Hybrid powertrain efficiency improvement by using electromagnetically controlled double-clutch transmission

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Abstract: This paper proposes a new type of hybrid powertrain with a specially designed double-clutch transmission. Compared with the existing hybrid powertrains, the proposed solution with double-clutch transmission, based on innovative double-clutch and mechanical gear sets, has the potential to achieve higher levels of performance efficiency at a lower production cost. The impact factors of system efficiency are analyzed on a dynamic non-linear simulation model. The control strategies of powertrain and the double-clutch transmission gear ratios are studied by simulation results. Additionally, the double-clutch transmission can increase the efficiency of regenerative braking.

Keywords: hybrid powertrain; efficiency; double-clutch transmission; simulation.


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Pawel Krawczyk graduated Warsaw University of Technology, Faculty of Automotive and Mechanical Engineering and obtained Master degree in 2010. Since then he is PhD candidate and part of research team of Professor Antoni Szumanowski in Department of Multisource Propulsion Systems. His field of consideration are electric and hybrid-electric drives, especially modelling, simulation and testing of drives and components.

1 Introduction

Based on continuous research on planetary transmission since earlier than 1997, the authors think that the hybrid powertrain could be designed with higher efficiency and lower cost. The first author patented the updated version of the hybrid powertrain equipped with planetary two degrees of freedom planetary transmission in 2011 (PL 209462 B1). This solution was long time tested by simulation and laboratory experiments. Of course, before preparing this patent many comparison analyses relating to existing hybrid powertrains were done (www2.toyota.co.jp/en/news/13/04/0417.html; www.kpmg.com/global/en/industry/automotive; Toyota Motor Corporation, 2003). The results persuaded authors to publish this paper. The authors designed and developed the Compact Hybrid Planetary Transmission Drive (CHPTD) as a solution to achieve this target (see Figure 1).

Figure 1  The configuration of the compact hybrid planetary transmission drive (CHPTD)

One of the main features of CHPTD is that it uses only one electric motor. With additional clutch/brake systems on shafts of power sources, the hybrid system can operate in different driving modes. Furthermore, the pure engine drive mode is an advantage of CHPTD to improve performance in highway driving.
The common method to improve the efficiency of an electric motor and ICE is to adjust their operating points for different working conditions, according to the efficiency map and fuel consumption map respectively (Szumanowski et al., 2005; Wirasingha and Emadi, 2011; Salmasi, 2007). For this reason, gearboxes usually apply to powertrains to enlarge the working range and to improve the efficiency of the motor and ICE. Since the speed control of hybrid powertrain is cooperating with the drive operational modes, only automatic transmissions such as AT, AMT and CVT are suitable for HEVs. Applying these transmissions may encounter the problem of either cost or efficiency.

To improve the efficiency of CHPTD, this paper presents a new solution of hybrid powertrain with double-clutch transmission, which is demonstrated in following section. This paper also presents how to optimise the gear ratio and the control method of double-clutch transmission by dynamic non-linear simulation.

2 Double-clutch transmission in hybrid powertrain

The double-clutch transmission is a gearbox integrated with the innovative zero steady-states electrical energy consumption clutch (Sumanowski et al., 2010; Wu, 2009; Li and Liu, 1990; Zhao et al., 2008). A possible configuration of hybrid powertrain with double-clutch transmission is presented in Figure 2. Based on the same structure, this double-clutch can be modified to clutch-brake or brake which could be used for other applications in hybrid powertrain (see Figure 2). According to the mechanical structure of such solution, the efficiency of double-clutch transmission should be as high as conventional manual transmission. The 4-speed double-clutch transmission is composed of two double-clutches and several sets of gears. The performance of this hybrid powertrain is tightly connected with design, adjustment and the control of clutches which are used in the hybrid powertrain.

In order to decrease the cost of such transmission, the proposed design used existing components instead of additional expensive planetary gear. By equipping the 4-speed double-clutch transmission, it ensures that the hybrid powertrain can achieve high efficiency, which also means the ICE and electric motor could be downsized, with smaller electric motor current, the copper loss is also decreased.

According to Figure 2, the clutch systems in the hybrid powertrains control not only the drive operational modes, but also gear shifting. It means the clutch systems applied in the hybrid powertrains can influence the performance of the whole powertrain.

However, the existing electromagnetic clutches or conventional friction clutches cannot fulfil the requirements of hybrid powertrain as below.

- adapted for the automatic control of powertrain (transmission)
- the clutch plate friction should be controlled dynamically
- as low as possible energy consumption
- low production cost.

To fulfil the requirements of hybrid powertrain, an innovative zero steady-states electrical energy consumption clutch system is designed (see Figure 3). The double-clutch system mentioned here is originally invented by one of the authors. A simulation model of this double-clutch system is developed in Simulink/Matlab environment. And laboratory tests
with similar system (electromagnetically actuated double-clutch system) in hybrid powertrain were made to prove the reliability (Sumanowski et al., 2013; Liu, 2014). This kind of clutch is different from an electromagnetic clutch in that it does not consume additional electrical energy to keep the engagement state. The mechanism with a symmetrical dual-diaphragm spring, which has two steady-states, ensures ‘zero energy consumption’ of the clutch system during full engagement. With a small modification, this system can be applied as clutch-brake and brake which give possibility of different hybrid powertrain operational modes - pure electric, pure engine and hybrid.

**Figure 2** CHPTD with 4-speed double-clutch transmission

**Figure 3** The configuration of the innovative zero steady-states electrical energy consumption clutch-brake system (Patented solution, patent number: PL.211409)
Figure 4 presents the configurations of double-clutch with one input shaft and two output shafts. This double-clutch system should be low cost for most components are available from the existing dry friction clutch.

Figure 4  Assembly of zero steady-states electrical energy consumption clutch-brake system for double-clutch application (see online version for colours)

The double-clutch system is also proving to be suitable for gear shifting by simulation and laboratory test (Sumanowski et al., 2013; Liu, 2014). The operation time constant of such double-clutch could be limited to 0.3–0.5 s with friction slip control. The control method of this clutch system is depicted in the next section.

3 Gear shifting control of double-clutch transmission

The control of double-clutch, which means the torque capacity of the clutch during engaging, influences the gear shifting performance of the powertrain.

The behaviour of clutch unit, which means the slipping between clutch plates during engaging, depends on its instant torque capacity. Figure 5 shows the controlled current shapes in the clutch actuator electromagnet coil during clutch engaging. The clutch actuation time, maximum current and current shapes are control variables which influence the instant torque capacity of the clutch.

\[
T_{cl} = f(I_{cl,coil}) \\
I_{cl, coil} = \begin{cases} 
I_{act}I_{max}f_n(t) & (n = 1, 2, 3, 4, 5), \text{ when } 0 \leq t \leq t_{act} \\
I_{max}, \text{ when } t > t_{act}
\end{cases}
\]

where

- \( T_{cl} \): the torque capacity of clutch
- \( I_{cl, coil} \): the current in electromagnet coil of clutch
- \( t_{act} \): the actuation clutch actuation time
- \( I_{max} \): the maximum current in electromagnet coil of clutch
- \( f_n(t) \): the function of current in electromagnet coil of clutch.
When the clutch plate in the engaging position, the reaction force of dual-diaphragm spring is a fixed value. Thus, the total axial force applied on clutch plate is as below.

\[ F_u = F_{\text{spring}} - F_{\text{em}} \]  

(2)

where

- \( F_u \): the axial force applied on friction plates
- \( F_{\text{spring}} \): the axial reaction force of dual-diaphragm spring
- \( F_{\text{em}} \): the axial force in electromagnetic actuator.

The input and output shafts of clutch are connected by the torque transferred to the clutch plate surface.

\[
\begin{align*}
J_{\text{in}} \frac{d\omega_{\text{in}}}{dt} &= T_{\text{in}} - T_{\text{cl}} - \omega_{\text{in}} b_{\text{in}} \\
J_{\text{out}} \frac{d\omega_{\text{out}}}{dt} &= T_{\text{cl}} - T_{\text{out}}
\end{align*}
\]

(3)

where

- \( J_{\text{in}}, J_{\text{out}} \): the equivalent moment of inertia reduced on input and output shafts
- \( T_{\text{in}}, T_{\text{out}} \): the input torque and resistant torque on the output shaft
- \( T_{\text{cl}} \): torque transmitted through the clutch friction plates
- \( \omega_{\text{in}}, \omega_{\text{out}} \): the angular velocities of input shaft and output shaft
- \( b_{\text{in}} [\text{kg m/s}^2] \): the torque-speed coefficient of input shaft referring to propulsion torque.

The control methods of the current shapes in clutch actuator electromagnet coils are studied to achieve better gear shifting performance by adjusting parameters to cooperate with time sequence of double-clutch system and other components in hybrid powertrain (see Figure 6).
In case of hybrid powertrain equipped with two degrees of freedom planetary gear (Figure 1), it is impossible to use the vehicle inertia. Because vehicle starts only by the electric motor, engine clutch is disengaged and its brake is switched on. When the vehicle's speed exceeds a proper value, the hybrid operation takes place. Then an engine start is possible by using vehicle kinetic energy and producing the braking torque on the crown wheel of the planetary gear. It means reducing two degrees of freedom to one degree. At the same time, the electric motor is working as a generator for a short moment, the braking torque on the crown wheel appears. And then the ICE start is possible by engaging the engine clutch. This method is applied in the proposed hybrid powertrain.

The output torque of the engine is not controllable until it starts. The engine gives only resistance torque when its speed changes from zero up to real starting speed during clutch engaging. The speed synchronisation of two clutch plates is indicated in the second subplot in Figure 6.

The double-clutch system, which contains twin clutches within a single housing utilised for gear shifting, is electronically controlled. They are not independent components, but an integrated set of components – two clutches, actuator, dual-diaphragm spring and other parts. According to the configuration of a double-clutch system, the gear shifting process is similar to the standard dry friction clutch operation.
Since each double-clutch’s output is connected to a set of two gear ratios, therefore timing and duration of gear shifting can be controlled. For the control of clutch engagement, the axial force of two clutches that are involved in gear shifting is controlled. It gives the possibility to change the instant clutch torque capacity during gear shifting.

4 Dynamic modelling of hybrid powertrain with double-clutch transmission

In this paper, the modelling and simulation is based on the parallel hybrid powertrain with a planetary transmission showed in Figure 2. The dynamic model of hybrid powertrain with a planetary transmission in Matlab/Simulink was depicted in authors’ previous publication (Szumanowski, 2013). The following equations are also used to build additional simulation model for evaluating the gear shifting control strategies.

The gear shifting control strategy is related to the dynamic parameters of vehicle – vehicle speed and dynamic load requirements. To evaluate gear shifting strategy, the gear shifting thresholds are calculated. And the maximal planetary gear output speed is calculated according to known desired maximal speed of electric machine and ICE (see Figure 7).

\[
\omega_3 = \frac{\omega_1 + k_p \left( \omega_2 \right)}{1 + k_p}
\]

(4)

where

\( \omega_1, \omega_2 \): ICE speed and electric motor speed

\( \omega_3 \): the planetary gear output speed

\( k_p \): the basic ratio of planetary gear

\( k_m \): the additional electric machine reduction gear ratio.

Figure 7 Scheme of planetary gear (see online version for colours)
For boundary conditions, it is assumed that changes from the first to the second gear occurs at the speed $V_1$ and the top speed for designed hybrid operation strategy is $V_4$. The gear shifting thresholds should obey the principle of geometrical progression.

$$V'_i = V_1 q^{i-1}$$  \hspace{1cm} (5)

Firstly, the $q$ factor of the series can be calculated. The extreme speed thresholds are used, therefore gear numbers used are $j = 4$ and $i = 1$ ($i, j \in \{1, 2, \ldots, n\}$).

$$q = \sqrt[4]{\frac{V_4}{V_1}}$$  \hspace{1cm} (6)

Then the gear ratios $i_i$ can be calculated in equation (5).

$$i_i = \frac{3.6 \omega_b r_i}{V_i j_d}$$  \hspace{1cm} (7)

where

- $i_i$: the ratio of selected gear $i$
- $\omega_b$: the motor speed operation limit
- $r_i$: the dynamic radius of wheel
- $V_i$: the vehicle speed threshold of gear $i$
- $j_d$: the ratio of differential gear.

The model of the planetary transmission has two degrees of freedom and three input torques. For hybrid drive, torques $T_1$ and $T_2$, which indicate ICE torque and electric motor torque respectively, are active. The inertias of rotating elements are included in coefficients in equations. Torque $T_3$ is represented by the torque of connected clutch. The inertia of the vehicle is not reduced on the planetary gear input shaft, but on the output shaft of the gearbox. The equations for planetary transmission are derived by using Lagrange equations, neglecting stiffness and damping. Also, inertia of planet wheels regarding their axis is neglected. And the kinematic relation between moving elements must be also included (Sumanowski, 2006).

$$\begin{bmatrix}
J_1 + \frac{1}{k_p} J_2 \\
-\frac{1+k_p}{k_p} J_2
\end{bmatrix} \ddot{\phi}_1 + \left[ J_3 + \frac{1+k_p}{k_p} \right] J_2 \ddot{\phi}_3 = T_1 - \frac{1}{k_p} T_2$$

$$\begin{bmatrix}
\frac{1+k_p}{k_p} J_2 \\
J_3 + \frac{1+k_p}{k_p}
\end{bmatrix} \ddot{\phi}_3 = T_3 + \frac{1+k_p}{k_p} T_2$$

\hspace{1cm} (8)

where

- $J_1, J_2, J_3$: the inertias associated with shafts of crown, ring and yoke respectively
- $T_1, T_2, T_3$: the torques on the shafts connected to the crown, ring and yoke respectively
\( \dot{\phi}_1, \dot{\phi}_2, \dot{\phi}_3: \) the acceleration of each shaft.

The first two equations in equation (8) can be transformed into different forms to estimate the system behaviour easily.

\[
\begin{align*}
\dot{\phi}_1 &= X_1 T_1 + X_2 T_2 + X_3 T_3 \\
\dot{\phi}_2 &= -\frac{1}{k_p} \phi_1 + \left(1 + \frac{k_p}{k_r}\right) \phi_3 \\
\dot{\phi}_3 &= X_1 T_1 + X_2 T_2 + X_3 T_3
\end{align*}
\]

(9)

where

\( X: \) the coefficients of equivalent inertia relevant to particular acceleration and torque.

The double-clutch model is based on the relation of axial force \( F_x \) and plates speed difference \( \Delta \omega \) which indicates the slip. For computational reasons, the plates speed difference cannot reach exact zero because of the torque-slip relation. The value of \( T_c \) and \( \Delta \omega \) must be calculated at every step by iterative methods, since the acceleration of clutch plates depends on carried torque. \( \Delta \omega_{lim} \) is some very small value which is set of calculation to approximate physical model.

\[
T_c = k(\Delta \omega) \mu r_c F_x
\]

(10)

where

\( k(\Delta \omega): \) the coefficient of clutch operation state as a function of clutch plates relative speed

\( \Delta \omega: \) the relative speed of clutch plates (slip).

The electric motor and ICE are dynamic sources of torque, which operate within their characteristics. Some further limits are applied to ensure that the operating points are in a certain area on those characteristics to decrease the energy consumption. ICE is assumed to operate around some selected speed range near 250 rad/s. This provides a faster transition to area of lower specific fuel consumption. The PM synchronous motor has a wider operating range which is limited by nominal torque and gear shifting speed threshold.

The vehicle model corresponds to light passenger car of 900 kg operating mass, used in urban and suburban areas. To achieve better regenerative braking performance, the control strategy includes proper gear shifting to avoid sudden electric motor speed drop which also increases electric motor efficiency. For gear shifting, it is crucial to avoid clutch slipping for decreasing system energy consumption. Therefore, the output torque from planetary gear during gear shifting is controlled (mainly by electric motor) to reduce
the speed difference between clutch plates that will be coupled together. The main target of clutch system control strategy study is to obtain good dynamic performance with low energy consumption.

5 Simulation of hybrid powertrain with double-clutch transmission

The simulation results are based on the dynamic simulation model of parallel hybrid powertrain with planetary gear and double-clutch transmission. Different simulations were done to investigate the powertrain control strategies, gear ratio adjusting, gear shifting control strategies and regenerative braking.

5.1 Simulation of parallel hybrid powertrain control strategies

To achieve lower fuel consumption and function of plug-in hybrid, two additional control strategies besides the basic one are designed for comparison (Figure 8). Separately, the torque on transmission shaft and demanded power of the vehicle influences changing operational modes between pure electric mode and hybrid mode for low speed driving (<70 km/h). For this reason, the two control strategies are named by ‘speed-torque’ and ‘speed-power’. The idea to compare these two strategies is to control the ICE starting time when the vehicle has enough electric energy. Furthermore, the battery SOC should be limited in a certain range for long distance driving.

Figure 8 Control strategy of planetary hybrid powertrain

The design and optimisation of the control strategy for hybrid powertrain involves the determination of powertrain parameters, physical features of components, driving performances, battery SOC operating range, etc. In ‘speed-torque’ strategy, the thresholds of speed and torque are set based on the engine map in order the engine can work in the high efficiency area. This control strategy may result in the battery SOC uncontrollable...
decreasing. For this reason, the ‘speed-power’ control strategy is proposed to control the battery SOC operating in proper range.

To study the influences of different control strategies, a comparison simulation was made. Figure 9 and Table 1 show the simulation results with different control strategies. For the same driving range, fuel consumption with ‘speed-torque’ control is 2% less than that with ‘speed-power’ control. With ‘speed-power’ control, the battery SOC is limited to the proper set value 0.5 at the end of the simulation. While with ‘speed-torque’ control, the battery SOC is out of control and decreases to 0.18, which means the powertrain could not work in hybrid mode for long distance driving to achieve better emission performance without too deeply discharging the battery. The detailed simulation results for different SOC ranges are presented in Figure 10.

Table 1  Simulation results with different control strategies for PHEV in NEDC cycle

<table>
<thead>
<tr>
<th></th>
<th>Speed-torque control strategy</th>
<th>Speed-power control strategy</th>
</tr>
</thead>
<tbody>
<tr>
<td>Total driving range [km]</td>
<td>540</td>
<td>540</td>
</tr>
<tr>
<td>Total fuel consumption [L]</td>
<td>13.55</td>
<td>13.82</td>
</tr>
<tr>
<td>Average fuel consumption [L/100km]</td>
<td>2.51</td>
<td>2.56</td>
</tr>
<tr>
<td>SOC at the end of simulation</td>
<td>0.18</td>
<td>0.52</td>
</tr>
</tbody>
</table>

Figure 9  Simulation results with different control strategies in repeating NEDC cycle
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Figure 10  Simulation results in one NEDC cycle with ‘Speed-power’ control strategy in different battery SOC ranges

Figure 11 shows the percentage of each operational mode time with different control strategy in the NEDC cycle. By analysing the simulation results, the pure electric mode, which is one main feature of the plug-in hybrid, is most used in driving cycle with both control strategies.

Considering the requirements of plug-in hybrid and similar fuel economy performance, a speed-power control strategy is better than speed-torque control strategy. The simulation in the following sections is based on speed-power control strategy. Additionally, the SOC threshold could be set to 0.4 in order to increase the fuel economy performance.

5.2  Simulation of parallel hybrid powertrain for gear ratio adjusting

The additional multi-speed gearbox is one of the significant changes in the parallel hybrid powertrain with planetary transmission. With properly adjusted gear ratios and gear shifting schedule, it could increase the energy efficiency for different driving
conditions relating to city traffic and travel in a suburban area. With consideration towards dynamic performance and fuel economy, the gear shifting schedule is as below.

1st-gear: 0~15 km/h  
2nd-gear: 15~40 km/h  
3rd-gear: 40~70 km/h  
4th-gear: 70~120 km/h (or higher speed)

Table 2 shows the trend that gear ratios influence the fuel consumption and efficiency of the motor. In selected range, the best ratios are highlighted. However, the real gear ratio optimisation is a more complicated approach because changing each gear ratio is connected with others. The adjustment of gear ratio should cooperate with observing the operating points of ICE and electric motor, because gear ratio optimisation is also limited by the practical performance and other boundary conditions. For example, the operating points of ICE and electric motor should be located in a limited range, and proper power of electric motor should be reserved for acceleration and grade ability.

**Figure 11** Percentage of operational mode time with different control strategies in NEDC cycle

Simulation results in Table 2 also show the necessity of a 4-speed gearbox. By equipping 4-speed gearbox, the average fuel consumption decreases 16.3% compared with a single gear ratio (see parameter No. 11 and 12 in Table 2). Figures 12 and 13 present operating points of ICE and electric motor with 4-speed gearbox transmission (see parameter No. 11 in Table 2). And Figures 14 and 15 present the operating points of ICE and electric motor with single gear ratio transmission (see parameter No. 12 in Table 2). With parameter No. 11, both ICE and electric motor operate in the zones with better efficiency.
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Table 2  Simulation results for different gear ratio of 4-speed gearbox

<table>
<thead>
<tr>
<th>No.</th>
<th>1st gear</th>
<th>2nd gear</th>
<th>3rd gear</th>
<th>4th gear</th>
<th>Average fuel consumption [L/100km]</th>
<th>Average efficiency of motor [%]</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>2.00</td>
<td>1.50</td>
<td>1.10</td>
<td>0.90</td>
<td>2.301</td>
<td>75.40</td>
</tr>
<tr>
<td>2</td>
<td>2.00</td>
<td>1.50</td>
<td>1.00</td>
<td>0.90</td>
<td>2.259</td>
<td>75.95</td>
</tr>
<tr>
<td>3</td>
<td>2.00</td>
<td>1.50</td>
<td>1.00</td>
<td>0.90</td>
<td>2.251</td>
<td>76.25</td>
</tr>
<tr>
<td>4</td>
<td>2.00</td>
<td>1.50</td>
<td>1.10</td>
<td>0.83</td>
<td>2.227</td>
<td>76.21</td>
</tr>
<tr>
<td>5</td>
<td>2.00</td>
<td>1.60</td>
<td>0.95</td>
<td>0.83</td>
<td>2.213</td>
<td>75.99</td>
</tr>
<tr>
<td>6</td>
<td>2.00</td>
<td>1.50</td>
<td>1.00</td>
<td>0.83</td>
<td>2.206</td>
<td>76.36</td>
</tr>
<tr>
<td>7</td>
<td>2.00</td>
<td>1.50</td>
<td>0.95</td>
<td>0.85</td>
<td>2.199</td>
<td>76.73</td>
</tr>
<tr>
<td>8</td>
<td>2.50</td>
<td>1.50</td>
<td>0.95</td>
<td>0.83</td>
<td>2.192</td>
<td>76.42</td>
</tr>
<tr>
<td>9</td>
<td>2.20</td>
<td>1.50</td>
<td>0.95</td>
<td>0.83</td>
<td>2.189</td>
<td>76.58</td>
</tr>
<tr>
<td>10</td>
<td>2.00</td>
<td>1.50</td>
<td>0.95</td>
<td>0.83</td>
<td>2.186</td>
<td>76.69</td>
</tr>
<tr>
<td>11</td>
<td>2.00</td>
<td>1.45</td>
<td>0.95</td>
<td>0.83</td>
<td>2.176</td>
<td>76.95</td>
</tr>
<tr>
<td>12</td>
<td>1.20</td>
<td>1.20</td>
<td>1.20</td>
<td>1.20</td>
<td>2.567</td>
<td>74.90</td>
</tr>
</tbody>
</table>

Simulation time: 30,000s; driving range: 270 km; battery state of charge alteration: 0.5 to 0.9.

The gear ratios in this table are referenced ratios which should be adjusted practically.

Figure 12  Operating points of ICE with 4-speed gearbox transmission (parameter no.11 in Table 2) (see online version for colours)

5.3  Simulation results of regenerative braking

When the hybrid powertrain works in the regenerative braking mode, the equivalent torque on the electric motor shaft influences energy efficiency. With proper control of a
4-speed gearbox, it could change the operating points of the electric motor to increase energy efficiency during the regenerative braking. The energy efficiency of regenerative braking relates to the kinetic energy of vehicle and electric energy received into the battery.

Figure 13  Operating points of electric motor with 4-speed gearbox transmission (parameter No. 11 in Table 2) (see online version for colours)

Figure 14  Operating points of ICE with 4-speed gearbox transmission (parameter No. 12 in Table 2) (see online version for colours)
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Figure 15  Operating points of electric motor without 4-speed gearbox transmission (parameter No. 12 in Table 2) (see online version for colours)

The comparison of exemplary simulation was carried out to analyse the energy efficiency of regenerative braking with and without gearbox for 900kg vehicle mass. Figures 16 and 17 demonstrate the operating points of electric motor during regenerative braking from 120 km/h to 0 km/h within 35 s. According to simulation results, the average efficiency of regenerative braking increases from 67.16% to 76.01% in the NEDC driving cycle by applying gear shifting.

Figure 16  Operating points of electric motor with 4-speed double clutch transmission during regenerative braking (parameter No.11 in Table 2) (see online version for colours)
6 Conclusions

This paper proposes the structure of CHPTD, which is a compact hybrid power train, with double-clutch transmission. An innovative double-clutch solution is demonstrated with aim of reducing energy loss and minimising cost.

The influence of gear ratios on powertrain’s efficiency improvement is studied in a dynamic simulation model of the designed hybrid powertrain with planetary and double-clutch transmission. According to the simulation results, the behaviours of hybrid powertrain are different when different control strategies are applied. The simulation results also show that the gear ratios of transmission influence the efficiency of powertrain sensitively. And it is proved that the double-clutch transmission could improve the efficiency of hybrid powertrain, and indeed also in the regenerative braking mode, by applying proper gear ratios and gear shifting strategy for different operational modes.

References


**Websites**
