Development of a Heavy Duty Commercial Vehicle Dynamic Model with a Hydraulic-Mechanical Transmission Model Based on Field Test Results of a Vehicle with Automatic Transmission

Beomjoon Pyun1, Chulwoo Moon1,*, Changyun Jeong1 and Dohyun Jung1
1Korea Automotive Technology Institute, 31214 Chungnam, Republic of Korea

*Corresponding author’s e-mail: cwmoon@katech.re.kr

Abstract. The modeling of dynamic systems is advantageous for saving time and effort when developing new systems. In the automobile industry, making a model is beneficial for predicting the feasibility and performance of a conceptual design. Hence, in this study, a heavy duty commercial vehicle with an Automatic Transmission (AT) was tested, and the model was developed and verified. A conceptual design model of Hydraulic-Mechanical Transmission (HMT) was then developed. The commercial vehicle model and the HMT model were integrated and compared to evaluate the feasibility and the difference in performance.

1. Introduction
Building a vehicle model is advantageous for saving time and effort when it comes to developing a new transmission module in the automobile industry. A transmission model should be considered in a vehicle model because the operational environment of transmission is affected by vehicular reaction forces. For this reason, a vehicle model with a commercial transmission model needs to be compared to a vehicle model with a newly developed transmission model to verify the performance [1]. Furthermore, in the conceptual design stage of a vehicle’s development, a conceptual transmission model should be compared to the commercial transmission as a vehicle model in order to predict its feasibility and performance. A conceptual transmission should then be verified with a commercial transmission to predict its feasibility and performance.

In this paper, a heavy duty commercial vehicle model was developed based on vehicle tests, and a Hydraulic-Mechanical Transmission (HMT) model was developed using conceptual specifications and integrated into the vehicle model to predict the feasibility and performance of the HMT. Section 2 describes how a heavy duty commercial vehicle (AT vehicle) model was developed and verified with the vehicle test results. Section 3 introduces the concept of HMT, and explains how the model developed. Section 4 explains how the commercial vehicle model was integrated with the HMT model and how the models compare.

2. Heavy Duty Commercial Vehicle Modeling Based on Vehicle Tests
Vehicle dynamics is basically considered in order to develop a simulation environment of a heavy duty commercial vehicle model based on vehicle test results [2]. In vehicle dynamics, longitudinal
dynamics is the predominant factor for developing a transmission, and the longitudinal model is comprised of a powertrain model and a vehicle and road loads model [3].

![Schematic of a powertrain power flow](image)

**Figure 1.** Schematic of a powertrain power flow

\[
I_e \alpha_e = T_e - T_c \quad (1)
\]

\[
I_e \alpha_e = T_e - \frac{1}{N_t} T_d \quad (2)
\]

\[
I_d \alpha_d = T_d - \frac{1}{N_f} T_w \quad (3)
\]

\[
I_w \alpha_w = T_w - r F_x \quad (4)
\]

\[
F_x = \left[ -I_w \alpha_w - N_f^2 (I_c + I_f) \alpha_w + N_f T_e - N_f^2 I_d \alpha_w \right] / r \quad (5)
\]

Where, \( \alpha_e = N_e \alpha_e, \alpha_d = N_d \alpha_d, \alpha_e = N_e \alpha_e, \) and \( I_e \) is the engine inertia (kgm\(^2\)), \( \alpha_e \) is the angular acceleration of the engine (rad/s), \( T_e \) is the engine torque (Nm), \( T_c \) is the torque of the torque converter (Nm); \( I_t \) is the transmission inertia (kgm\(^2\)), \( N_t \) is the gear ratio of transmission (-), \( T_d \) is the differential torque (Nm), \( I_d \) is the differential gear inertia (kgm\(^2\)), \( \alpha_d \) is the angular acceleration of the differential gear (rad/s), \( N_f \) is the gear ratio of the differential gear (-), \( T_w \) is the wheel torque (Nm), \( I_w \) is the wheel inertia (kgm\(^2\)), \( \alpha_w \) is the angular acceleration of the wheel (rad/s), \( r \) is the wheel effective radius (m), and \( F_x \) is the traction force (N).

Figure 1 shows the schematic of a powertrain power flow. The power flow can be described by the following formulas: Formula (1) describes the power transfer of the engine, formula (2) describes the power transfer of the transmission, formula (3) describes the power transfer of the differential gear, and formula (4) describes the power transfer of the wheel. In formula (5), engine torque and angular acceleration are transferred to the wheel as a tractive force [4]. Many other aspects should also be considered for powertrain system modeling: Engine map, torque converter properties, transmission scheduling map, type of transfer case, gear ratio of the differential gear, wheel torque, and so on. In this modeling of a powertrain system, these are obtained and updated to the powertrain model.
Figure 2. Schematic of a vehicle and road load model

\[ F_a = F_{sf} + F_{sr} \]  
\[ R_x = R_{sf} + R_{sr} \]  
\[ D_A = \frac{1}{2} \rho C_d A V^2 \]  
\[ \frac{W}{g} a_x = F_{sf} + F_{sr} - D_A - (R_{sf} + R_{sr}) - W \sin \theta \]

Where, \( F_a \) is the traction force (N), \( R_x \) is the rolling resistance (N), \( D_A \) is the drag force (N), \( a_x \) is the vehicle acceleration (m/s^2), \( \theta \) is the road slope (rad), \( \rho \) is the atmospheric pressure (Pa), \( C_d \) is the drag coefficient (-), and \( A \) is the frontal area (m^2).

Figure 2 shows a vehicle and road loads model composed of traction force and opposite forces. Formula (9) is the traction force of a vehicle, and formulas (10) and (11) are the opposite forces. Formula (12) is a longitudinal dynamics formula considering these forces. These formulas can be described as the schematic shown in Figure 2. With these formulas, a vehicle and road loads model can be developed.

Figure 3. Heavy duty commercial vehicle (AT vehicle) model verification

To verify a longitudinal vehicle model, maximum error and correlation (Pearson’s r) [5] of the longitudinal speed are generally used. Hence, using the Throttle Position input in Figure 3 (a) from the
vehicle test, the Longitudinal Speed outputs could be compared. The correlation result is shown in Figure 3 (b). The AT heavy vehicle model was developed and verified with a maximum error of 2.95km/h and correlation of 99.68%.

3. Hydraulic-Mechanical Transmission Modeling

Hydraulic-Mechanical Transmission (HMT) is a transmission that combines mechanical transfer parts and a Hydrostatic Unit (HSU) or a Hydrostatic Transmission (HST) part.

![Figure 4. Schematic of a Hydrostatic Unit](image)

Figure 4 shows an HSU operated by hydraulic power. The flow rate in the HSU is changed by Plate Angle Control, and the gear ratio is changed by that.

HMT is a transmission that has an HSU part integrated with mechanical parts because an HSU has the following disadvantages: low efficiency at high speed and narrow range of gear ratio. An HMT is more advantageous than Continuously Variable Transmission (CVT) because an HMT has better instantaneous response, durability, and efficiency in cruising speed. Additionally, an HMT is more advantageous than Automatic Transmission (AT) in terms of its instantaneous response, and it also has better efficiency at low speed. However, an HMT has disadvantages in that it has high processing costs, and low efficiency at high speed. As a result, increasing powertrain efficiency and fuel efficiency is the ultimate goal using a control algorithm because an HMT can provide good performance with a control algorithm. Section 3 describes how an HMT model became developed from a conceptual design.

![Figure 5. Conceptual design of Hydraulic-Mechanical Transmission](image)

Figure 5 shows a conceptual design of an HMT. In Figure 5, $Z_k$ indicates that the k-th gear’s name and formulas including division are the gear ratio. The HMT is composed of two clutches, two planetary gears, one brake, and one HSU part. From this conceptual design, an HMT model was developed using the commercial simulation tool, AMESim.
Figure 6. Simulation model of Hydraulic-Mechanical Transmission

Figure 6 shows the simulation model of HMT, developed from the conceptual design. Torque from the engine is transferred to the Input Part, and reaction rotational speed is transferred from the drive shaft to the Output Part. With the power transfer, Plate Angle is controlled to control the continuous gear ratio, and Clutches 1/2/3 are controlled to control discrete gear ratio.

4. Integrated Simulation Environment of an AT vehicle model and an HMT model
A heavy duty commercial vehicle model was developed with an Automatic Transmission model from the vehicle test results. The model comprised of a powertrain model and a vehicle and road loads model because the major consideration is longitudinal dynamics [6]. Based on the AT vehicle model, the HMT model from Section 3 was integrated into the AT vehicle model.

Figure 7. Simulation environment of an AT vehicle model in Matlab/Simulink

Figure 7 shows the AT vehicle model using Matlab/Simulink. In Figure 7, the gray-colored block is a powertrain model that includes an engine map, a torque converter model, and a transmission shifting algorithm from the test results, and the blue-colored block is a vehicle and road loads model that includes driving environment and vehicle information.
Based on the AT vehicle model in Figure 7, the AT model was replaced with the HMT model using AMESim. In Figure 8, the ‘AMESim co-Sim’ block is the HMT model using AMESim, and it is integrated in the vehicle model [7]. Because the HMT model was developed from a conceptual design, the HMT model and the AT model had different gear ratio ranges (AT gear ratio: 4.714, 2.341, 0.974; HMT gear ratio: 10~0.9). The HMT gear ratio was controlled manually. That is, the plate angle was controlled with a ‘revised transmission map’ from the Automatic Transmission map [8], and the clutches were controlled with commands of engagement and disengagement time.

The Figure 9 graphs show the comparison results between the AT vehicle model and the HMT vehicle model. Figure 9 (a) shows that the input values are the same as the vehicle models. In the simulation comparison in Section 2, the Throttle Position 100% test data are used. This is because a driver’s stable accelerator control is possible with full acceleration mode. However, in this section, Throttle Position 40% was used for describing the general input of a driver. Figure 9 (b) shows that the HMT Gear Ratio (red line) changed continuously and discretely. This is because the HSU caused the smooth change, and HMT discrete gears caused the stepped change at 2 seconds. From the control input, the HMT vehicle model showed better fuel efficiency and worse powertrain efficiency than the AT vehicle model. Therefore, the HMT vehicle model was developed to adapt a control algorithm.

5. Conclusions
In this paper, a heavy duty commercial vehicle model was developed and integrated with a Hydraulic-Mechanical Transmission (HMT) model that was in conceptual design stage. From the test results of a
vehicle with Automatic Transmission (AT), an AT vehicle model was developed and verified. An HMT model was developed based on a conceptual design. The AT vehicle model and the HMT model were integrated, and the integrated model was compared with the AT vehicle model to consider the feasibility and difference in performance.

In future work, the conceptual design will be tested. The specifications of the HMT will be confirmed, and the HMT model will be developed and tested. The HMT model will be verified based on the model test, and the model correlation will be increased.

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