Controller Model

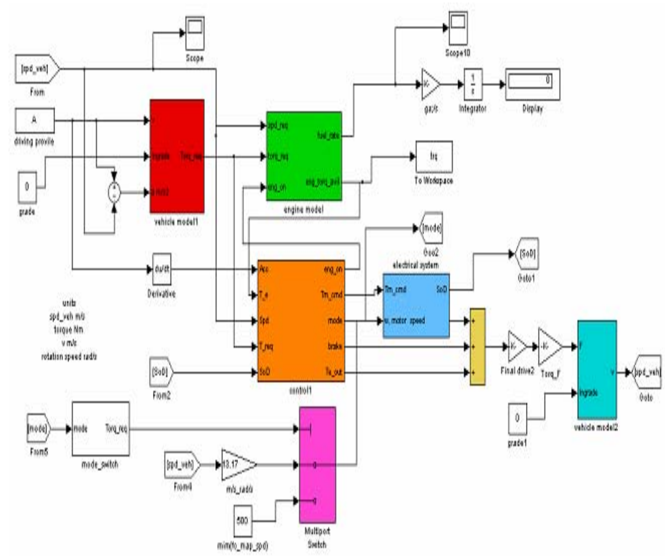

Fig. 6 : Vehicle Model

The control problem is often referred to as an optimal control problem, although this is often used in a general sense as opposed to a strict mathematical definition. The problem may be characterized in many ways depending on the performance objective, the hybrid vehicle model considered, the constraints imposed and the available control actions. However, as a general dynamics and control problem it is clearly not straightforward, and is likely to involve combinations of linear and non-linear elements, discrete and continuous systems, algebraic and dynamical systems

IV. Fuel Consumption

For HEVs, the difference between the initial and final battery SOC can significantly affect the measurement of fuel economy. To eliminate this effect, the concept of 'overall fuel consumption (OFC)' was introduced. The total additional energy stored or drawn from the battery (kWh) is calculated and then converted into how much fuel (liter) would be used for the engine to produce this amount of energy.

Engine fuel consumption (EFC, litre/100km): actual fuel burned by the engine divided by the driving distance Overall fuel consumption (OFC, liter/100km): the fuel consumption after taking the battery energy changed (BEC) into consideration

$$OFC = EFC + \frac{100 \times BEC \times \eta_{eng}}{\rho} / D \qquad (1)$$

in which ρ is the fuel density (g/ml), η_{eng} is the engine efficiency (g/kWh) and D is the driving distance (m). The values for ρ and η are 0.76 g/ml and 240 g/kWh, respectively.

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In the simulation, BEC is positive if energy is drawn from the battery and negative if the energy is stored into the battery. So at the end of each driving cycle, if final SOC is smaller than the initial SOC, namely the energy is drawn from the battery, overall fuel consumption is greater than the engine fuel consumption, and vice versa.

It is very important to take account of the battery SOC in the calculations, because if it is different at the end of the driving cycle from its value at the start then some net energy has effectively been lost or gained in the vehicle calculations. In several examples of results in the literature, it is not clear whether this effect has been accounted for. Also, some researches actually use the control system to ensure that the battery start and finish conditions are exactly the same. However, this can cause difficulties because the control system is not necessarily representative of what it would be doing during normal practical driving.

The first set of results was used to compare the two PST arrangements with a baseline, conventional vehicle equipped with a five speed gearbox (3.84, 2.11, 1.36, 0.86 and 0.63 with the same final dive ratio). The control strategies for the two PST arrangements were based on a rule-based approach to compromise between overall energy efficiency and maintaining the battery state of charge (SOC) under control. The vehicle models were run over NEDC and USA FTP-75 driving cycles, and the overall fuel consumption results are shown in table 2.

As expected, both hybrid vehicles show economy advantages over the conventional, manual gearbox vehicle. However, the improvements are not as great as published in some other studies, but this is understandable because the systems used here – and in particular their controllers – have not yet been optimized. The main aim of this work was rather to compare the 3 and 4 branch drivelines under exactly comparable conditions; this comparison is shown in table 3 and it shows that the 4 branch system offers around significant improvements over all the driving cycles. The improvements vary substantially with the different cycles, varying from 7.4% for the NEDC to 20% over the USA FTP-75 cycle.

Table 2 : Fuel consumption for the hybrid vehicle fitted with the 3 and 4 branch systems compared with a conventional, manual gearbox vehicle over different driving cycles

Driving cycle	Fuel consumption over driving cycle, l/100km				
	Traditional ICE	Single epicyclic system (3 branch system)		Dual epicyclic system (4 branch system)	
		Engine FC	Overall FC	Engine FC	Overall FC
Europe NEDC	3.8	4.0	2.7	3.9	2.5
USA FTP-75	3.6	3.8	3.0	3.7	2.4

Table 3 : Percentage improvement of the 4 branch over the 3 branch driveline over different driving cycles

Driving cycle	Percentage improvement of 4 branch over the 3 branch driveline (%)
Europe NEDC	7.4
USA FTP-75	20.0

The associated engine utilization maps are shown in figures 7 - 12 for the baseline gearbox, the single epicyclic gearbox and the twin epicyclic gearbox vehicles, respectively. Each point on the map of engine torque vs. speed is the solution at a single point during the NEDC cycle; the cycle defines input from t = 0s to t = 1220s. However, the NEDC cycle contains a percentage of constant speed running conditions, so that several points will sit on top of each other.

First, these results highlight in figures 7 and 10 the shortcoming associated with conventional IC engine cars – namely that they inevitably spend considerable time at part load conditions well away from the areas of maximum efficiency. In contrast, it can be seen in figures 8 and 11 for the single epicyclic gearbox, that it manages the IC engine rather well through the combination of effectively its continuously variable gear ratio plus its ability to manage the electrical power flows in and out of the battery. This results in the usage points being constrained around the area of maximum specific fuel consumption of the engine. Finally, in figures 9 and 12 it can be see that the dual epicyclic gearbox actually manages some further improvement and also reduces the use of the higher engine speeds.

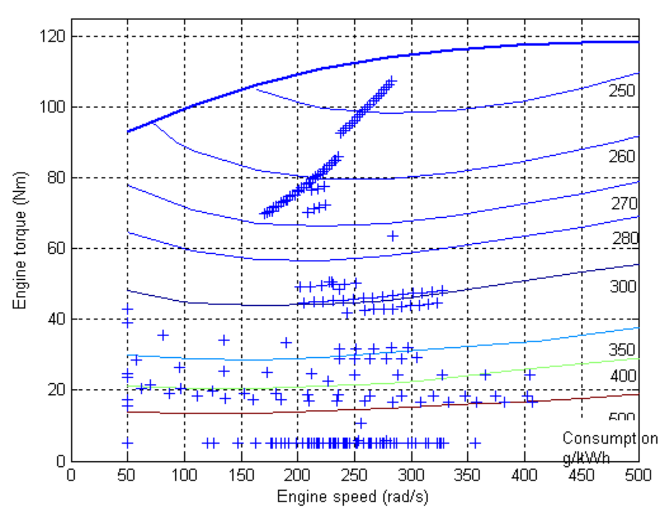

Fig. 7 : Engine operation points, NEDC cycle, traditional ICE vehicle

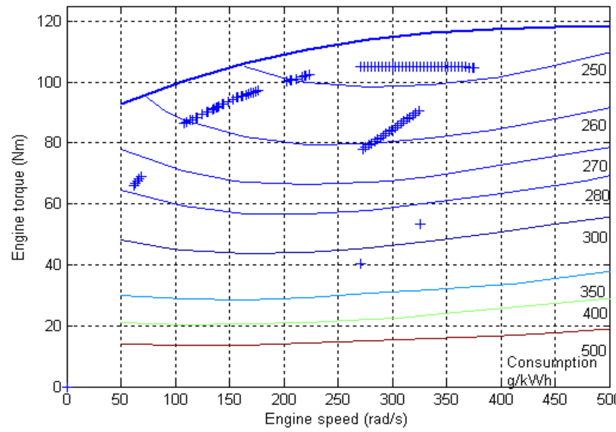

Fig. 8 : Engine operation points, NEDC cycle, single epicyclic syste

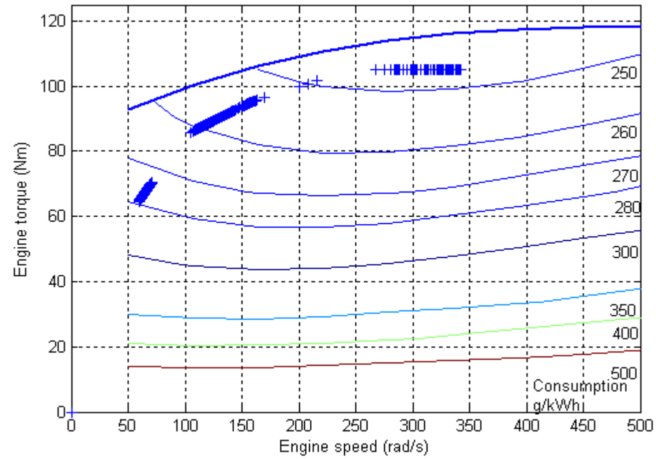

Fig. 11 : Engine operation points, FTP75 cycle, single epicyclic system

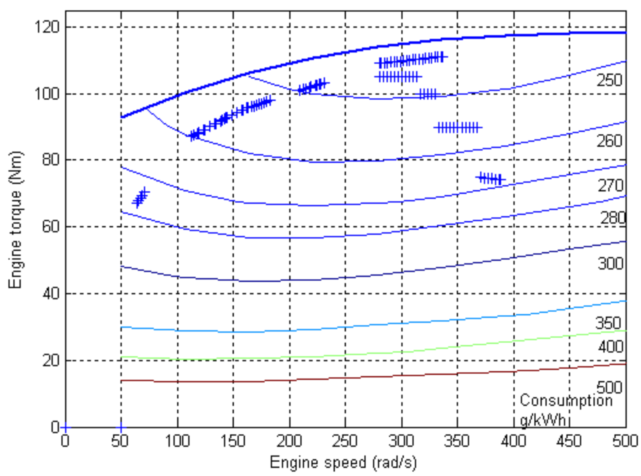

Fig. 9 : Engine operation points, NEDC cycle, dual epicyclic system

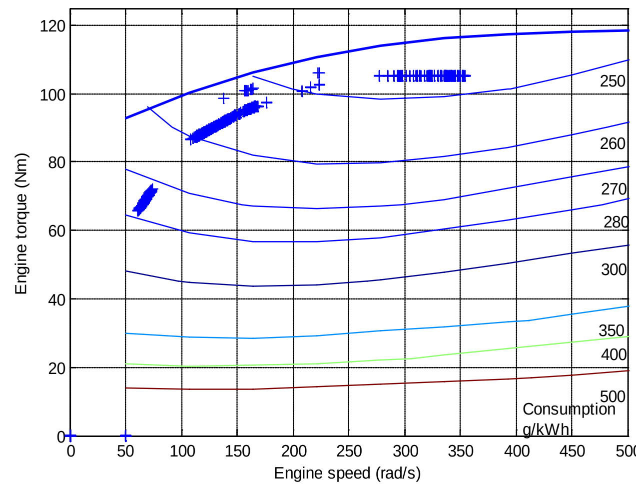

Fig. 12 : Engine operation points, FTP75 cycle, dual epicyclic system

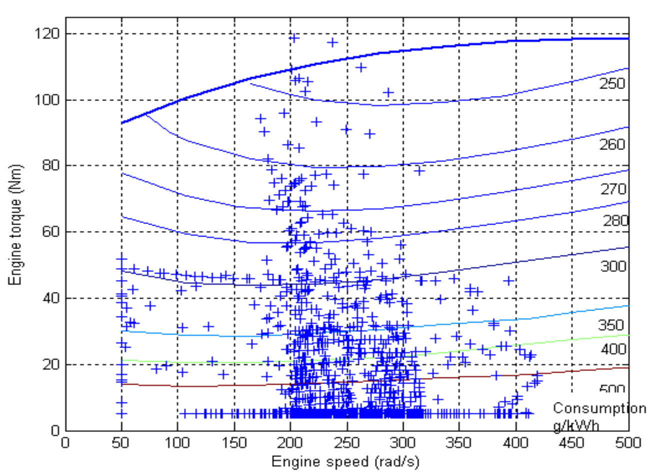

Fig. 10 : Engine operating points, FTP 75, traditional ICE vehicle

Further insight into the detailed behavior of the single and dual epicyclic gearboxes can be seen in the time history plots in figures. 13 and 14 for the NEDC cycle and figures. 15 and 16 for the USA FTP-75 cycle. The power utilization of the IC engine and two motor generator units, MG1 and MG2 are plotted along with the vehicle speed profile specified in each of these driving cycles.

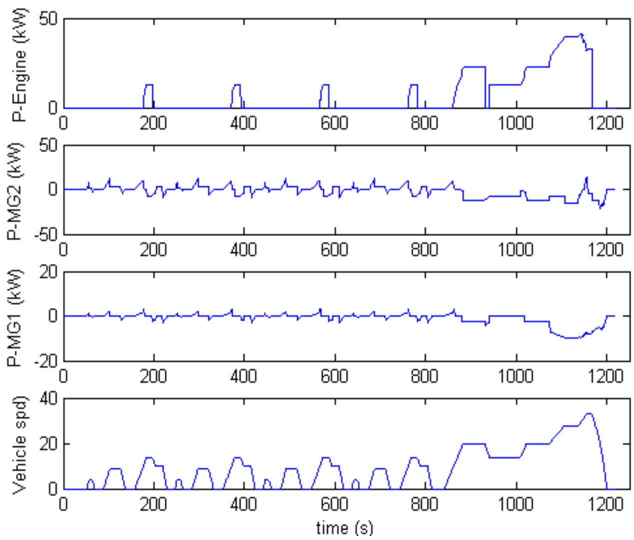

Fig. 13 : Power flows in the HEV with the single epicyclic gearbox over NEDC driving cycle

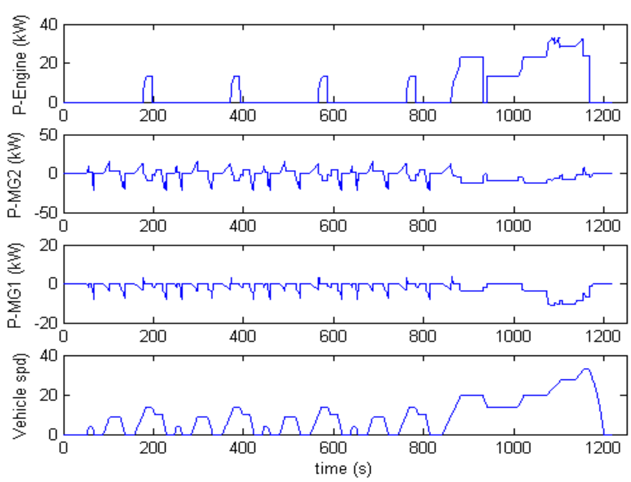

Fig. 14 : Power flows in the HEV with the dual epicyclic gearbox over NEDC driving cycle

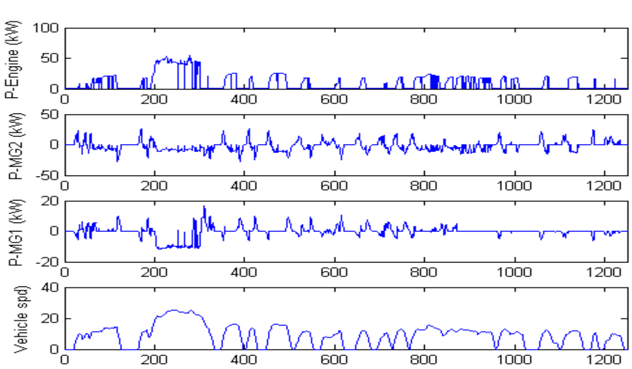

Fig. 15 : Power flows in the HEV with the single epicyclic gearbox over FTP-75driving cycle

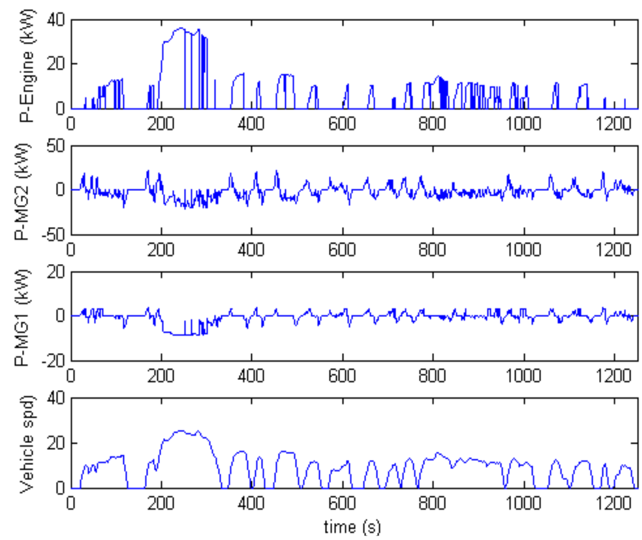

Fig. 16 : Power flows in the HEV with the dual epicyclic gearbox over the FTP-75driving cycle

The results in figures 13 to 16 also highlight the substantial differences between the European and USA standard driving cycles; the European version contains a much greater number of stationary and constant speed running events, whereas the USA version is almost continually changing speed in its version of an urban cycle. This reflects a fundamental difficulty within the vehicle industry when it has to try and make fair comparisons of competing driveline technologies regarding their energy consumption – what actually constitutes a representative driving pattern over which to make comparisons? The answer is likely to vary across the three major global automotive markets – Europe, USA and the Far East.

The acceptance of a so-called standard cycle also raises another potential problem – that in the quest for the best headline figures, the driveline and particularly its controller is actually optimized around the specific cycle. This can lead to engineering developments based more around the standard than around a drivable, efficient vehicle across a wider range of operating conditions.

V. Vehicle Performance

a) Top Speed

For conventional IC engine vehicle, the top speed that can be reached on level road with a given transmission ratio can be found by intersecting the curve of the available power at the wheels with that of the required power on level road (Genta and Morello, 208). The available power for gears I, II, III, IV and V are calculated as: for the given gear ratio, assume the engine always works on the "engine speed-maximum engine torque" curve.

$$P_a = \eta_t P_e = \eta_t \omega_{eng} \times T_{eng\_max} \quad (2)$$

$$P_n = (F_{rr} + F_{ad}) \times V \quad (3)$$

Where P_a is the available power, P_n is the required power, T_{eng_max} is the engine's maximum torque, F_{rr} is the rolling resistance force, F_{ad} is the aerodynamic drag, and η_t is the efficiency of the whole powertrain. The vehicle speed under each gear is

$$V = \frac{\omega_{eng}}{i_{gear} \times i_{differential}} \times r \quad (4)$$

where r is the radius of the tire, i_{gear} is the transmission ratio of each gear, and $i_{differential}$ is the transmission ratio of the differential. The power required and the available power for each gear ratio are shown in figure 17. For the IC engine vehicle the top speed is 54.5 m/s (196.2km/h). For full hybrid vehicles, if the engine is not downsized, the vehicle is almost retaining to have better overall performance compared with a conventional vehicle (Chan, 2007). As far as the top speed is concerned, because the power split device is actually a CVT, it can change the transmission ratio to achieve the highest top speed, as shown in figure 18. The top speed for the HEV with an E_CVT is 58.6 m/s (210.9 km/h).

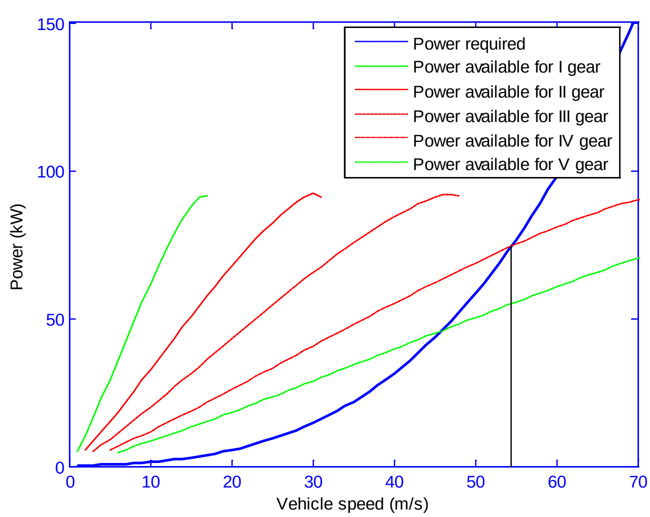

Fig. 17 : Top speed for the conventional vehicle (55.92m/s)

In practice, for a full HEV it is normal design practice to downsize the IC engine. The challenge for the controller design is then to maximize the time spent by the engine toward its optimum efficiency region by controlling the power flows to and from the battery. However, this should not compromise overall performance and drivability compared to the equivalent IC engine vehicle. So the controller exploits the continuously variable transmission ratio to address this compromise.

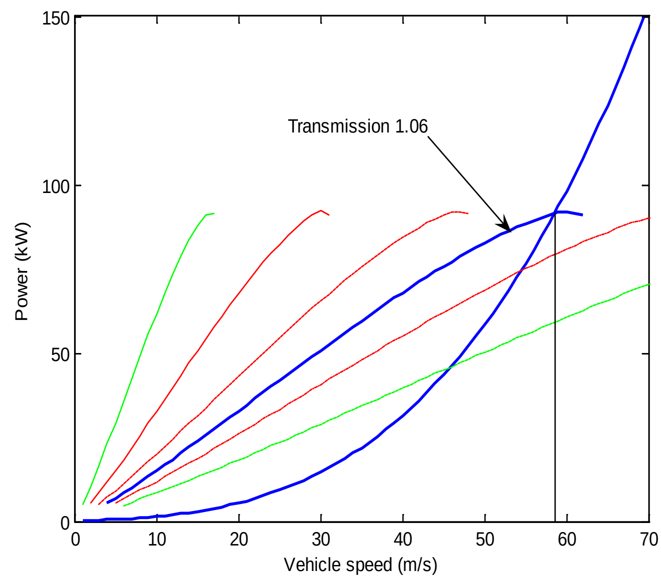

Fig. 18 : Top speed for a HEV with an E-CVT

b) Acceleration

The maximum acceleration a vehicle is capable of at various speeds is [9]:

$$\left(\frac{dV}{dt}\right)_{\max} = \frac{\eta_t P_e - P_n}{m_e V} = \frac{\eta_t P_e - (F_{rr} + F_{ad})V}{m_e (V+1)} \quad (5)$$

where m_e is the equivalent mass of the vehicle, P_e is the engine power, and P_n is the required power. The plot of maximum acceleration versus vehicle speed of the tradition IC engine vehicle with a five speed gear box is shown in figure 19. The minimum time need to accelerate from speed V_1 to V_2 can be calculated by integrating Eq. (5), but usually numerical integration is performed. A graphical interpretation of the integration is shown in figure 20, which plots (1/acceleration) vs. vehicle speed. The area under the curve 1/a is the minimum time required for the acceleration.

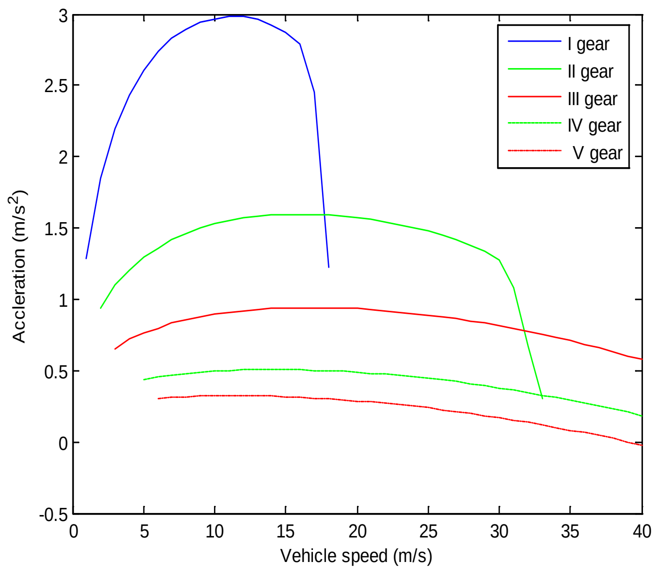

Fig. 19 : Maximum acceleration vs. vehicle speed

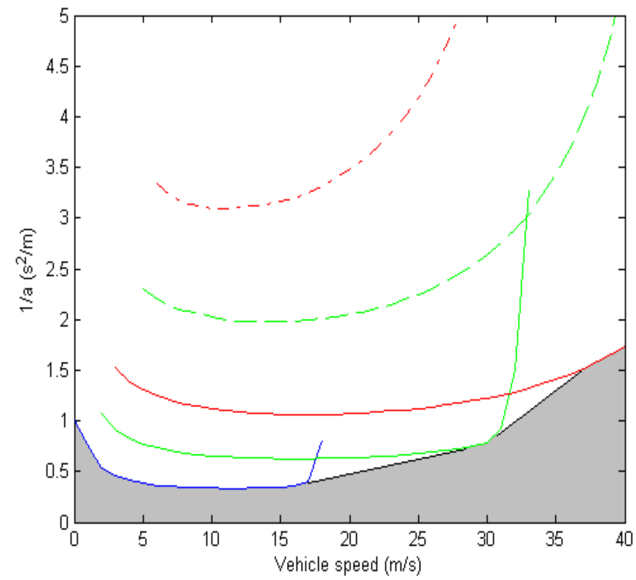

Fig. 21 : 1/a versus vehicle speed

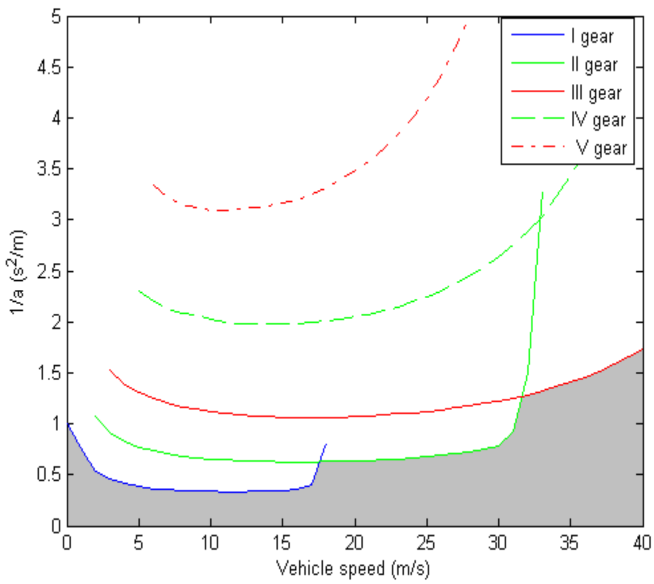

Fig. 20 : 1/a versus vehicle speed

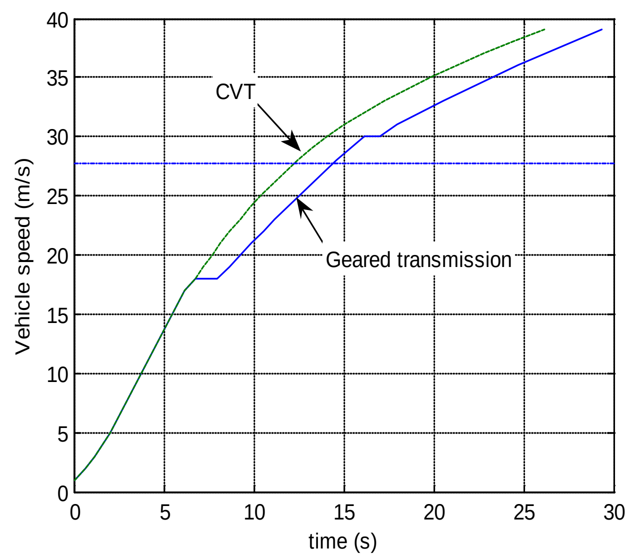

Fig. 22 : Speed vs. time curve

The minimum time of acceleration for a HEV with a PST (power split transmission) is shown in figure 21. The dark area is the time to speed for a HEV with an E-CVT. The time-speed curve, as shown in figure 22 can be obtained by integrating the dark area in figures. 20 and 21. From figure 22, it shows that the time to accelerate from 0 to 100 km/h (27.78 m/s) for a vehicle with a geared transmission and a HEV with a PST is 12.3 s and 14.4 s, respectively.

VI. Driving Aggressiveness

A study at the Argonne laboratory demonstrated that HEVs have higher fuel consumption sensitivity to aggressive driving (Sharer, 2007). They define more aggressive driving to mean more use of periods of higher accelerations that indicated in the typical driving cycles. In this study, the sensitivity of the two HEV models to driving aggressiveness was calculated and the results are shown below.

The FTP 75 driving cycle is used as a baseline driving input for the comparisons. Then, a simple multiplier factor of 0.8. 0.9, 1.1 and 1.2 was imposed to get cycles that represent different driving aggressiveness. The bigger the factor, the more aggressive the driving cycle becomes since the speeds and hence accelerations are simply increased. Factor

0.8 means the cycle is less aggressive than the baseline cycle and factor 1.2 means the cycle is more aggressive than the baseline cycle, as shown in figure 23. The simulation results are summarized in table. 4.

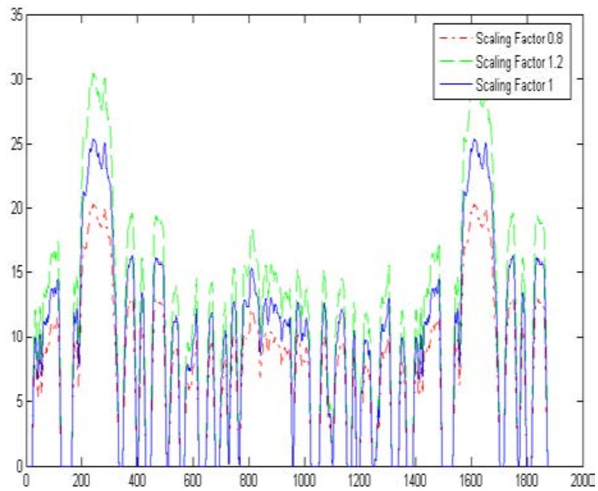

Fig. 23 : Driving cycles with different driving aggressiveness

Table 4 : Fuel consumptions with different driving aggressiveness

Scaling factor	Traditional ICE		Single epicyclic system		Dual epicyclic system	
	Overall FC	Norm-alized FC	Overall FC	Norm-alized FC	Overall FC	Norm-alized FC
0.8	3.39	0.93	3.07	0.84	2.64	0.73
0.9	3.50	0.96	3.03	0.83	2.37	0.65
1.0	3.64	1.0	3.02	0.83	2.36	0.65
1.1	3.79	1.04	3.57	0.98	2.64	0.72
1.2	3.97	1.09	4.42	1.21	3.22	0.88

Note: all the normalized fuel consumptions are normalized to 3.64, which is the overall fuel consumption of the traditional ICE with scaling factor 1.

The overall benefits of the dual epicyclic over the single epicyclic and the traditional ICE are shown to occur consistently over all limited range of driving aggressiveness tested here. Then, the same results are plotted in figure 24 showing the trend of the fuel consumption figures normalized to the traditional ICE at a scaling factor of 1. These curves provide information on how sensitive each system is to driving aggressiveness as indicated by the slope of the lines. The ICE case is shown to be rather insensitive; the twin epicyclic is somewhat less sensitive than the single epicyclic as the driving aggressiveness increases above 1. This again relates to the greater flexibility of managing the power flows between the mechanical and electrical paths.

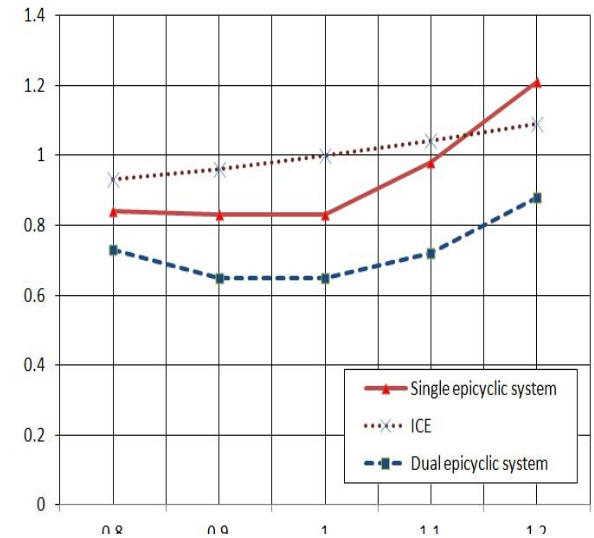

Fig. 24 : Sensitivity to driving aggressiveness – fuel consumption normalized to the baseline ICE condition

VII. Conclusions

The following conclusions are made:

- The four branch dual epicyclic gearbox arrangement offers a significant performance benefit over the three branch, single epicyclic arrangement; fuel economy improvements of 7 and 20% were shown over the two main European and USA driving cycles.
- The performance benefits arise from the greater flexibility of control over the torques, speeds and power flows through the two motor generator units available with the dual epicyclic scheme.
- For a HEV with a PST, which provides the benefits of a CVT gearbox, if the engine is not downsized, the HEV will have better drivability, namely higher top speed and shorter acceleration time.
- In practice, the normal design approach for a HEV is to downsize the engine, and improve overall fuel consumption, whilst exploiting the transmission and controller properties to obtain similar performance to the equivalent ICE vehicle.
- The four branch system is slightly less sensitive to increased driving aggressiveness than the 3 branch system, because it can control the power flows better according to different driving conditions.
- In practice, further benefits are probably available; first, the dual arrangement has two nodal positions at which zero electrical power circulates and these can be designed to occur at convenient speeds, e.g. in the UK, 30 mile/h in urban driving and 70 mile/h motorway cruising. Second, with the dual arrangement it is possible to downsize the motor

generator units to retain the same driveability but with reduced weight and cost.

- Although the results presented here have been based on relatively simple vehicle models, it is likely that the promising results for the twin epicyclic transmission could be further improved using more sophisticated optimisation strategies for the control system.

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