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Noise and Vibration Analysis of Electric Oil Pump with Uneven Pitch Control for Gearbox in HEV and BEV

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Executive Summary

Acoustic noise and vibration are always main issues to overcome in electric oil pumps for gearboxes in HEV and BEV. This paper introduces a noise reduction control method of electric oil pump in wet-type dual clutch transmissions. This paper proposes an uneven pitch control for electric oil pumps. For the noise reduction of vane pumps, mechanical arrangements of uneven pitch vain angle are widely used. However, the tooth angle of gear-type pumps should be even mechanically. The proposed uneven pitch control provides similar effects of the mechanical uneven pitch arrangement by instantaneous motor torque controls of the electric oil pump which cannot have uneven pitch mechanically. The magnitude of motor torque for each pump tooth is determined by an uneven pitch formula which is widely used for mechanical vane pumps in previous study and patents. A formula for the shape of motor torque is proposed by analyzing pressure fluctuations of pump as a combination of trigonometric and exponential functions. The calibration factors for the magnitude and shape are adjusted by characteristics of pumps. The experimental results showed that noise reduction and dispersion effects of the proposed method.

1 Introduction

In order to improve the fuel efficiency of eco-friendly vehicle, parts that were previously driven by engine power or hydraulics are being converted to electric. Mechanical oil pumps that are directly connected to the engine and transmission are being re-placed with electric oil pumps driven by permanent magnet synchronous motors and controllers to improve the fuel efficiency of eco-friendly vehicles [1].

Unlike mechanical oil pumps that operate in proportion to the engine rotation speed, electric oil pumps that can operate independently of the engine require relatively high hydraulic pressure and flow rates even when the vehicle is stopped or driving at low speeds, making noise reduction design even more important.

The noise sources and noise composition of oil pumps are known, and the design technology to reduce them is a unique technology of the pump manufacturer, requiring a lot of design experience and know-how [2]. The main noise reduction design factors are the pump blade and tooth shape, clearance, and port shape, and by adjusting these shape variables, the internal flow can be improved, and the noise reduction effect can be obtained. The technology to reduce noise by changing these shape variables has a close in-fluence on pump performance, so there is a limit to reducing noise while maintaining hydraulic performance. In other words, increasing the clearance between the pump rotor and housing and the clearance between the pump teeth can reduce noise, but it will lower the pump's volumetric efficiency, which will lower the hydraulic performance.

A design technology that distributes the frequency band of generated noise by arranging pump vanes with uneven pitches to reduce noise in a specific frequency band and avoid resonant frequency bands was introduced for vane pumps. [1-3]

Compared to vane pumps, the gear type pump, which is widely used in automobile oil pumps due to its relatively low cost, generates hydraulic pressure by meshing two gears with the same clearance and rotating, so the gap between each tooth must be even, and uneven pitch application is not possible. Unlike mechanical oil pumps, electric oil pumps can apply uneven pitch control that can drive gear type pumps that cannot mechanically apply uneven pitches by actively varying the rotation speed and torque of the motor under conditions similar to those applied with uneven pitches.

This paper applies the uneven pitch control technique to electric oil pumps based on internal and external gear pumps that cannot mechanically apply uneven pitches, and analyzes the noise characteristics through experiments using an electric oil pump based on an external gear pump for a transmission to confirm its effectiveness.

2 Pump geometry and uneven pitch

2.1 Uneven pitch of vane type oil pump

Fig. 1 shows the general form of a binary vane pump. The vanes are inserted into grooves installed radially in the pump rotor and rotate inwardly in the housing. It is a generally known method to apply uneven pitches to reduce noise in fluid machines with multiple blades. In the vane pump of Fig. 1, the spacing between each blade can be arranged unevenly as in Equation (1).

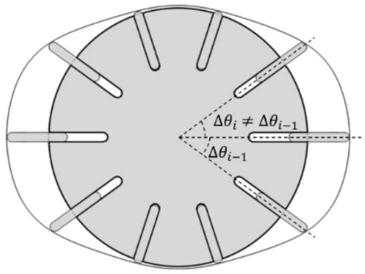


Fig. 1 Binary vane pump with uneven pitch

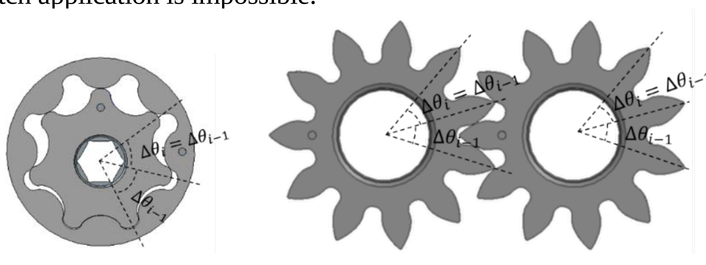
The process of arranging uneven pitches mainly uses a random number generator. If the positions of the wings are arranged randomly, there is a lack of design basis for the determined shape, so the effect must be confirmed through performance analysis. In addition, since the random number arrangement cannot be artificially adjusted, a lot of trial and error occurs in order to derive the optimal arrangement [3]. In order to solve the above problems, an uneven pitch generation function that prevents wing drift in a specific part while arranging the wings with uneven pitches and reduces the sound pressure peak, and generates the uneven pitch periodically to make it easy to predict and control the wing arrangement, as shown in Equation (2). [4]

$$\Delta\theta_i \neq \Delta\theta_{i-1} \quad (1)$$

$$\Delta\theta_n = \frac{2\pi}{N} + (-1)^n \times A_m \left\{ sin(P_1 \frac{2\pi}{N} n) \times cos(P_2 \frac{2\pi}{N} n) \right\} \quad (2)$$

2.2 Shape of gear type oil pump

Vane pumps are relatively complex in shape, making them difficult to miniaturize, and are expensive compared to gear-shaped pumps called internal and external gear pumps. Internal and external gear pumps, which are widely used in automobile oil pumps due to their relatively low cost, generate hydraulic pressure by meshing two gears with the same clearance as shown in Fig. 2, so the gap between each tooth must be even, and therefore, mechanically, uneven pitch application is impossible.



(a) (b)

Fig. 2 Structures of gear-type pumps: (a) internal gear type, (b) external gear type.

The main source of noise in internal and external gear oil pumps is hydraulic pulsation, which occurs in various forms depending on the design of the pump's teeth and port shape, as shown in Fig. 3. Gear-type pumps with a mechanically constant tooth-to-tooth gap have relatively uniform hydraulic pulsation depending on the teeth shape during rotation, as shown in Fig. 3. Therefore, the spectrum analysis results of the generated noise show that most of the noise is dominated by high-frequency components of the frequency of the product of the rotational speed and the size.

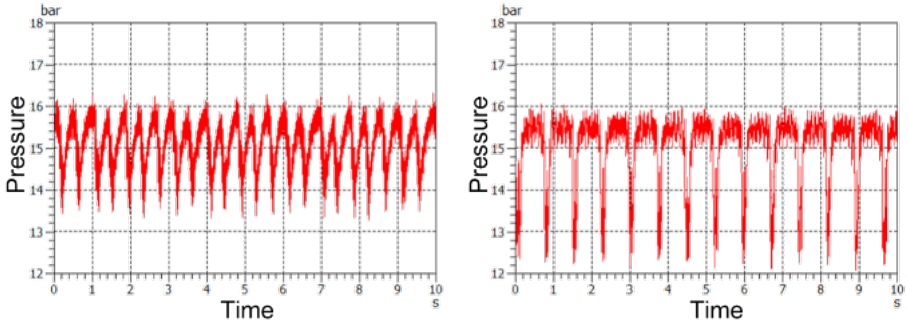


Figure 3. Analysis results of pressure fluctuation according to pump designs

3 Uneven pitch control

Unlike mechanical oil pumps that are directly connected to engines and transmissions, electric oil pumps driven by permanent magnet synchronous motors and controllers can actively control the rotation speed and torque of the motor. [5] We propose an uneven pitch simulation control technique that operates under conditions similar to applying uneven pitch by varying the rotation speed and torque of the motor to internal and external gear pumps that cannot mechanically apply uneven pitch.

Fig. 4 shows the general configuration of the control system of an electric oil pump. The system can be broadly divided into a pump, a motor, a controller, and various sensors.

The voltage reference value obtained through the speed controller and current controller based on the proportional integral (PI) controller is output to the three-phase inverter and controlled. The speed and current values measured by the rotor position sensor and the current sensor are fed back to control so that the error with the speed reference value becomes 0. The current of the motor is a value proportional to the torque, so the motor speed and torque can be instantaneously controlled through the two PI controllers. [6,7]

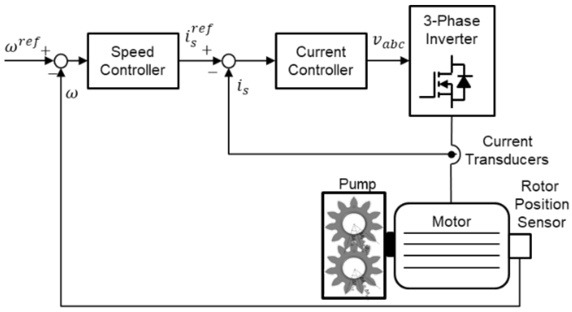


Fig. 4 Control diagram of electric oil pumps

By instantaneously increasing or decreasing the motor torque according to the position of each tooth that is mechanically evenly arranged, the same effect as applying an uneven pitch can be expected. The instantaneous shape of increase or decrease in the motor torque is presented as in Equation (3).

$$y = e^{-B_m x^2} \tag{3}$$

where $-1 \le x \le 1$

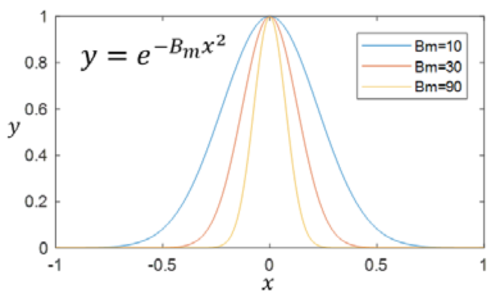


Fig. 5 The proposed shape of uneven pitch-simulated control

This is applied by adjusting the magnitude of B_m, which is the uneven pitch control shape factor in Equation (3), according to the hydraulic pulsation shape of the pump interpreted or measured as in Fig. 3. Fig. 5 shows the shape of uneven pitch control according to B_m.

In addition to the current command value determined by the speed controller, the current command value, i_{uneven}^{ref}, for uneven pitch simulation is additionally applied as shown in Fig. 3, and the motor torque is instantaneously and variably controlled according to the number of pump teeth and tooth positions. The magnitude of the uneven pitch simulation control command to be differentially applied to each tooth sequence is defined as in Equation (4) using Equation (2), which is known through prior research and patents, of the uneven pitch function.

$$I_{\Delta\theta_n} = (-1)^n \times A_m \left\{ sin(P_1 \frac{2\pi}{N} n) \times cos(P_2 \frac{2\pi}{N} n) \right\} \tag{4}$$

In order to maintain the pump hydraulic performance, the average of the control command is 0, and its maximum size needs to be experimentally determined without deteriorating the hydraulic performance of the pump. Finally, the uneven pitch simulation control current command applied to each tooth sequence is implemented as in Equation (5).

$$i_{uneven}^{ref} = I_{\Delta\theta n} \times e^{-B_m \left\{ mod\left(\theta, \frac{2\pi}{N}\right) - 1 \right\}^2} \tag{5}$$

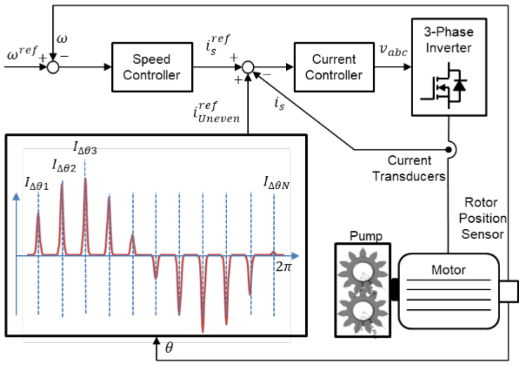


Fig. 6 The proposed uneven pitch-simulated control

4 Experimental Results

Table 1 shows the main characteristics of the electric oil pump used in the experiment, and Photo 1 shows the experimental environment, including the electric oil pump installed in the anechoic chamber and the location of the installed acoustic noise and vibration sensors. To measure airborne noise that is transmitted directly into the interior of the vehicle through gaps, a microphone (1/2 inch free-field type, PCB 377B02 amplifier 426E01, 50 mV/Pa) was installed at a distance of 30cm from the system under tests. To measure structure-borne noise that is transmitted into the interior of the vehicle through the vibration of the pump, a three-axis vibration acceleration sensor (PCB ICP HT356A33, 1.02 mV/(m/s^2)) was attached to the surface of the electric oil pump as shown in Photo 1. Automatic transmission oil (ATF SP-4M) was used, and the oil temperature was 60°C. A strain gauge type (MNEBEA NS30T) hydraulic pressure sensor was installed at the pump discharge section to measure the hydraulic pressure. The pump driving speed was tested at 1500 rpm (25 Hz), which is mainly required for low-speed driving and stopping, and 3000 rpm (50 Hz), which is required for high-speed driving.

Table 1. Specification of the tested system.

	Value	Unit
Pump type	External gear	-
Number of teeth	11	EA
Max Speed	3200	RPM
Max Hydraulic Power	4bar@16lpm	-
Max Electric Power	25A@12V	-

In the case of the pump, the impact of pump noise is smaller than that of vehicle driving noise and engine noise when driving at high speeds, which requires 3000 rpm of pump driving. Rather, the pump driving noise of 1500 rpm can be irritating to the driver's ears when driving at low speeds or stopping, so noise reduction is required even more.

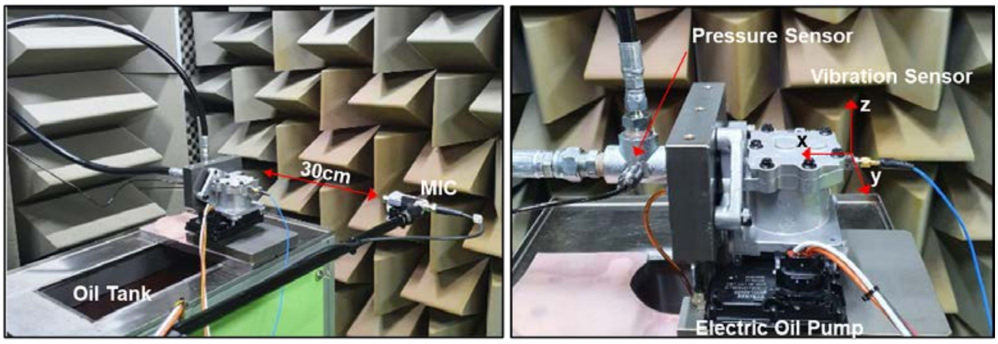


Photo. 1 Noise and vibration measurements in the anechoic room.

Fig. 7 shows the results of comparing the pump hydraulic performance and current consumption according to the application of general control and uneven pitch simulation control. It can be seen that when the maximum size of the uneven pitch simulation control command is increased to 40% of the rated current, the flow rate performance is reduced by about 10%. The experiment was conducted by setting the maximum size of the uneven pitch simulation control command to 20% of the rated current so that the change in hydraulic performance is maintained within 1%.

Fig. 8 shows the change in pump hydraulic pulsation according to the control method. The uneven pitch control shape factor B_m was determined to be 30 to have a similar pulsation width from the hydraulic pulsation measurement results and applied. The average value of the hydraulic pulsation is the same, but it changes depending on the tooth position when applying the uneven pitch simulation control, compared to the general control where it occurs evenly depending on the tooth position

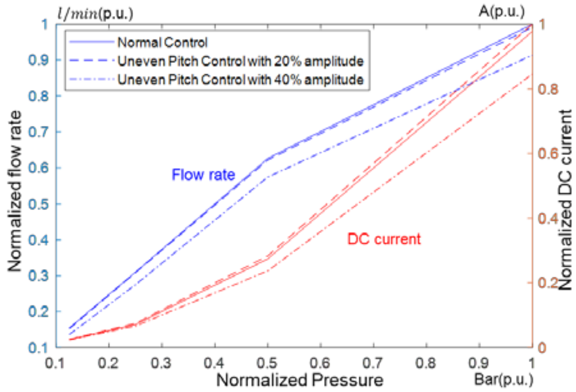


Fig. 7 Hydraulic performances according to the control methods.

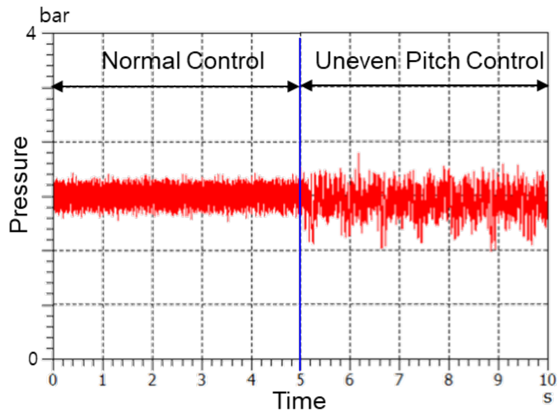


Fig. 8 Pressure fluctuation according to the control methods.

Fig. 9, 10 and 11 are the results of frequency spectrum analysis of measured noise and vibration. In the case of general control, the peak of the frequency band of 550 Hz and 825 Hz, which is the order component of 275 Hz, which is the combination frequency of the rotational speed (25 Hz) and the number of teeth is dominantly occurring.

When the uneven pitch simulation control is applied, it can be confirmed that the noise of the frequencies of 550 Hz and 825 Hz is reduced. In contrast, it was confirmed that the frequency components of 525 Hz, 575 Hz, 800 Hz, and 825 Hz, which are distributed at intervals of 25 Hz, which is the rotational frequency, based on 550 Hz and 825 Hz, tend to increase symmetrically and be distributed. This is because the size of the uneven pitch simulation control command corresponding to each tooth was set to one rotation cycle as in Equation (4), and it can be confirmed that this result has the same tendency as the results of a previous study that mechanically applied the uneven pitch as in Equation (2). [3,4]

In the vibration measurement results, it can be confirmed that it is symmetrically distributed based on 550 Hz, just like the noise measurement results. As summarized in Table 2, when the uneven pitch simulation control is applied, the maximum peak value of the noise is reduced by 4.5 dB (A), but the overall size of the entire noise is maintained at the same level. This shows that when applying the uneven pitch, rather than absolutely reducing the noise, it plays a role in dispersing the frequency components and changing them into a form of noise that is less irritating to the driver's ears.

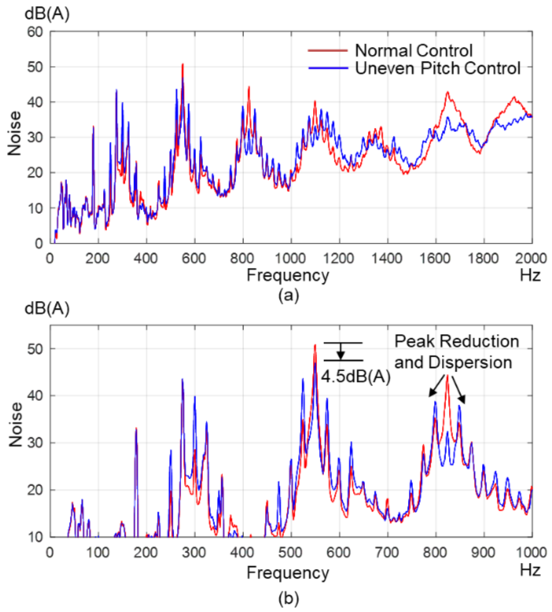


Fig. 4 Noise spectrum according to the control methods (1500rpm).

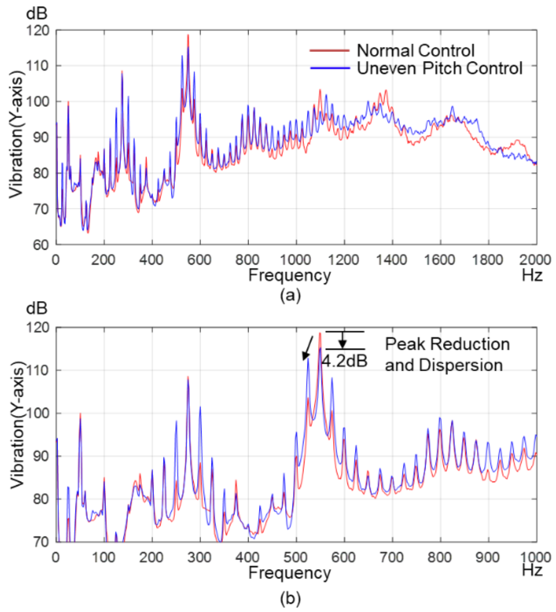


Fig. 5 Vibration spectrum according to the control methods (1500rpm)

Table 2 Noise and vibration measurement results at 1500rpm

	Noise [dB(A)]		Vibration [dB]	
	Overall	Max Peak	Overall	Max Peak
Normal Control	58.2	52.0	134.0	118.1
Uneven Pitch Control	58.0	47.5	133.5	113.9

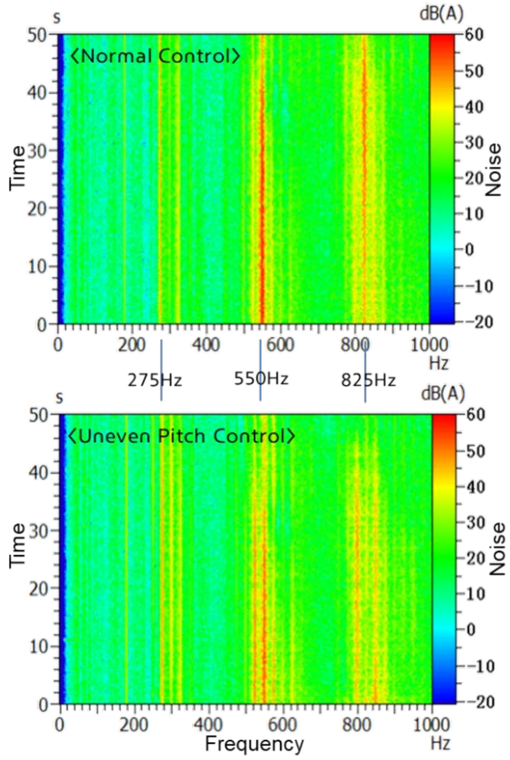


Fig. 11 Noise color map according to the control methods (1500rpm)

References

[1] J. W. Chang and D. Y. Yoon, “Sensorless Starting Method and Fuel Pressure Control of BLDC Motor for Fuel Pump of Vehicle,” Transactions of KSAE, Vol.21, No.2, pp.114-121, 2013.

[2] T. Son, “Analytical Study on Inner Flow and Structural Stiffness in Vane Type of Vacuum Pump,” Transactions of KSAE, Vol.25, No.2, pp.201-206, 2017.

[3] K.-Y. Lee, U.-H. Jung, J.-H. Kim, C.-H. Kim, Y.-S. Choi, J.-H. Ma, K.-H. Jeong and W.-J. Park, “Analysis of Noise Characteristic of Uneven Pitch Regenerative Blower,” Journal of Fluid Machinery, Vol.18, No.6, pp.71-75, 2015.

[4] J. W. Jang, Y.-T. Hwang, Y.-S. Choi, K.-Y. Lee, U.-H. Jeong and M. Seiji, “Random Pitch Impeller for Fuel Pump,” United States Patent, US8092186B2, 2012.

[5] S. Kim, W.-S. Choi, M. Kim, H. Kim and W. Lim, “Analysis of Fuel Economy of Mild Hybrid Vehicle by the Backward Simulation with Considering Power Loss of Oil Pump,” Transactions of KSAE, Vol.26, No.4, pp.533-539, 2018.

[6] C. Choi and W. Lee, “Improvement of Low Speed Sensorless Drive with Gain Attenuation and Delay Compensation of Measured Phase Voltage for Transmission Electric Oil Pump,” Transactions of KSAE, Vol.27, No.10, pp.819-826, 2019.

[7] C. Choi, J. Kang and W. Lee, “Improvement of Sensorless Drive with Varying Electric Motor Parameters for Transmission Electric Oil Pump at Lower Limit Temperature and Speed Region,” Transactions of KSAE, Vol.27, No.12, pp.949-955, 2019.

Presenter Biography

Chinchul Choi received the B.S., M.S. and Ph.D. degrees in control and instrumentation engineering from Changwon National University, Changwon, Korea, in 2006, 2008 and 2013, respectively. From 2014 to 2021, he worked at the software development team in Myunghwa, Seoul, as a senior research engineer. And from 2021 to 2023, he was with Hyundai MOBIS, Yongin, where he was a senior research engineer for developing control software of electric propulsion systems. Since 2023, he has been with Department of Control and Instrumentation Engineering, Changwon National University, Changwon, Korea, where he is currently an Assistant Professor. His research interests include the control, modeling, and fault diagnosis of automotive electric drive systems.