Trade-off between Multi-mode Powertrain Complexity and Fuel Consumption

Namdoob Kim, Jason Kwon, and Aymeric Rousseau
Argonne National Laboratory, 9700 S. Cass Avenue, Argonne, IL, 60439-4815 USA
E-mail: nkim@anl.gov, jkwon@anl.gov, arousseau@anl.gov

Abstract— During the past couple of years, numerous powertrain configurations for hybrid electric vehicles have been introduced into the market-place. The current dominant architecture is the power-split configuration with the input split (one-mode) from Toyota and Ford. General Motors recently introduced a two-mode power-split configuration for applications in sport utility vehicles. Also, several car manufacturers have obtained patents for various additional options. The objective of this paper is to assess the benefits to fuel consumption as a result of added powertrain complexity (i.e., one-, two-, three-, and four-mode power splits), as demonstrated through several drive cycles. By using validated powertrain models and vehicle controls for the input and two-mode systems, the three-mode and four-mode vehicles were developed in Argonne’s Autonomie - a forward-looking powertrain simulation toolkit with dynamic plant models. This paper provides a detailed review of the benefits and drawbacks of each option.

Keywords— “EVT(electro-mechanical infinitely variable transmission),” “power-split,” “multi-mode,” “HEV(hybrid electric vehicle).”

1. Introduction

Various hybrid electric vehicle (HEV) architectures have been proposed, though one of the earliest and most commercially successful systems has been the power split, as used on all three generations of the Toyota Prius, other Toyota/Lexus models, as well as on the Ford Escape. The powertrain configuration of the power-split hybrid system, sometimes referred as the parallel/series hybrid, combines the previous two configurations with a power split device. It is appealing because with proper control strategy it can be designed to take advantage of both parallel and series types while avoiding their disadvantages. The power-split configurations have both all-mechanical and electro-mechanical paths combining the planetary gear set and two electric machines, as shown in Figure 1. In one path (the all-mechanical path), the power from the internal combustion engine (ICE) is directly transmitted to the wheels. In the other path (the electro-mechanical path), the power from the engine is converted into the electricity by a generator to drive the electric motor or to charge the battery [1,2]. A major advantage of this configuration stands in the possibility to de-couple the ICE and wheels speed as long as the output power demand is met, which gives much more flexibility to choose the ICE working point in order to optimize fuel consumption[3].

![Power-split transmission](image)

Figure 1: Power-split transmission

However, the power-split system is characterized by internal power circulation. In the power-split configuration, the internal power circulation occurs along the closed loop depending on the speed ratio, and sometimes the circulated power increases enormously. Therefore, electric machines are significantly oversized in order to meet requirements. This power circulation can lead to high losses and thereby to a low efficiency of the power transmission. It should be noted that power circulation is the primary reason that the Toyota Prius-type input-split HEV shows relatively low efficiency in the high-speed region. Such drawbacks can be addressed by combining several EVT (electro-mechanical infinitely variable transmission) modes in to one multi-mode hybrid system, thereby increasing the number of mechanical points and allowing greater operation flexibility. Various multi-mode EVT design configurations have been proposed, as indicated by patents and publications [4-8].

The EVT efficiency of the electro-mechanical power path is proportional to the powertrain (PT) configuration complexity in the multi-mode hybrid system, since electric power can stay low with wide ratio coverage and efficiency can remain high over a wider range. However, the multi-mode hybrid system should have more planetary gears (PGs) and clutches/brakes (CLs/BKs). Therefore, EVT mechanical loss is also proportional to powertrain configuration complexity in the multi-mode hybrid system. In this study, we evaluate the benefits of several multi-mode powertrain configurations with regard to fuel consumption and cost.

In the first part of the paper, we introduce the multi-mode hybrid system used in the study, as well as the modeling environment - the forward-looking Autonomie developed at Argonne National Laboratory [9,10]. By using validated powertrain models and vehicle controls for the input and two-mode systems, the three and four mode HEVs are developed. Each powertrain is sized to represent a small-size sport utility vehicle (SUV) application, following the same vehicle technical specifications, such as acceleration and gradeability. The impacts on fuel consumption as well as component operating conditions are examined.
2. Description of the Multi-mode Hybrid System

2.1. One-mode EVT

Figure 2 is a schematic diagram of the single-mode power split transmission (TM) with a reduction gear (RG). Since the input power from the ICE is split at the planetary gear which is located at the input side, and the power transmission characteristic is represented by a single relationship for the whole speed range, this power-split configuration is called the “input-split type” or “single-mode EVT.” This input-split configuration consists of two planetary gears, and two electric machines (MC1 and MC2). The larger electric machine on the right (MC1) is connected to the output shaft through the second planetary gear and does not affect the speed ratio. Therefore, for this particular EVT arrangement, which maximizes the output torque, the speed of the output is the weighted average of the speed of the input and the speed of MC2. The second planetary gear set multiplies the torque from the input and both of the electric motors during input-split operation. For comparison, the single mode powertrain without RG is also investigated in this study.

In Figure 3, the electro-mechanical power ratio and the EVT system efficiency (η) are plotted with respect to the speed ratio (SR). In this analysis, it is assumed that there is no power loss through the all-mechanical path and only electric machine loss is considered by using the efficiency maps of electric machines. The power ratio is defined as the ratio of the electro-mechanical power to the ICE input power and the SR is defined as the ratio of the ICE input speed to output speed. In high SR range, the system efficiency is low because the electrical machines have relatively low efficiency. This low system efficiency can be avoided by propelling the vehicle by using the electric motor directly instead of using the engine. When SR=0.7, the electro-mechanical power ratio becomes 0, and all the power is transmitted through the mechanical part. This point is called the mechanical point (MP). The system efficiency shows the highest value at the MP. For SR<0.7, the electro-mechanical power ratio has a negative(-) value, which means that the power is circulating along the closed path. It is apparent that the circulated power increases as the SR decreases. Once the power circulation occurs, the EVT efficiency decreases due to the relatively low efficiency of the electro-mechanical power path. The high circulated power results in the decreased transmission efficiency and requires large electric machines. In addition, this high power requires consideration of the mechanical part design. The analysis results demonstrate why the Toyota hybrid system (THS), a typical example of the input-split HEV, adopts large capacity electric machines.

2.2. Two-mode EVT

Figure 4 shows a schematic diagram of the two-mode EVT used in buses, which is the direct ancestor and basis for the two-mode hybrid for large cars and trucks [5]. The two-mode EVT contains two planetary gear sets, which are required for a compound power split. In the two-mode EVT, they are used for both the input split and compound split, depending on which of the two clutches in the transmission is activated. The second planetary gear set multiplies the torque from the input and both electric motors during input-split operation. The first clutch (CL1) is a stationary clutch or brake that activates the input-split mode and low-speed torque multiplication by holding the ring gear of the second planetary gear set. The second clutch (CL2) is a rotating clutch that activates the
compound-split mode by connecting the sun of the first planetary gear set to the ring of the second gear set.

Figure 5 shows that the electro-mechanical power ratio becomes 0 at MP1 and MP2, where the SR becomes SR=1.67 and SR=0.56, respectively, in the EVT1 mode. At SR=1.67, the operating mode is changed to the EVT2 mode. In the EVT2 mode, the power ratio also becomes 0 at SR=0.56, which is a second mechanical point. The EVT1 mode is maintained until the speed ratio reaches SR=1.67. At this point, the shift from the EVT1 mode to EVT2 mode is carried out by the clutch and brake operations. The system efficiency at the EVT2 mode shows relatively high value, because the electric power ratio at the EVT2 mode is relatively smaller than that of the EVT1 until the speed ratio becomes SR=0.56. A second mechanical point provides the ability to restrain both continuous electric machine power during cruising and peak motor power during acceleration. A two-mode EVT with both an input-split mode, with one mechanical point, and a compound-split mode, with two additional mechanical points, fundamentally lowered the requirement for motor power, thus allowing the EVT to be selected as a sound basis for large cars and trucks.

2.3. Two-mode EVT with Fixed Gear Ratios

Figure 6 is a schematic of the two-mode EVT, which is called the General Motors Advanced Hybrid System2 (AHS2) for front-wheel drive (FWD) [6]. This system has an additional stationary clutch and an additional rotating clutch. Through engaging or disengaging the four clutches, it realizes six different operation modes including two EVT modes and four fixed gear (FG) modes. When operated in any of the four fixed gear modes, the vehicle is comparable to a parallel pre-transmission HEV.

In Figure 7, the two-mode EVT already has a native fixed gear ratio, the synchronous shift ratio, where the action of two clutches at the same time provides a fixed ratio. For the two-mode hybrid, one fixed gear was added within the ratio range of the first EVT mode, and two more fixed gears were added within the ratio range of the second EVT mode. So, for the two-mode hybrid the native fixed gear between the two EVT modes is fixed gear 2 (FG2). The top fixed gear ratio, fixed gear 4 (FG4), was added by putting a stationary clutch on one of the motors that regulates the speed ratio through the transmission, MC1. Fixed gear 1 (FG1), and fixed gear 3 (FG3), were both added with a rotating clutch. The FG1 comes from locking up the input-split mode, so the speed, torque, and power from the engine go through the torque multiplication of the second planetary gear set. The FG3 comes from locking up the compound-split mode, so the speed, torque, and power from the engine are coupled directly to the output. This study also investigates the additional two-mode EVT with fixed gears, which is called AHS2 for rear-wheel drive (RWD) [6,8].
2.4. Three-mode EVT with Fixed Gear Ratios

Figure 8: Schematic of the three-mode EVT with FGs

In order to develop the plant model and the controller of the multi-mode hybrid system, we need to use equations that describe the operations of the system. To provide ease of use, the equations are simplified versions of the more complex ones. In the case of an EVT mode, only two differential equations are required to represent the powertrain system, since there are only two independent state variables – engine speed ($\omega_e$) and vehicle velocity ($V$). With mathematical manipulation, the dynamic equations can be obtained as follows by using four factors, $p_j$, $q_j$, $r_j$, and $s_j$ that are specific to mode $j$:

$$\alpha \cdot \dot{X} = \beta \cdot u$$

where $X = [\omega_e \ V]^T$, $u = [T_e \ T_{MC2} \ T_{MC1} \ F_L]^T$, $\alpha = \frac{J_e + p_j^2 J_{MC2} + r_j^2 J_{MC1}}{p_j q_j J_{MC2} + r_j s_j J_{MC1}}$, $\beta = \begin{bmatrix} 1 & p_j & r_j & 0 \\ 0 & q_j & s_j & -1 \end{bmatrix}$

where $J_e$ is the inertia of the ICE, $J_{MC2}$ and $J_{MC1}$ are the inertia of the electric machine, $J_w$ is the wheel inertia, $R_t$ is the tire radius, $M$ is the vehicle mass and $F_i$ is the road load. $T$ and $\omega$ denote the torque and speed of each component. The parameters are defined in Appendix Table A-1, as well as in Appendix Table A-2. From Eq. (1), the dynamics of the EVT hybrid powertrain can be represented in a state-space equation as follows:

$$\begin{cases} \dot{X} = A \cdot X + B \cdot u \\ Y = C \cdot X + D \cdot u \end{cases}$$

where $Y = [\omega_e \ \omega_{MC2} \ \omega_{MC1} \ V]^T$, $A = 0$, $B = \alpha^{-1} \beta$, $C = \begin{bmatrix} 1 & p_j & r_j & 0 \\ 0 & q_j & s_j & 1 \end{bmatrix}$, $D = 0$

In the case of a fixed gear mode, the speed and torque relationships are similar to those of a conventional multispeed transmission. They are given as follows:

$$T_e = k_e T_o^i + k_{MC2} T_{MC2} + k_{MC1} T_{MC1}$$

$$\begin{cases} \omega_e = k_e^i \cdot \omega_o \\ \omega_{MC2} = k_{MC2}^i \cdot \omega_o \\ \omega_{MC1} = k_{MC1}^i \cdot \omega_o \end{cases}$$

where $T_o / \omega_o$ is the output torque/speed of the transmission, and $k^i$ is the multiplication ratio for each component, as given in Appendix Table A-3.

3. Component Sizing

3.1. Modeling the Vehicle in Autonomie

Autonomie is a forward-looking modeling tool that can simulate a broad range of powertrain configurations. The driver model computes the torque demand needed to meet the vehicle speed trace. The torque demand is interpreted by a high-level controller that computes the component’s torque demands while ensuring that the system operates within its constraints. Detailed transmission models were developed by using SimDriveline, including specific
losses for gear spin and hydraulic oil [11], as shown in Figure 10. Such a level of detail is necessary to properly assess the trade-off between complexity and efficiency.

![Figure 10: Transmission model for the AHS2 FWD](image)

3.2. Mode Selection during Acceleration

The powertrain must be operated in such a way that the output shaft can transmit the maximum torque from the powertrain to the final drive. This can be regarded as an optimization problem, and it is solved for the corresponding vehicle speed. An off-line computation was used to generate the maximum powertrain torque that could be used in Simulink, indexed by gearbox output speed and battery power. To compute that look-up table, a brute-force algorithm is used, possibly several hundreds of times, along an output speed grid ranging from 0 to its maximum.

The maximum powertrain torque curves are obtained along the scheme and displayed in Figure 11. In acceleration simulations, the mode that allows the maximum output torque is selected. From Figure 11b, it can be seen that FG1 increases the vehicle tractive capability significantly in the range of 10-45 mph. The use of FG1 over a large range of vehicle speeds eliminates the need to use the motors for processing engine power, thereby freeing up capacity to boost acceleration.

![Figure 11: Max torque during acceleration](image)

3.3. Sizing Process

To quickly size the component models of the powertrain, an automated sizing process was used [12]. The sizing process defines the peak mechanical power of the electric machine as being equal to the peak power needed to follow the acceleration constraints. The peak discharge power of the battery is then defined as the electrical power that the electric machine requires to produce its peak mechanical power. The sizing process then calculates the peak power of the engine by using the power of the drivetrain required to achieve the gradeability requirement of the vehicle.

<table>
<thead>
<tr>
<th>Specifications of the small-size SUV</th>
<th>Body and chassis mass</th>
<th>1180 kg</th>
<th>Frontal area</th>
<th>2.64 m²</th>
</tr>
</thead>
<tbody>
<tr>
<td>Drag coefficient</td>
<td>0.37</td>
<td>Wheel radius</td>
<td>0.3423 m</td>
<td></td>
</tr>
<tr>
<td>Battery</td>
<td>NIMH, 1.2 volt/cell, 6 Ah</td>
<td>Electrical accessory</td>
<td>200 W</td>
<td></td>
</tr>
<tr>
<td>Final drive ratio</td>
<td>Single mode : 4.11</td>
<td>Multi mode : 3.02</td>
<td></td>
<td></td>
</tr>
<tr>
<td>PGs ratio (Zr/Zs)</td>
<td>Single mode : 2.6</td>
<td>Single mode with RG : 2.4, 2.0</td>
<td>Two mode without FGs: 1.5, 1.5</td>
<td>AHS2 FWD : 2.36, 2.24</td>
</tr>
</tbody>
</table>
As detailed previously, the component’s characteristics determine the constraints. The main vehicle characteristics used in this study are summarized in Table 1. Particularly, the ratios of planetary gear sets are from patents and references. The 0-60 mph performance requirement for the vehicle is satisfied implicitly by the constraints on the peak motor power and the peak engine power. The power required by the motor for the vehicle to follow the UDDS cycle added to the power required by the engine for the vehicle to drive up a 13% grade at 65 mph exceeds the power that the vehicle needs to go from 0 to 60mph in 7.8 sec.

3.4. Sizing Results

The sizing results are summarized in Figure 12 and Appendix Table A-4. For comparison, two single-mode EVT hybrid systems and four multi-mode EVT hybrid systems are investigated, and the results are presented. As noted in the introduction, the multi-mode system results in significant improvements in dynamic performance at reduced capacities of the electro-mechanical power. As can be seen in Figure 12, the amount of capacities that saved by the multi-mode system ranges from 31.7% to 64.3%, relative to the single-mode. The main contributor is the addition of the EVT mode, which causes the difference between the single-mode and multi-mode systems. However, there is little difference between the three-mode and four-mode systems.

In order to evaluate the benefits of several multi-mode powertrain configurations from the standpoint of fuel consumption, an HEV control strategy is required first. One of the major challenges of the multi-mode control strategy is to properly select the operating mode. In order to develop mode shift strategy, a brute-force algorithm is used. The algorithm generates an optimal input speed and torque for each EVT mode, indexed by gearbox output speed, battery power, and gearbox output torque. The knowledge of these parameters allows us to compute the fuel power and to compare it with that in the other EVT modes [13,14]. Meanwhile, obtaining a candidate input set for FGs is same as conventional way. Figure 13a depicts the optimal mode selections for various output load conditions. If we convert these results into new map by using vehicle speed and engine speed indexes, the mode selection rule is defined based on the speed ratio. The reason for this is because the selected optimal mode could be divided according to the speed ratio, which is defined as the ratio of the target engine speed to the output speed. When in propelling mode, the target engine speed previously computed by the simplified system-optimal-operating-line for driver power demand. In Figure 13b, the FG1 mode appears in the transition area between the EVT1 and EVT2 modes and the FG2 mode appears in the area between the EVT2 and EVT3 modes. The FG1 and FG2 mode are inherent modes needed for the synchronous shift between the two EVT modes. The FG3 mode supplements the EVT3 mode. The logic was validated for both single-mode and two-mode hybrid systems by using vehicle test data [15]. Similar algorithms were implemented for the three-mode and four-mode configurations.

4. Control Strategy

The hybrid operating functions are the rule-based part of an HEV control strategy. Based on rules commonly used for HEV control and that are not configuration specific, their role is to compute the engine ON/OFF demand signal as well as the battery power demand needed to keep the state-of-charge (SOC) balanced.

The engine is turned ON when the electric torque is insufficient to remain in electric vehicle (EV) mode or if the battery SOC is too low. However, the most common reason for the engine to turn ON occurs when the driver...
power demand is higher than the power threshold (and is not decreasing) and the engine has been OFF for a minimum time. Alternatively, the engine is generally shut down because the driver power demand falls below a certain threshold while it has been ON for a minimum time. The other necessary conditions are that the EV mode is possible and the vehicle speed is not too high.

The SOC control is accomplished through a load-leveling power strategy and a proportional power gain: given the difference between the target SOC and the current SOC, a control outputs a power demand that is later used to compute the component torques. Furthermore, the difference between the target SOC and the current SOC defines the engine ON and OFF thresholds based on the difference between target SOC and current SOC. Thus, when the battery SOC is lower than the target SOC, the engine will be started “earlier” (i.e., for lower levels of driver power demand). As a result, less electrical energy will be discharged, which, combined with regenerative braking, may lead to a net SOC increase in a short-term time frame. Finally, a hysteresis-type of control sets discrete battery levels (i.e., low, normal, or high) that will determine some decisions (e.g., the engine cannot shut down if the battery SOC level is low).

5. Simulation Results

5.1. Comparative Analysis

![Figure 14: Fuel economy summary](image)

With the transmission models and controller described in the previous section, the vehicle was simulated on standard drive cycles: the urban dynamometer driving schedule (UDDS); the highway fuel economy test (HWFET) cycle; the new European driving cycle (NEDC); a more aggressive urban cycle with some short highway cycles (LA92); and a highly aggressive cycle, predominantly at high speed (US06). The fuel economy results are reported in Figure 14. For urban driving, the single-mode hybrid system has relatively high fuel economy compared with that of the multi-mode hybrid system. On the other hand, the trend shown by the different cycles indicates that the higher the speed of the driving pattern, the greater the advantage of the multi-mode hybrid system. Figure 15 also shows that the more aggressive/faster the driving pattern, the greater the advantage of the multi-mode with a high fixed gear ratio, FG1. However, the three-mode and four-mode systems do not result in more benefits for the aggressive drive cycle. As a consequence, the AHS2 FWD provides a greater fuel consumption advantage for the vehicle application considered on the small-size SUV specification.

![Figure 15: Fuel economy for drive cycle statistics](image)

5.2. Urban and Highway Cycle Operations

Figure 16 reports the operating points of the powertrain for urban and highway driving. It is remarkable how the vehicles operate in the regions of higher efficiency to reduce fuel consumption. As shown in the figures, the single-mode hybrid system has relatively lower system efficiency in the primary operating region, particularly for highway driving. This occurs because the electro-mechanical power increases sharply as the transmission reaches higher overdrive (Figure 3). The operating points of the multi-mode hybrid system are between the mechanical points to achieve the high EVT efficiency. For the multi-mode hybrid system, the EVT2 or EVT3 modes are highly used in urban driving, and highway cycles favor the use of fixed gears. The first mode, EVT1, is used only in some low-speed, low-acceleration areas or during EV operation.
Figure 16: Operating points for UDDS and HWFET

Table 2 shows the efficiencies of all three power sources and powertrains. The transmission efficiency refers only to the all-mechanical path. The single-mode hybrid system has the highest transmission efficiency, since there is no need for more planetary gears or clutches. For the multi-mode system with the fixed gear, it is interesting to note that the efficiencies of the ICE are not particularly high. This effect is due to the fixed gear, which improves the highway fuel economy by avoiding the lower system efficiency region to maintain holding torque at the second mechanical point. The energy losses results of the transmissions under urban and highway cycles are shown in Figure 17. The single-mode hybrid system has the smallest transmission energy losses. Because there is need for the clutches in the multi-mode transmission, there are additional hydraulic oil losses.

Table 2: Component average efficiencies, %

<table>
<thead>
<tr>
<th></th>
<th>UDDS</th>
<th>S1</th>
<th>S2</th>
<th>M1</th>
<th>M2</th>
<th>M3</th>
<th>M4</th>
<th>M5</th>
</tr>
</thead>
<tbody>
<tr>
<td>Engine</td>
<td>33.3</td>
<td>33.4</td>
<td>33.1</td>
<td>30.9</td>
<td>31.1</td>
<td>31.9</td>
<td>32.0</td>
<td></td>
</tr>
<tr>
<td>MC1</td>
<td>87.0</td>
<td>87.5</td>
<td>86.1</td>
<td>86.5</td>
<td>86.7</td>
<td>86.4</td>
<td>86.4</td>
<td></td>
</tr>
<tr>
<td>MC2</td>
<td>86.2</td>
<td>86.1</td>
<td>86.0</td>
<td>86.0</td>
<td>86.7</td>
<td>86.9</td>
<td></td>
<td></td>
</tr>
<tr>
<td>TM</td>
<td>96.1</td>
<td>96.4</td>
<td>92.9</td>
<td>90.5</td>
<td>89.5</td>
<td>89.1</td>
<td>88.7</td>
<td></td>
</tr>
<tr>
<td>PT</td>
<td>33.5</td>
<td>33.6</td>
<td>33.2</td>
<td>31.6</td>
<td>30.9</td>
<td>30.6</td>
<td>30.1</td>
<td></td>
</tr>
<tr>
<td>HWFET</td>
<td></td>
<td>S1</td>
<td>S2</td>
<td>M1</td>
<td>M2</td>
<td>M3</td>
<td>M4</td>
<td>M5</td>
</tr>
<tr>
<td>Engine</td>
<td>33.8</td>
<td>33.7</td>
<td>33.5</td>
<td>31.6</td>
<td>30.6</td>
<td>30.7</td>
<td>30.6</td>
<td></td>
</tr>
<tr>
<td>MC1</td>
<td>91.4</td>
<td>89.9</td>
<td>86.1</td>
<td>86.6</td>
<td>87.2</td>
<td>87.0</td>
<td>87.0</td>
<td></td>
</tr>
<tr>
<td>MC2</td>
<td>86.5</td>
<td>86.5</td>
<td>86.7</td>
<td>86.6</td>
<td>85.4</td>
<td>85.4</td>
<td></td>
<td></td>
</tr>
<tr>
<td>TM</td>
<td>94.7</td>
<td>93.4</td>
<td>88.9</td>
<td>89.0</td>
<td>88.9</td>
<td>90.2</td>
<td>90.3</td>
<td></td>
</tr>
<tr>
<td>PT</td>
<td>25.9</td>
<td>26.0</td>
<td>26.9</td>
<td>26.7</td>
<td>26.2</td>
<td>26.5</td>
<td>26.5</td>
<td></td>
</tr>
</tbody>
</table>

(S1: single mode; S2: single mode with RG; M1: two mode without FGs; M2: AHS2 FWD; M3: AHS2 RWD; M4: three mode; M5: four mode; TM: transmission; PT: powertrain)

Figure 17: Transmission losses for UDDS and HWFET

6. Conclusion

This paper examines the seven different power-split configurations and vehicle-level controls developed in Autonomie. Detailed transmission models were implemented to allow a fair assessment of the trade-offs between complexity and fuel efficiency. The powertrains were compared theoretically from EVT system efficiency.
It was found that the multi-mode EVT minimizes the power ratio, which means that we can directly transfer as much mechanical power as possible. Each powertrain was sized to represent a small-size SUV application, following the same vehicle technical specifications, such as acceleration and gradeability. The results predicted that the multi-mode system would have better acceleration performance than a single-mode system, since the additional EVT modes significantly lower the requirement for the electric machine power. In addition, simulations were performed on a small-size SUV to characterize the impact on component operating conditions and fuel consumption for several driving cycles. It was determined that the multi-mode system has more fuel economy advantage during the high-speed cycle due to the relatively higher system efficiency. When the cycle is more aggressive, the multi-mode system with FGs has the advantage due to the relatively higher tractive capability.

To ensure a fair comparison, further work should strive to integrate additional vehicle classes (e.g., compact, midsize car, midsize SUV, etc.) and also consider additional vehicle technical specifications (e.g. passing, towing, etc.). Several drive cycles, including real-world drive cycles, should also be evaluated.

7. Abbreviations

AHS advanced hybrid system
BK brake
CL clutch
EV electric vehicle
EVT electro-mechanical infinitely variable transmission
FG fixed gear
FWD front-wheel drive
HEV hybrid electric vehicle
HWFET highway fuel economy test
ICE internal combustion engine
MC electric machine/motor
MP mechanical point
NEDC new European drive cycle
PG planetary gear
PT power train
RG reduction gear
RWD rear-wheel drive
SOC state-of-charge
SR speed ratio
SUV sport utility vehicle
TM transmission
UDDS urban dynamometer driving schedule

8. References

10. Argonne National Laboratory, PSAT (Powertrain System Analysis Toolkit), www.transportation.anl.gov/modeling_simulation/PSAT

9. Authors

Namdoou Kim
Research Engineer
Argonne National Laboratory,
9700 S. Cass Avenue,
Argonne, IL 60439, USA
Tel: +1-630-252-2843
Email: nkim@anl.gov

Namdoou Kim graduated in 2007 from the University of Sungkyunkwan, Korea, with a Master’s degree in School of Mechanical Engineering. He is currently working in...
Argonne National Laboratory’s Vehicle Modeling and Simulation group.

**Jason Kwon**  
Research Engineer  
Argonne National Laboratory,  
9700 S. Cass Avenue,  
Argonne, IL 60439, USA  
Tel: +1-630-252-4154  
Email: jkwon@anl.gov

Jason Kwon received his MSME degree from Ohio State University in 2000. Previously he worked at Ford Motor Company as a System Analyst. He is currently working as research engineer Argonne National Laboratory’s Vehicle Modeling and Simulation group since 2005.

**Aymeric Rousseau**  
Program Manager  
Argonne National Laboratory,  
9700 S. Cass Avenue,  
Argonne, IL 60439, USA  
Tel: +1-630-252-7261  
Email: arouss@anl.gov

Aymeric Rousseau received his Master of Science in Industrial Systems from EIGSI in La Rochelle, France, in 1997. He is currently leading Argonne’s Vehicle modeling and simulation group.

10. Acknowledgment

This work was supported by DOE’s Vehicle Technology Office under the direction of Lee Slezak. The submitted manuscript has been created by UChicago Argonne, LLC, Operator of Argonne National Laboratory (“Argonne”). Argonne, a U.S. Department of Energy Office of Science laboratory, is operated under Contract No. DE-AC02-06CH11357. The U.S. Government retains for itself, and others acting on its behalf, a paid-up nonexclusive, irrevocable worldwide license in said article to reproduce, prepare derivative works, distribute copies to the public, and perform publicly and display publicly, by or on behalf of the Government.

11. Appendix

**Table A-1: Definitions of ratio variables**

<table>
<thead>
<tr>
<th>Configuration</th>
<th>EVT</th>
<th>p_j</th>
<th>q_j</th>
<th>r_j</th>
<th>s_j</th>
</tr>
</thead>
<tbody>
<tr>
<td>Single mode</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Single mode w/RG</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Two mode(1)</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>AHS2 FWD</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

**Table A-2: Definitions of simplified factors**

$$ z = \text{Final drive ratio} \quad \Delta = (ad+acf-f) $$

<table>
<thead>
<tr>
<th>Configuration</th>
<th>a</th>
<th>b</th>
<th>c</th>
<th>d</th>
<th>e</th>
<th>f</th>
</tr>
</thead>
<tbody>
<tr>
<td>Single mode</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Single mode w/RG</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Two mode(1)</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>AHS2 FWD</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>AHS2 RWD</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

**Table A-3: Fixed gear ratios**

<table>
<thead>
<tr>
<th>Configuration</th>
<th>FGs</th>
<th>k_i</th>
<th>k_{MC2}</th>
<th>k_{MC3}</th>
</tr>
</thead>
<tbody>
<tr>
<td>AHS2 FWD</td>
<td>1</td>
<td>c</td>
<td>c</td>
<td>c</td>
</tr>
<tr>
<td>AHS2 RWD</td>
<td>3</td>
<td>1</td>
<td>1</td>
<td>1</td>
</tr>
</tbody>
</table>

**Table A-4: Component sizing results**

<table>
<thead>
<tr>
<th>Configuration</th>
<th>ICE (kW)</th>
<th>MC2 (kW)</th>
<th>MC1 (kW)</th>
<th>Battery (Cells)</th>
<th>Mass (kg)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Single mode</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Single mode w/RG</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Two mode(1)</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>AHS2 FWD</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>AHS2 RWD</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Mode</td>
<td>131.5</td>
<td>91.7</td>
<td>91.2</td>
<td>180</td>
<td>1953</td>
</tr>
<tr>
<td>---------------------</td>
<td>-------</td>
<td>------</td>
<td>------</td>
<td>-------</td>
<td>--------</td>
</tr>
<tr>
<td>Single mode</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Single mode w/ RG</td>
<td>131.1</td>
<td>80.9</td>
<td>88.2</td>
<td>173</td>
<td>1940</td>
</tr>
<tr>
<td>Two mode(1)</td>
<td>125.3</td>
<td>62.6</td>
<td>52.4</td>
<td>170</td>
<td>1883</td>
</tr>
<tr>
<td>AHS2 FWD</td>
<td>124.8</td>
<td>58.1</td>
<td>47.2</td>
<td>170</td>
<td>1875</td>
</tr>
<tr>
<td>AHS2 RWD</td>
<td>125.3</td>
<td>62.6</td>
<td>52.2</td>
<td>171</td>
<td>1883</td>
</tr>
<tr>
<td>Three mode</td>
<td>123.3</td>
<td>61.6</td>
<td>32.6</td>
<td>157</td>
<td>1857</td>
</tr>
<tr>
<td>Four mode</td>
<td>123.3</td>
<td>61.6</td>
<td>32.6</td>
<td>157</td>
<td>1857</td>
</tr>
</tbody>
</table>