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A Novel Dual-Motor Two-Speed Direct Drive Battery Electric Vehicle Drivetrain

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ABSTRACT Although the demand of battery electric vehicle (BEV) growths fast as the requirement of reducing greenhouse gas emission and the usage of fossil fuels, the limited driving range and unfriendly retail price present barriers to BEV to provide comparable performance as a traditional vehicle. This paper proposes a dual-motor two-speed direct drive BEV powertrain to boost average motor operational efficiency in daily driving without increasing any complexity of manufacturing or control, ultimately, saving limited battery energy and manufacturing cost. The specifications of the proposed powertrain are first identified through mathematical and graphical calculations, which split traditional one propelling motor to two with separate permanent engaged gears to maximize the motor efficiency. Based on dynamic powertrain modeling in a Simulink/Simscape, economic shifting strategy, and dynamic torque transfer control are designed and tested. According to the simulation results, it is noticed that significant energy efficiency improvement can be achieved. Thanks to the optimized torque transfer control strategy, extremely low vehicle jerk are recorded during the shifting process. At last, conclusions can be made that the proposed dual-motor powertrain superior to the traditional single motor counterpart in terms of fuel economy, driving range, and cost.

INDEX TERMS Dual motor, two speeds, electric vehicle, dynamic modeling, energy economy.

I. INTRODUCTION

Nowadays, Battery Electric Vehicles (BEVs) attract greater attention and become more and more popular. However, comparing to the traditional vehicles, the relatively short driving range per charge [1]–[3], longer 'refill' time [4] and unfriendly price [5] still presents significant barriers for its large-scale commercialization. Although the outstanding dynamic and efficiency performance of electric machine (motor), such as 100% torque available from standstill and higher average energy converting efficiency comparing to internal combustion engine (ICE), enable a direct drive from motor to wheel through a fixed ratio gear reduction, the demand of energy efficiency improvement and drivability enhancement has stimulated a lot of academics and manufacturers to develop a range of multi-speed transmissions for BEVs. A two-speed Dual Clutch Transmission (DCT) along with hybrid energy storage system was proposed in our previous work for BEV [6], [7]. Similar proposals with various multi-speed transmissions implementations in BEV, like Automatic Transmission (AT), Automated Manual Transmission (AMT) and Continuously Variable Transmission (CVT), can be found in [8]–[10]. Results from above studies clearly demonstrated that multi-speed transmission not only improve the dynamic and economic performance of BEVs, also ease the high upfront cost by a cheaper daily cost and maintenance fee from the viewpoint of long-term ownership. Extra benefit is also available in regenerative braking through gear shifting [11]. However, the complex mechanism, e.g. additional gear pairs, inefficient hydraulic system and synchronizer for gear shifting, and extra transmission manufacturing cost present the barriers for successfully turning the ideas to reality [12]. Even if some specially designed transmissions are capable of achieving smooth gear shifting by electrifying the torque transferring process [13], the requirements of fast and accurate control for motor and actuators require significant calibration and field testing.

Instead of improving BEV motor efficiency through adding extra gears, some power-split systems were proposed to realize the same function of multi-speed transmission,

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which use motor control to change gear and transfer torque between gears. Peng and He introduced a dual-motor drivetrain, which uses two driving motors and a dual-input and dual-output transmission to achieve several driving modes, e.g. high speed four-wheel-drive (FWD), low speed FWD and real wheel drive, through two shifting motors [14]. It is a smart structure for FWD EV, however, it is not an energy friendly structure for popular two-wheel-drive EV considering the six gears, two synchronizers and two shifting actuators components. Zhu *et al.* proposed a two-speed two-motor drivetrain with a one-way clutch and a two-part shaft (inner & outer like DCT) [15]. The efficiency of this proposed powertrain is relatively high, comparing to planetary gears, as the price of high control requirements and manufacturing cost. A two-motor powertrain proposed by Hu *et al.* claimed about 4% motor efficiency improvement through a planet gear based power split system [16]. The switching of driving modes between single motor drive and combined motor drive in terms of speed and torque depends on a planetary gear unit, a fixed gear unit, two clutches and one synchronizer, which makes the structure complicated. Another dual-motor coupled powertrain was applied in battery electric bus with Dynamic Programming (DP) method by Wang *et al.* to save the cost of motor, at the same time, boost energy efficiency [17] and reduce the impact of motor torque mutation on transmission system [18]. However, the coaxial dual-motor powertrain structure can only provide a 'add-on' torque, rather than selecting propelling motor for efficiency. Comparing to implementing a traditional multi-speed transmission in BEV, the advantages of above dual-motor proposals reduce the requirement of transmission control to achieve quality shifting, in which motors are actively involved with engaged gear in power-splitting system to realize smooth gear shifting and driving mode selection. Furthermore, the structures increase the flexibility of the speed/torque coupling modes from multiple power source. However, clutches and/or synchronizer are still essential elements in above proposals, which makes little difference to the traditional transmission application in BEV in terms of manufacturing cost and mechanical efficiency improvement. Additionally, neither the motor nor the corresponding gear ratio of above proposals was specially designed to provide better dynamic and economic performance.

Given above various powertrain structures proposed for multiple power source EVs, it is clear that most of them rely on clutches and synchronizers to achieve their functions. In this study, aiming at making the most of motor excellent torque delivery ability, at the same time, simplify the mechanical structure as much as possible, a novel Dual-Motor Two-Speed (DMTS) direct drive drivetrain is proposed to improve motor operational efficiency without increasing the complexity of control and mechanism. Unlike other traditional transmission or power-splitting system based multi-speed BEV powertrains, this proposal implements a parallel energy flow layout in BEV, which is a popular hybrid electric vehicle powertrain to manage the power flow from engine

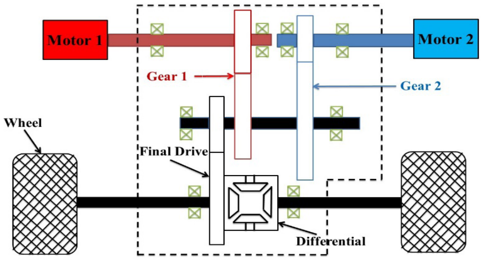


FIGURE 1. Schematic layout of DMTS direct drive EV powertrain.

and motor. The proposed layout has the potential to eliminate the requirement of clutch, synchronizer and hydraulic system to achieve gear shifting or driving modes switching. The function of shifting gears is realized by switching propelling motors with permanent engaged gear pairs, as shown in Fig.1. The shifting process is fully controlled by motor, ultimately, improve powertrain efficiency and reduce mechanical complexity. Furthermore, splitting one propelling motor to two increase the possibility of higher motor operational efficiency via optimized design of motor torque-speed characteristic and engaged gear ratios. The reason of adopting a two-speed powertrain, rather than others, is that it can achieve the best balance of motor efficiency improvement and additional cost [10]. Through the proposed powertrain and related control strategy, the scientific contribution of this paper will fill the knowledge gap of existing research in following aspects:

1. The popular 'one motor + one speed' BEV powertrain is replaced by two smaller motors with approximate same total output power in this paper to improve overall motor efficiency.
2. The determination process of motor torque-speed profile in proposed powertrain provides the experience and guidelines on energy efficiency-oriented motor specifications design;
3. Aiming at efficiency improvement and smooth torque transferring, the control strategy of alternative motor propelling is designed and optimized;
4. With the special designed control strategy, the 'multi-speed' function is realized by switching driving motors with permanently engaged gear pairs, rather than implicating mechanical shifting actuator, i.e. clutch or synchronizer;

In the rest of the paper, powertrain specifications, including motor capacity and gear ratios, are determined to provide a comparable performance to the benchmarking one-motor powertrain. Then, driving motor switching, namely gear shifting in this powertrain, and torque handover control strategies are proposed. Next, a dynamic modeling is built in Matlab/Simscape® to simulate vehicle performance in typical driving cycles and verify the effectiveness of proposed control strategies. At last, based on analysis of energy efficiency, torque transfer quality in motor switching and potential economic benefit, conclusions are made that proposed dual-motor two-speed direct drive powertrain can provide

BEV substantial benefit in performance and cost for both manufacturers and customers.

II. POWERTRAIN CONFIGURATIONS- WHY WE NEED TWO MOTORS INSTEAD OF ONE WITH SAME POWER

The schematic layout of proposed novel DMTS direct drive powertrain is shown in Fig.1, which does not adopt any clutch or synchronizer. Each motor is connected to the final drive shaft via a fixed ratio gear pair respectively.

Given the outstanding torque capability of motor with 100% available torque from standstill, all the market available EV use a trade-off fixed-ratio single speed transmission to reduce motor speed and amplify torque at the same time. The speed and torque of motor is determined by vehicle condition, in consequence, it could not achieve high efficiency performance everywhere.

Regarding the motor performance of the specification Table.A1 and single motor based EVs in daily driving cycles, i.e. WLTC, FTP75, HWFET and JP1015, Fig.2(e-f) show their operation tracks and reveal the spare power and torque. Regarding Fig.2 (e), the different color dots show the required torque on wheel of typical driving cycles with specifications Table.A1. The solid blue line represents the required torque of wheel on a 30% grade at 30 km/h, which is usually taken as one of the criteria to determine the maximum torque requirement for motor. As we can see, a significant gap between these two requirements is presented. Additionally, the torque requirements of typical cycles reveal two clear trends, namely relatively high torque-low speed events and relatively high speed-low torque events, which cannot be catered by one single gear.

The tracking maps indicate that the ample spare torque/power is rarely used in daily driving, leading to most of the operation tracks are far from the high efficiency area. Although a relatively small gear ratio could offer a better motor efficiency by increasing the required torque of motor, the requirement of relatively big ratio for acceleration, climbing and high-speed overtaking exclude the possibility as the trade-off of single gear implementation in EV. To boost the low operational efficiency in daily driving for a powerful motor with high torque capability, reduce the maximum available torque could improve the motor operating area, in consequence, increase the average energy converting efficiency. In addition, the corresponding engaged gear should be adjusted to compensate the omitting motor torque.

III. POWERTRAIN SPECIFICATION DESIGN - DETERMINE THE SIZE OF TWO MOTORS AND CORRESPONDING GEAR RATIOS

The specification design of two propelling motors in proposed powertrain should be carried out carefully, since the proportion of power between two motors and the torque-speed characteristics may affect the economic and dynamic performance. Considering the specifications of two popular market products in Table.A2 [2], [19]., i.e. VW e-Golf and

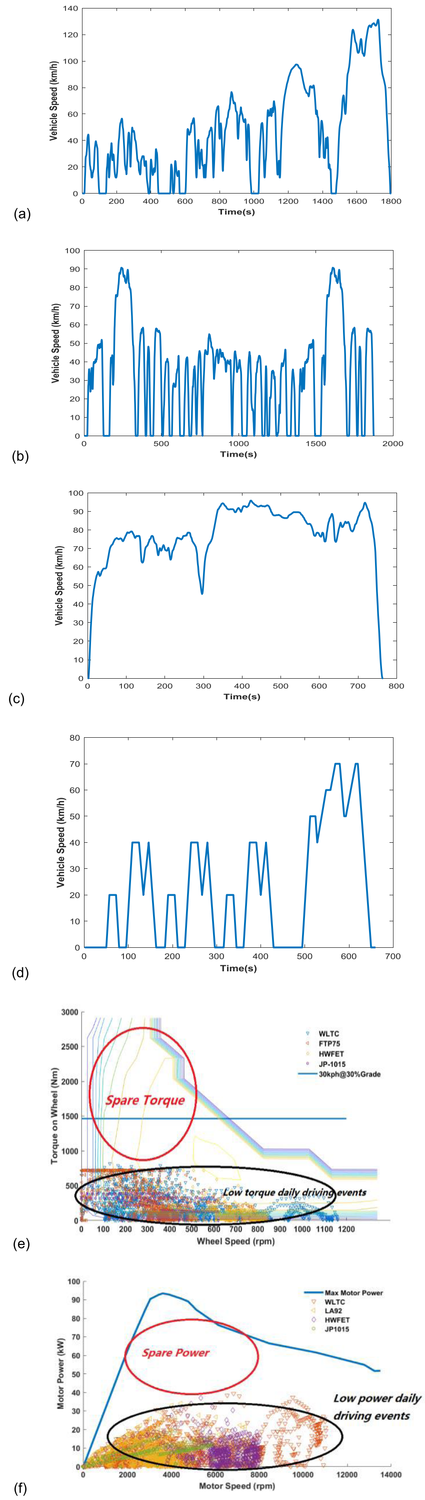


FIGURE 2. a) Worldwide harmonized Light Vehicles Test Cycles (WLTC) (b) Federal Test Procedure (FTP75) (c) Highway Fuel Economic Test (HWFET) (d) Japan 10-15 Test I Motor operating tracks (f) Classified Driving Events.

Tesla P85D, an EV powertrain with 300Nm-14000rpm motor and 9.7 single speed ratio is taken as the benchmark to test the proposed DMTS powertrains.

The maximum output torque of benchmark powertrain is:

$$T_{\max\_\sin gle} = 300Nm \times 9.7 = 2910Nm \quad (1)$$

The maximum speed 14000 rpm is kept unchanged in DMTS. The target 140km/h top speed achieved in the vehicle can then be used to determine the celling of possible gear ratios:

$$\gamma_{\text{speed}} \leq \left(\frac{2\pi r_t N_{\max\_14000rpm}}{60}\right) \div \left(\frac{V_{\max}}{3.6}\right) = 12.3 \quad (2)$$

Based on simulation results of typical driving cycles in Fig.2(e-f), the motor efficiency could be improved through increasing operating torque and reducing its speed. Therefore, a dual-motor system, combining two 'less' powerful motors, has the potential to improve motor efficiency by narrowing the efficiency range to cater the daily driving events, and provide at least the same maximum torque and power of single motor powertrain. One of the motors is responsible for the low torque/power driving events, such as the frequent starts & stops in city and low speed cruising. The other motor works as a supplementary power source for the high torque/power driving events, such as urgent acceleration and high-speed overtaking. These two motors will work together if the capability of single motor is insufficient in extreme conditions.

The specification design of two motors depends on the analyzing and summarizing of different combination candidates in various driving cycles, including a range of standard testing cycles like FTP75 (Federal Test Procedure), Japan 10-15, HWFET (Highway Fuel Economy Test) and WLTC (Worldwide harmonized Light Vehicles Test Cycles). Table.A3 summarizes the motor operating tracks with various torque ability and engaged gear ratios, i.e. 100 Nm, 150 Nm, 200 Nm and ratio 5, 6 and 7. According to the figures, relatively small motor torque abilities, like 100 Nm motor, show a greater potential to improve efficiency by more dots (operating track) falling in higher efficiency area, followed by 150 Nm and 200 Nm candidate motors in each engaged ratio. Regarding the engaged ratio for same motor torque, a bigger gear ratio tends to achieve higher motor operational efficiency. However, it is worth noting that part of the operating tracks falls out of the motor efficiency map, which indicates that the motor torque ability is insufficient to cover the requirement, resulting in the second motor involvement. Since the maximum required torque is shown as approximate 700 Nm in Fig.2, along with efficiency performance in Table.A3, the combination of $T_{EM1\_\max} = 100Nm$ motor torque and gear ratio $g_{EM1} = 7$ is selected as one of two power sources in DMTS, which is used to cover the minimum 700Nm wheel torque in typical cycles shown in Fig.2. To reduce the maximum torque of another motor in DMTS, in the meanwhile, meeting the constrain in (2), the gear ratio g_{EM2} is selected as 12. The torque profile constrains apply on the DMTS powertrain can be expressed as:

$$T_{\sin\text{gle}\_\max} \times g_{\sin gle} = T_{EM1\_\max} \times g_{EM1\_\max} + T_{EM2\_\max} \times g_{EM2\_\max} \quad (3)$$

Given $T_{\sin gle\_\max}$, $g_{\sin gle}$, $T_{EM1\_\max}$ and g_{EM1} have been selected, the $T_{EM2\_\max}$ can be achieved:

$$T_{EM2\_\max(DMTS1)} = (300 \times 9.7 - 100 \times 7) \div 12 = 184Nm \quad (4)$$

To demonstrate a comprehensive investigation on how torque distributions between motors affect the DMTS' economic performance, another DMTS with same ratios and balanced torque abilities in two motors is adopted as a control group:

$$\begin{cases} T_{EM1\_\max(DMTS2)} = 150N\text{m} \\ T_{EM2\_\max(DMTS2)} = 155N\text{m} \\ 150 \times 7 + 155 \times 12 = 100 \times 7 + 184 \times 12 = 2910Nm \end{cases} \quad (5)$$

Therefore, in this study, two DMTS powertrains will be investigated with specifications present in Table.1.

TABLE 1. Specifications of single motor powertrain, DMTS 1 and DMTS 2.

Description	Single Motor Single Speed	DMTS 1	DMTS 2
Motor 1	Max 300Nm Max 14000 rpm	Max 100Nm Max 14000 rpm	Max 150Nm Max 14000 rpm
Gear Ratio 1	9.7	7	7
Motor 2	N/A	Max 185Nm Max 14000 rpm	Max 155Nm Max 14000 rpm
Gear Ratio 2	N/A	12	12

Fig.3 presents the motors efficiency maps and available wheel torque of DMTS 1 and DMTS 2. As shown in the picture, DMTS 1 show a greater difference between two motors, in terms of torque, while two motors in DMTS 2 are more balance. Regarding the wheel available torque, both DMTS 1 and DMTS 2 provide similar maximum torques, compared to single motor powertrain, guaranteeing a similar dynamic performance. Overall, both DMTS powertrains can offer similar dynamic performance to single speed powertrain, while provide efficiency improvement potential by two designed motor characteristics.

IV. DUAL MOTOR POWERTRAIN DYNAMIC MODELLING AND CONTROL STRATEGY

A. SHIFTING STRATEGY

Compared to one motor multi-speed EV powertrain, the layout and control strategy of proposed DMTS are straightforward. The decision of gear shifting, namely the switching between two propelling motors, is only determined by motor efficiency, which can be realized via motor torque control alone. The simple strategy ensures that vehicle is always driven by the motor with higher working efficiency in each

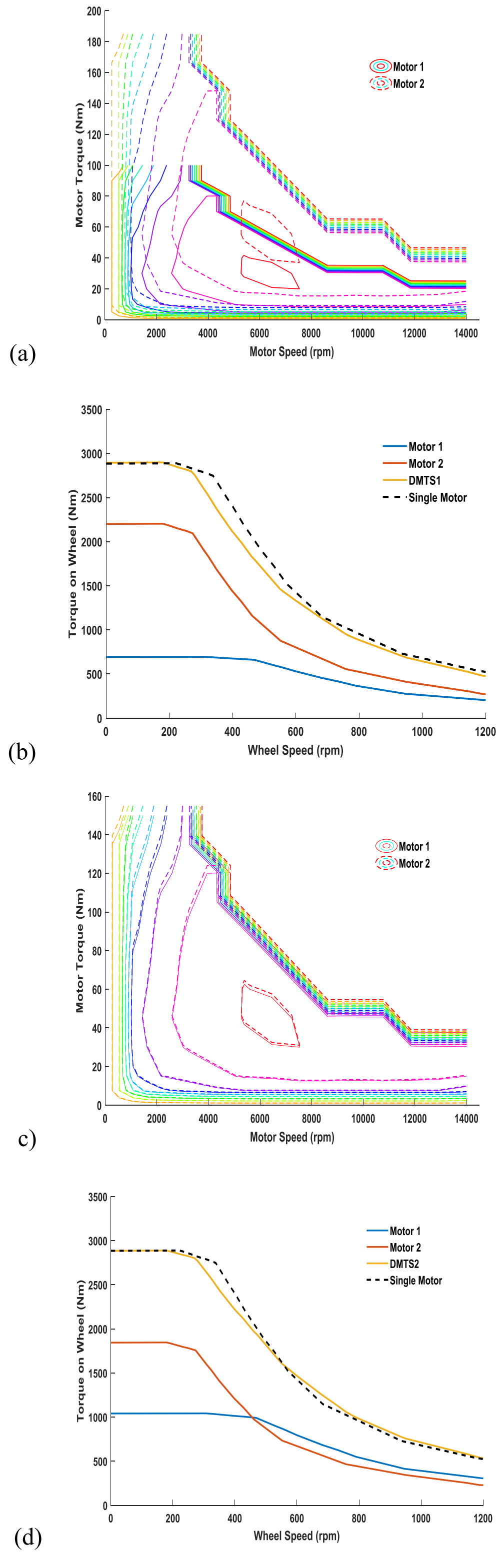


FIGURE 3. (a) DMTS1 motors efficiency maps (b) DMTS1 Max wheel torque (c) DMTS2 motors efficiency maps (d) DMTS2 Max wheel torque.

moment unless required torque cannot be covered by single motor. Fig.4 demonstrates the basic shifting decision flow.

The equations of motions and schematic diagram of DMTS are adopted in the following to achieve smooth torque transferring between two gears. In Fig.5, the gear pair permanently engaged with motor 1 consists of gear 1 and 3; gear pair 2 and 4 represents the gear reduction for motor 2; final gear, i.e. gear pair 5 and 6, drive the wheel.

J_{M1}, J_{M2} and J_{1-6} note the inertia of motor 1, motor 2 and gear 1-6; ω represents the rotating speed of each component; in consequence, $\dot{\omega}$ denotes their acceleration; i_1 is the ratio of gear 3 to gear 1; i_2 is the ratio of gear 4 to gear 2; i_{final} is the ratio of gear 6 to gear 5; T_{M1} and T_{M2} stand for the output torque of motor 1 and motor 2; T_1 and T_2 are the torque on motor output shafts; $T_{Layshaft}$ and T_{Final} denote the torque on layshaft and final shaft respectively.

According to Fig.5 the powertrain rigid model can be derived as follows:

$$(J_{M1}+J_1)\dot{\omega}_{M1} = T_{M1} - T_1 \tag{6}$$

$$(J_{M2}+J_2)\dot{\omega}_{M2} = T_{M2} - T_2 \tag{7}$$

$$(J_3+J_4+J_5)\dot{\omega}_5 = i_1 T_1 + i_2 T_2 - T_{Layshaft} \tag{8}$$

$$J_6\dot{\omega}_6 = i_f T_{Layshaft} - T_{Final} \tag{9}$$

$$\omega_5 = \frac{\omega_{M1}}{i_1} = \frac{\omega_{M2}}{i_2} \tag{10}$$

Substitute (6), (7) and (8) to (9):

$$\begin{aligned} J_E\dot{\omega}_6 = {} & i_1 i_f T_{M1} + i_2 i_f T_{M2} - i_1 i_f (J_{M1}+J_1)\dot{\omega}_1 \\ & - i_2 i_f (J_{M2}+J_2)\dot{\omega}_2 - i_f (J_3+J_4+J_5) \\ & \times \dot{\omega}_5 - T_{Final} \end{aligned} \tag{11}$$

where

$$J_E = (J_{M1}+J_1)i_1^2 i_f^2 + (J_{M2}+J_2)i_2^2 i_f^2 + (J_3+J_4+J_4)i_f^2 + J_6 \tag{12}$$

Aiming at smooth torque transfer without 'torque hole' between two gears, the overall output torque of DMTS on final shaft should be well kept unvaried during shifting process, in other words, $\dot{T}_{Final}$ should be zero. In consequence, ideally, there would be no change of final shaft speed $\dot{\omega}_6$

$$\begin{aligned} 0 = {} & i_1 i_f T_{M1} + i_2 i_f T_{M2} - i_1 i_f (J_{M1}+J_1)\dot{\omega}_1 \\ & - i_2 i_f (J_{M2}+J_2)\dot{\omega}_2 - i_f (J_3+J_4+J_5)\dot{\omega}_5 - T_{Final} \end{aligned} \tag{13}$$

As the shift process is supposed to be finalized in a very short period, normally less than 0.5 second, the vehicle longitudinal speed can be assumed as constant during this short period. Thus, the resistive torque of vehicle on final shaft $\dot{T}_{Final}$ can be regarded as time-invariant, in other words, $\dot{T}_{Final} = 0$. Then, substitute (10) to (13):

$$\begin{aligned} 0 = {} & i_1 i_f T_{M1} + i_2 i_f T_{M2} - i_1 i_f (J_{M1}+J_1)\dot{\omega}_1 \\ & - i_2 i_f (J_{M2}+J_2)\dot{\omega}_2 - \frac{i_f}{i_1}(J_3+J_4+J_5)\dot{\omega}_{M1} \end{aligned} \tag{14}$$

According to (14), when the torque transmitted from motor 1 to motor 2:

$$\begin{aligned} T_{M2} = {} & \int_{t_0}^{t_1} -\frac{i_1}{i_2}\dot{T}_{M1} + \frac{1}{i_1 i_2}(J_3+J_4+J_5) \\ & + \frac{i_1}{i_2}\dot{\omega}_{M1}(J_{M1}+J_1) + \frac{i_2}{i_1}\dot{\omega}_{M1}(J_{M2}+J_2)\,dt \end{aligned} \tag{15}$$

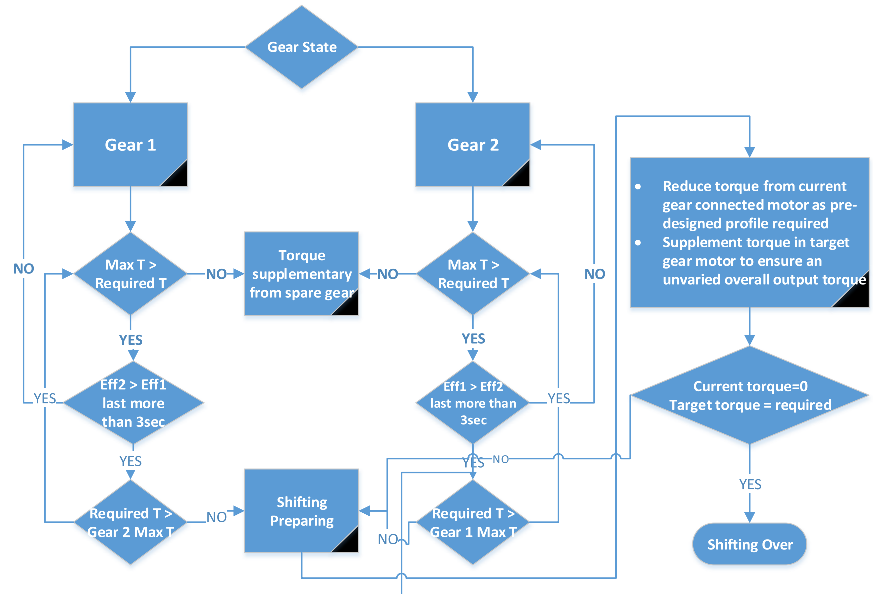


FIGURE 4. Dual motor direct drive shifting strategy.

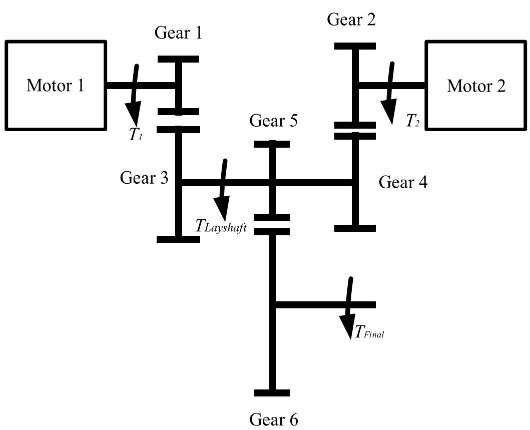


FIGURE 5. The powertrain schematic diagram.

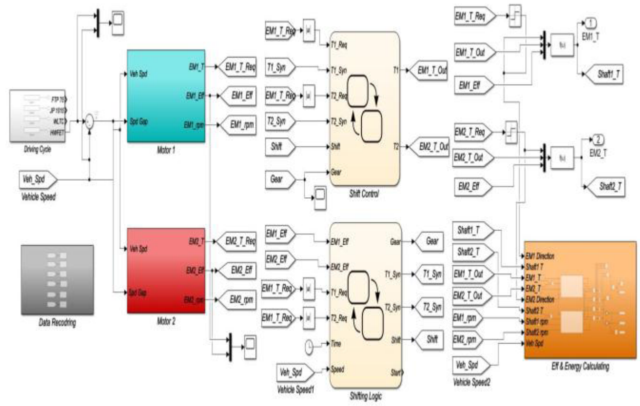


FIGURE 6. ECU Simulink model.

when the torque transmitted from motor 2 to motor 1:

$$T_{M1} = \int_{t_0}^{t_1} -\frac{i_2}{i_1}\dot{T}_{M2} + \frac{1}{i_1 i_2}(J_3 + J_4 + J_5) + \frac{i_1}{i_2}\dot{\omega}_{M2}(J_{M1} + J_1) + \frac{i_2}{i_1}\dot{\omega}_{M2}(J_{M2} + J_2)\, dt \quad (16)$$

Aiming at a smooth torque transfer, cubic polynomial is adopted in torque increasing in engaging motor and torque decreasing in disengaging motor as a reference trajectory.

Torque decreasing trajectory:

$$T_M = \frac{2T_{M0}}{(t_1 - t_0)^3}(t - t_0)^3 - \frac{3T_{M0}}{(t_1 - t_0)^2}(t - t_0)^2 + T_{M0} \quad (17)$$

Torque increasing trajectory:

$$T_M = \frac{3T_{M0}}{(t_1 - t_0)^2}(t - t_0)^2 - \frac{2T_{M0}}{(t_1 - t_0)^3}(t - t_0)^3 \quad (18)$$

Herein, T_M is the initial value of motor torque; t_0 and t_1 are the start time and end time of shifting process.

B. DYNAMIC MODELLING

Based on aforementioned shifting equation of motions and basic vehicle dynamics, a forward Matlab/Simulink® model is built by using both mathematic and Simscape® block to simulate the dynamic performance of proposed powertrain.

The shifting decision and control of two driving motors are implemented in Electronic Control Unit (ECU) part, which is demonstrated in Fig.6 (b). Two independent PID controller in motor blocks decide the required torque of each motor when it drives the vehicle solely. According to each motor's efficiency map, when the torque ability is sufficient, the Stateflow® based shifting logic will only enable one motor, which works in a higher efficiency at each moment. The demand torque of disabled motor is taken as the target torque reference when motor shifting is required. According to the signal 'Shift', an output signal of shifting logic block, the shift control block will be enabled to determine whether output torque follows the dynamic varying signals 'T1_Syn' and 'T2_Syn' during shifting, or the regular motor torque 'T1_Req' and 'T2_Req'. 'T1_Syn' and 'T2_Syn' are the shifting reference torque according to (17) and (18). The energy and efficiency performance of proposed powertrain will be calculated in orange 'Eff & Energy Calculating' block and stored by grey 'Data Recording' block respectively.

V. RESULTS AND ANALYSIS

A. ENERGY EFFICIENCY

Based on above strategies and model, the economic and dynamic performance of proposed DMTS are shown in this section. To provide a better readership, overall motor efficiency and energy consuming in each typical cycle are summarized at the beginning as a guide for following analyzing.

TABLE 2. Overall motor efficiency of DMTS1 and DMTS2 in typical cycles (bi-direction, incl. driving and regenerative braking).

	Single Motor Eff	DMTS1 Motor Eff	Improve ment	DMTS2 Motor Eff	Improve ment
WLTC	84.6%	90.9%	**7.5%**	90.6%	**7.5%**
FTP-75	83.3%	88.0%	**5.6%**	88.0%	**5.6%**
HWFET	79.2%	89.8%	**13.4%**	88.8%	**12.1%**
JP10-15	83.3%	89.1%	**7.0%**	89.5%	**7.4%**

Comparing the columns and rows in Table.2, following conclusions can be driven:

(1) Both DMTS1 and DMTS2 achieve significant improvement in overall motor operational efficiency, compared to the performance of single motor powertrain.

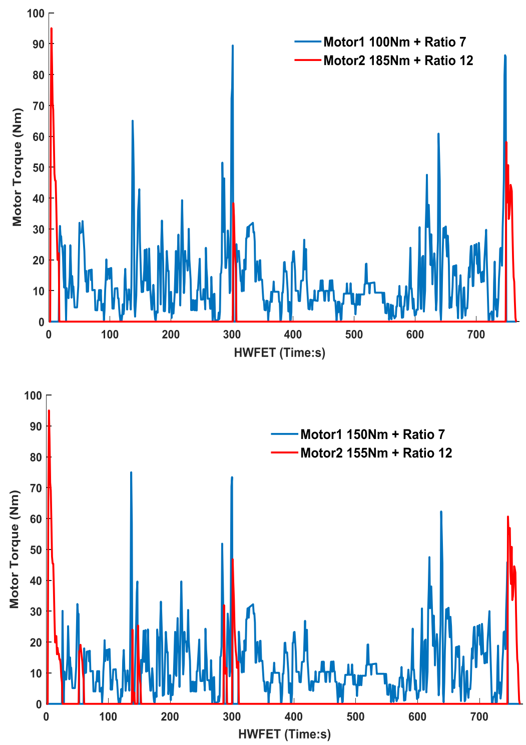
(2) More than 12% motor efficiency improvement can be expected in highway driving patterns for DMTS1 and DMTS2.

(3) 5%-8% efficiency improvement could be available for other city driving cycles, such as WLTC, FTP-75 and JP 10-15.

(4) Comparing DMTS1 to DMTS2, except HWFET, they show similar performance in terms of overall motor efficiency and energy consuming. The reason of DMTS1 superior DMTS2 in HWFET is the relatively high-speed low-torque events provide more opportunities for the 100Nm motor in DMTS1 to operate in a higher efficiency zone, while both motors in DMTS2 have a relatively large torque capacity. However, in city cycles, the advantage of smaller torque capacity could be reduced to some extent by the relatively large torque requirement, especially when the torque ability of smaller motor in DMTS1 is insufficient, which is shown in Fig.11(a).

TABLE 3. Energy consuming per cycle of DMTS1 and DMTS2 (excl. regenerative braking).

kWh	Single Motor Energy Cost	DMTS1	Improve ment	DMTS2	Improve ment
WLTC	2.77	2.64	**4.7%**	2.66	**4.0%**
FTP-75	1.96	1.92	**2.0%**	1.93	**1.6%**
HWFET	1.34	1.20	**10.5%**	1.21	**9.7%**
JP10-15	0.44	0.42	**4.6%**	0.42	**4.6%**



a)

b)

FIGURE 7. Torque comparison of (a) DMTS1 and (b) DMTS2 in HWFET.

Regarding Table.3, both DMTS1 and DMTS2 have proven that they have the ability to reduce the energy consumption regardless the driving cycles. However, it does not perform as good as motor efficiency improvement in terms of percentage achievement due to the energy recovering (regenerative braking) are excluded from this summary, whose performance is highly depends on strategies [20] but could also be improved by dual-motor powertrain.

The above results and conclusions can be verified from the viewpoint of motor torque and efficiency tracks in following figures. As shown in Fig.7, the comparison of motor

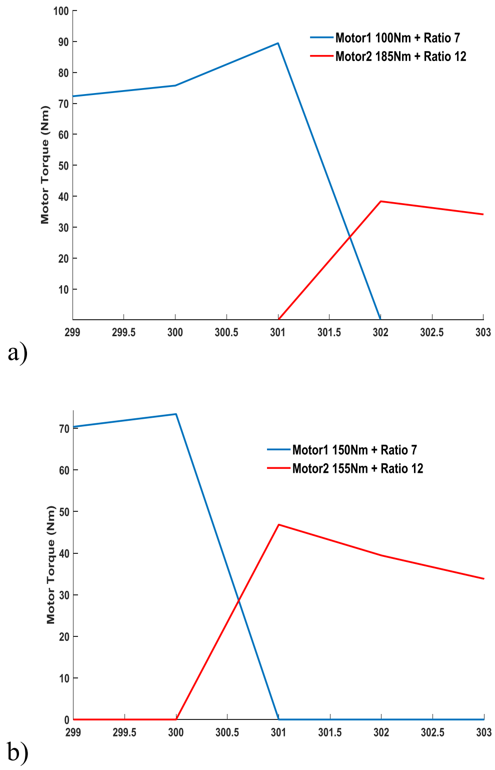


FIGURE 8. Torque varying in partial period of HWFET (a) DMTS1 and (b) DMTS2.

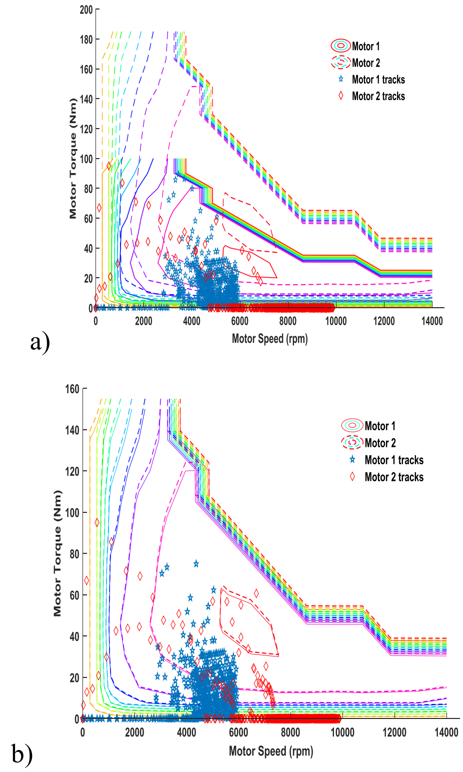


FIGURE 9. Operating efficiency comparison in map in HWFET (a) DMTS1 and (b) DMTS2.

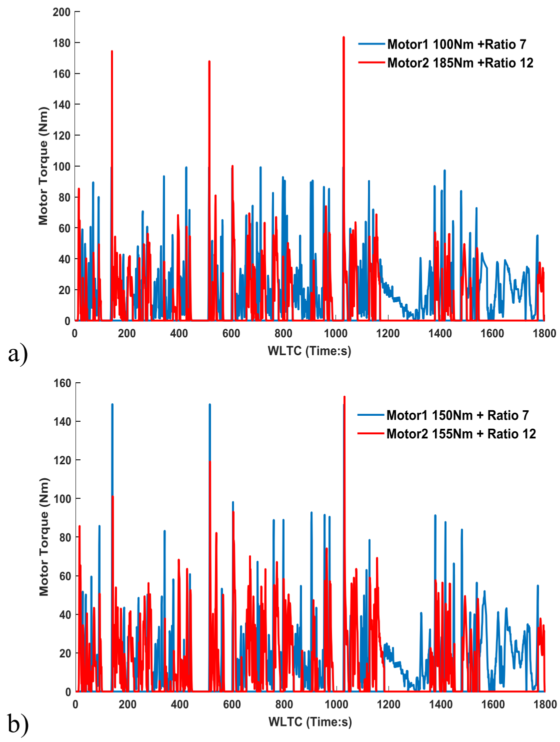


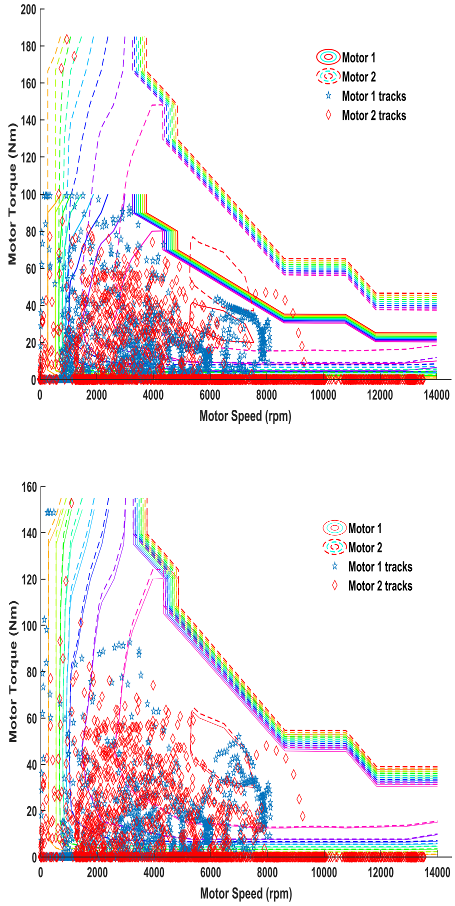
FIGURE 10. Torque comparison of (a) DMTS1 and (b) DMTS2 in WLTC.

torque varying in HWFET, it is easy for DMTS1 to keep the engaged motor 1 runs at higher efficiency area due to the relatively small torque range, while DMTS2 has to switch driving motors more frequently to pursue higher efficiency. In other words, the relatively low torque requirement, namely less than 30Nm torque events, is more suit for the small torque capacity motor, just as the 100Nm motor in DMTS1.

It is also worth noting that the overall output torque (Motor 1 + Motor 2) performs different at same moment in Fig.7(a) and (b). For instance, the output of motor 1 in DMTS1 is around 90Nm at 300s, however, it is only roughly 70Nm in DMTS2. Actually, the driving motor switching is responsible for the torque difference. It can be seen in Fig.8, which is an amplified partial of Fig.7, the output torque of DMTS1 is around $75Nm \times 7 = 525Nm$ at 300s in Fig.8(a), which is similar to the torque of DMTS2 in Fig.8(b) at the same time; at 301s, the output torque of DMTS1 is still from motor 1 alone, but rise to about $85Nm \times 7 = 595Nm$, while the output torque of DMTS2 is from motor 2 after switching, rising to $49Nm \times 12 = 588Nm$. They show a good agreement considering the different system efficiency.

Fig.9 presents the motor operating tracks in efficiency map for DMTS1 and DMTS2 in HWFET. The results also verify the conclusion from Table.2 and 3 that DMTS1 superior DMTS2 in HWFET since there are more tracks of 100Nm motor 1 in DMTS1 falls in the higher efficient areas.

Given the similar overall efficiency and energy consuming results of city cycles in Table.2 and 3, only WLTC is selected as a representative of city cycles to limit this article in a reasonable length.



a)

b)

FIGURE 11. Operating efficiency comparison in map in WLTC (a) DMTS1 and (b) DMTS2.

Unlike the highway cycles, DMTS1 and DMTS2 do not show significant difference in WLTC. The relatively high torque requirement in acceleration provides more chance for the two motors of DMTS2 to operate in high efficiency zone, which is missing in highway cycles.

B. TORQUE TRANFER DYNAMICS DURING DRIVING MOTOR SWITCHING

The driving motor switching can be taken as gear shifting in this study since each motor is permanently engaged to a fixed ratio gear. Thanks to the outstanding performance of electric machine (motor), i.e. fast and accurate response to demanding torque and speed, smooth and quick torque transferring between two motors can be achieved by controlling motor solely. The upper picture of Fig.12 illustrates the torque varying and transfer during shifting. As we can see, the overall output torque is well controlled and kept almost unvaried during the process. The middle picture of Fig.12 indicates the time period by the red line equaling 1, which is about 0.5s. The indicating gear number does not change until the shifting process completed.

Vehicle jerk is the varying rate of vehicle longitudinal acceleration in unit of m/s^3. It is used to evaluate the quality

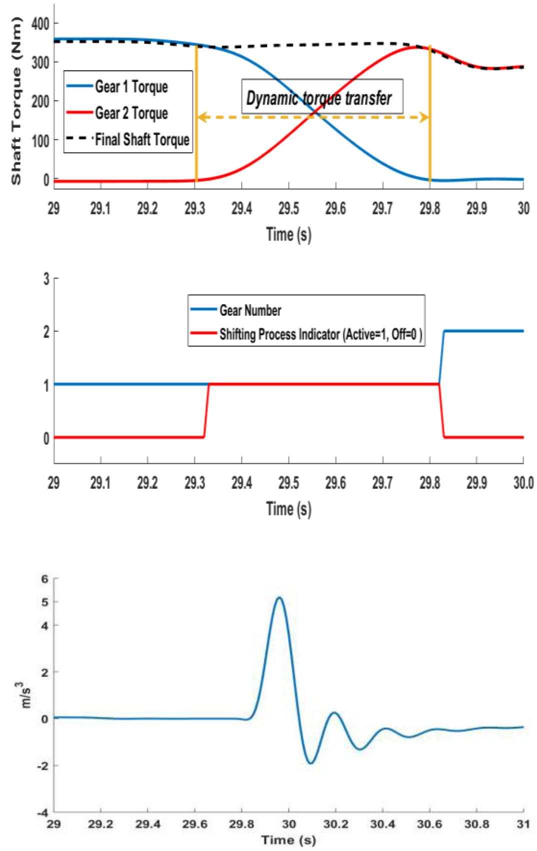


FIGURE 12. Torque hand over between motor 1 and motor 2 in WLTC.

TABLE 4. Summary of economic benefit to DMTS.

	Single Motor	DMTS1	DMTS2
Fuel economic* ($/100km)	2.83	2.59	2.61
Electricity cost per annual*(18240km) (USD)	637	582	588
Driving range (35kWh, hybrid cycle)	370	406	402
Required battery capacity (400km, kWh)	37.76	34.48	34.80
Battery cost ($USD) ($300USD/kWh[28])	11328	10344	10440

***The figure is based on 0.3 USD/kWh electricity fee** [29].

of shifting. The criterion for preferred maximum vehicle jerk during traditional gear shifting varies between regions. For example, Germany recommends [J] $< 10\ m/s^3$ [21], whilst in China it is [J] $< 17\ m/s^3$. Since the proposed powertrain in this study does not include any clutch or synchronizer, the vehicle jerk caused by torque transfer between motors is expected significantly lower than the above criteria.

TABLE 5. Vehicle specification.

Description	Parameter	Value	Unit
Gross Weight	M	1635	Kg
Extra Weight of 2nd Motor	ΔM	40	Kg
Vehicle Front Areas	A	2.16	m^2
Aero-drag Coefficient	Cd	0.25	
Tyre Radius	r	0.325	m
Tyre Rolling Coefficient	Ct	0.016	
Air Density	ρ	1.18	kg/m^3
Battery Capacity	C	35	kWh
Single Gear Ratio		9.7	
Gear Ratio 1 (Incl. Final gear)		7	
Gear Ratio 2 (Incl. Final gear)		12	
Gear Friction Eff [30]	e_{Gears}	99%	
Differential Eff [30]	e_{Diff}	95%	
DMTS 1 Motor 1 Max Torque	$T1_{max}$	100(DMTS1) / 150(DMTS2)	Nm
DMTS 1 Motor 2 Max Torque	$T2_{max}$	185(DMTS1) / 155(DMTS2)	Nm
DMTS 1 Motor 1 Max Speed	$N1_{max}$	14000	Revolution per minute
DMTS 1 Motor 2 Max Speed	$N2_{max}$	14000	Revolution per minute
Single Motor Max Torque	T_{max}	300	Nm
Single Motor Max Speed	N_{max}	14000	Revolution per minute

The results are not far away from the expectations shown in the bottom picture of Fig.12. It shows the maximum vehicle jerk during driving motor shifting is around 5 m/s^3, which is a very satisfied result.

C. POTENTIAL DRIVING RANGE EXTENSION AND COST SAVING

To test the impact of proposed powertrain in daily driving for ordinary commuters, a hybrid driving pattern is adopted to investigate the potential driving range extension and corresponding cost saving for BEV. The adopted hybrid driving cycle includes 57% highway driving and 43% city driving respectively, which is suggested in [22]. Considering original adopted Federal testing procedures-75 was created many year ago, WLTC replaces FTP75 to represent the city driving patterns to make the test reflect the reality. The fuel (electricity) economy is measured by (19) in kilowatts hour per 100 km (kpk):

TABLE 6. Specifications of VW e-GOLF [1] and TESLA P85D [2].

Description	VW e-Golf 2017	Tesla P85D	Unit
Gross Weight	1556	2200	Kg
Motor Type	Permanent-magnet synchronous AC motor	AC induction motor	
Max Motor Torque	290	690	Nm
Max Motor Speed	12000	16000	rpm
Gear Ratio (Incl. Final ratio)	9.74	9.73	
Battery Capacity	35.8	85	kWh

$$Hybrid_{kpk} = \frac{1}{0.43/WLTC_{kpk} + 0.57/HWFET_{kpk}} \tag{19}$$

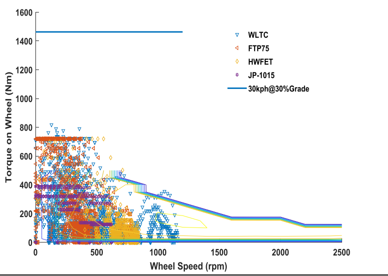
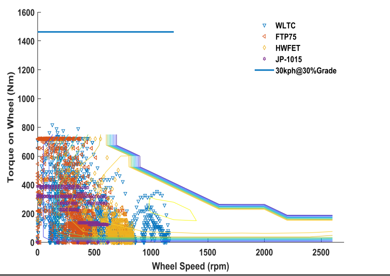
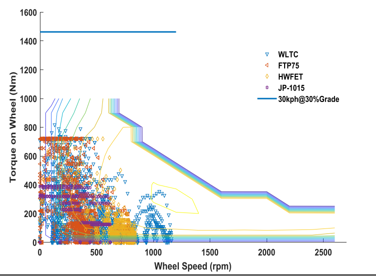
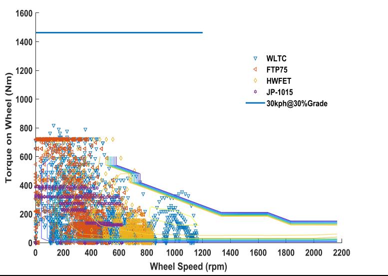
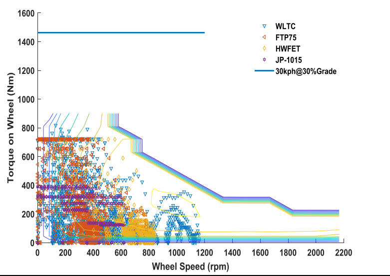
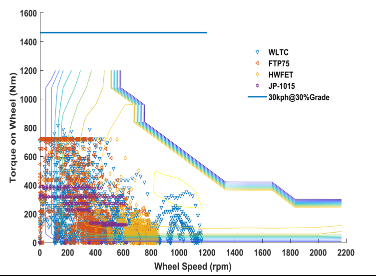
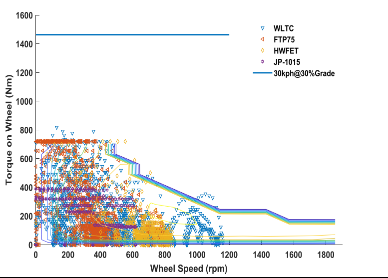
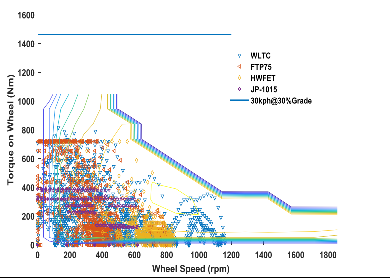
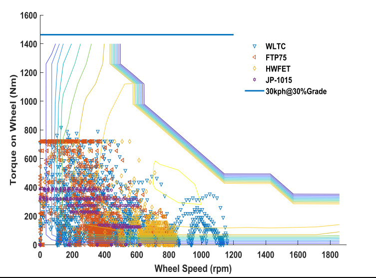
Given the range of WLTC and HWFET are 23.194 km [23] and 16.45 km [24], the fuel economic can be achieved in terms of kWh/100km through the results in Table.3:

$$\begin{cases} WLTC_{Single-kpk} = 2.77 \times (100 \div 23.194) \\ \quad = 11.94kWh/100km \\ WLTC_{DMTP1-kpk} = 2.64 \times (100 \div 23.194) \\ \quad = 11.38kWh/100km \\ WLTC_{DMTP2-kpk} = 2.66 \times (100 \div 23.194) \\ \quad = 11.47kWh/100km \end{cases} \tag{20}$$

$$\begin{cases} HWFET_{Single-kpk} = 1.34 \times (100 \div 16.45) \\ \quad = 8.15kWh/100km \\ HWFET_{DMTP1-kpk} = 1.20 \times (100 \div 16.45) \\ \quad = 7.29kWh/100km \\ HWFET_{DMTP2-kpk} = 1.21 \times (100 \div 16.45) \\ \quad = 7.36kWh/100km \end{cases} \tag{21}$$

$$\begin{cases} Hybrid_{Single-kpk} = 1 \div (0.43/11.94 + 0.57/8.15) \\ \quad = 9.44kWh/100km \\ Hybrid_{DMTP1-kpk} = 1 \div (0.43/11.38 + 0.57/7.29) \\ \quad = 8.62kWh/100km \\ Hybrid_{DMTP2-kpk} = 1 \div (0.43/11.47 + 0.57/7.36) \\ \quad = 8.70kWh/100km \end{cases} \tag{22}$$

TABLE 7. Motor operating tracks with candidate motor and gear specifications.

	Motor Maximum Torque 100 Nm	Motor Maximum Torque 150 Nm	Motor Maximum Torque 200 Nm
Ratio 5			
Ratio 6			
Ratio 7			

In terms of driving range extension, for a 35kWh battery, the expected range per charge in hybrid cycle are:

$$\begin{cases} Range_{\sin gle} = 100 \times 35 \div 9.44 = 370km \\ Range_{DMTP1} = 100 \times 35 \div 8.62 = 406km \\ Range_{DMTP2} = 100 \times 35 \div 8.70 = 402km \end{cases} \tag{23}$$

If the target range is set as 400 km per charge, the required battery capacity are as follows:

$$\begin{cases} C_{\sin gle} = 400 \div 100 \times 9.44 = 37.76kWh \\ C_{DMTP1} = 400 \div 100 \times 8.62 = 34.48kWh \\ C_{DMTP2} = 400 \div 100 \times 8.70 = 34.80kWh \end{cases} \tag{24}$$

In terms of electricity cost saving in daily driving, some survey [25] shows the annual vehicle travel range is 18,240 km, which can be verified by a typical 50 km daily range [26] times 365 days a year. Considering the efficiency of charger is 81% at Level 2 standard charging voltage, as a result of same 90% efficiency for both plug-in charger and lithium–ion battery charge/discharge [27]. The total electricity cost per year is summarized in Table.4. The Li-ion battery cost is based on an industry report to date which estimating $300(USD)/kWh for major battery manufacturers [28].

As we can see from Table.4, about $0.23/100km and $60 per annual can be saved in electricity fee through proposed DMTS. If EV equipped with a 35kWh battery, extra 40 km range is possible to be used through higher motor efficiency, which offer more than 10% longer driving range per charge. If the driving range per charge is set as a target 400 km, 3.28 kWh and 2.96 kWh battery capacity could be saved respectively for DMTS1 and DMTS2, which will save manufacturer about $1000 per car. Ultimately, both manufacturers and customers can benefit from this proposed powertrain.

It is worth noting than the above figures do not include the loss in other mechanical components, like gear, axle, differential and auxiliary equipment, resulting in a better than real energy performance, compared to market product, e.g. e-Golf. However, since the negative factors affect both the traditional single motor powertrain and DMTS, these figures still provide a good sample and comparison to show the advantage of DMTS to single motor powertrain.

VI. CONCLUSIONS

In this study, a dual-motor two-speed direct drive EV powertrain is proposed to boost motor efficiency, ultimately, saving limited battery energy and manufacturing cost.

Initially, considering the massive spare torque and power is wasted in daily driving events and low overall efficiency, the proposed powertrain configuration split one driving motor to two, targeting at different driving patterns. Motor capacity and permanent engaged gear ratio are designed carefully catering the vehicle dynamic and economic performance, followed by a simple and efficiency-oriented driving motor switching strategy. To ensure a smooth torque transfer during gear shifting, the designed shifting control strategy is then tested in a dynamic Simulink® model. Simulation results show significant motor efficiency improvement can be expected with a satisfied vehicle jerk during shifting, i.e. less than $5m/s^3$. At last, the proposed dual motor EV powertrain shows the potential benefit to costumers and manufacturers in terms of cost saving in daily driving and battery manufacturing.

Overall, the dual-motor two-speed direct drive EV powertrain demonstrates its potential to superior the traditional single motor fixed speed powertrain from the viewpoint of economic benefit, in the meanwhile, without any compromising in simplicity of structure or control strategy. It could be a feasible option for manufacturers to replace the current widely used single motor powertrain in near future.

VII. FUTURE WORK

Although this paper shows the advantage of dual motor drive BEV to single motor drive, the possibility of boosting this benefit through reallocating the maximum motor torque and gear ratio needs to be discovered. The future researching work should focus on efficiency further improvement via specifications design of dual motor and two gear pairs, and the power allocation between two motors with fixed overall output power.

APPENDIX

See Tables 5–7.

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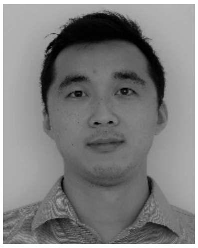

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