Model-Based Control Design and Experimental Validation of an Automated Manual Transmission

THESIS

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By

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Abstract

With the increasingly rigorous government regulations for fuel economy and exhaust emissions, automotive manufacturers are dedicated to developing vehicles which could have higher fuel economies, lower emissions and at the same time maintain drivability and customer acceptability. Under these demands, automotive manufactures have been developing advanced vehicle powertrains such as hybrid electric vehicle (HEV) and plug-in hybrid electric vehicles (PHEV). This thesis describes the methodology to model and control the linear actuation system of a ‘clutchless’ automated manual transmission (AMT) in a plug-in hybrid electric vehicle, which could improve the drivability and customer acceptability of PHEVs.

The ECOCAR 2 architecture adopts the belt coupling between engine and front electric motor, which utilizes the front electric motor to achieve speed matching between the engine and the transmission; so that the AMT in PHEV could realize ‘clutchless’ shifting. The AMT used in this thesis is a modified version of conventional manual transmission which utilizes two linear actuators to move the transmission shifting lever through two cables; therefore, new control method needs to be developed for this system. In order to obtain accurate, fast and robust gear shifting during AMT operation, the control system
was developed using model-based control theory; with adaptive control algorithm, as well as fault diagnosis.

This thesis presents the development of modeling procedure of AMT actuation system and an adaptive control algorithm of AMT gear shifting which could improve the gear shifting speed and fault mitigation. Using system equivalent modeling methodology and test data extraction, the equivalent model for AMT actuation system was developed. Based on the model, the linear actuator position controller was designed to meet the design specifications. In order to prevent and mitigate the transmission shifting fault, an adaptive controller was developed and tested; which could automatically retrieve the neutral position, renew the position values for all gears, provide shifting stage feedback to the supervisory controller and guarantee the completion of shifting. Coupling the above functions with system DFMEA, the AMT could realize stable and reliable gear shifting with reduced sensitivity towards the installation errors. The model, controller and algorithm were verified in the Hardware-in-the-loop devices and transmission test platform. The research described in this thesis shows the reliability and stability of using a model-based method to control AMT gear shifting, and indicates the importance of the adaptive gear shifting algorithm for AMT system.
Dedication

This thesis is dedicated to my parents and my wife for their selfless love and support.
Acknowledgments

I would like to thank my advisor Dr. Shawn Midlam-Mohler for his guidance and leadership through my Master’s degree research. I would also like to thank the Center for Automotive Research for providing me the needed devices and facilities to support my research. I would like to thank Dr. Giorgio Rizzoni for his great advice and support for Ohio State University ECOCAR 2 program.

I would also like to thank the entire Ohio State University ECOCAR 2 team for their efforts in the mechanical design, control development and electrical building of our team’s series-parallel plug-in hybrid electric vehicle. Design, building and testing the ECOCAR 2 need inestimable time and require team members to work together. Without the help and support from the rest of team member, my research could not be completed. Therefore, I would especially like to thank my teammates Gaurav Krishnaraj, Luke DeBruin, Matthew Yard, Andrew Michael Garcia, Matthew Organiscak, Katherine Bovee for their enthusiasm and help during my research period.

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1.1 The ECOCAR 2 Competition

1.1.1 The ECOCAR 2 Architecture

ECOCAR 2 is a three-year competition program sponsored by General Motors and the Department of Energy which aims to design, control and implement a hybrid powertrain in a 2013 Chevrolet Malibu. The main goals of ECOCAR 2 are reducing fuel consumption, decreasing the greenhouse gas emissions and the exhaust gas pollution, while keeping the conventional vehicles’ performance, drivability and consumer acceptability.

For Ohio State University ECOCAR 2 vehicle, a Parallel-Series Plug-in Hybrid Electric vehicle (PHEV) is selected and designed, which consists of a front electric motor, an inline four-cylinder E85 Honda engine, a six-speed Automated Manual Transmission coupled with engine and front motor, a rear electric motor connected with a single-speed gearbox and a battery package. The chosen and developed architecture of ECOCAR 2 is shown in the following Figure.
In the selected Parallel-Series Plug-in architecture, the 1.8L four cylinder Honda engine is directly connected to an 82kW peak power front electric motor. The electric machine and engine are connected to the six-speed automated manual transmission through the belt-coupling, which transmits the power to the front wheel axle by the embedded final drive differential. The rear powertrain also contains an 82 kW electric motor, which is connected with a single speed gearbox to output power and torque to rear wheel axle. Both of the two electric machines could get power from the A123 battery package and realize the regenerative braking. The series mode is achieved when the transmission is engaged in neutral position. Therefore, in order to realize the optimum fuel economy and switch between series mode and parallel mode, the precise control of the Automated Manual Transmission is very important.

1.1.2 Six-Speed Automated Manual Transmission Configuration
For the conventional manual transmission actuation system, ECOCAR 2 mechanical team uses linear actuators with two push-pull cables. The two cables connect the transmission shifter and linear actuators. Mechanical team also designed a sealed aluminum case for the linear actuators. In order to save room for other front powertrain components the actuators were installed underneath the vehicle chassis. The detailed actuation system configuration will be described in Chapter 3.

1.2 Motivation

Recently, developing vehicles with higher fuel economy and lower emissions has been the trend for most of the automotive companies, due to the increasing oil prices and strict government regulations. In addition to lower fuel consumption and lower emissions, the performance, drivability and customer acceptability should also be guaranteed.

Many automotive manufacturers are developing advanced technologies to make vehicles more efficient and more powerful. These techniques include optimizing the aerodynamics which could lower the drag coefficient, direct injection in internal combustion engines, turbocharged and supercharged engines. In addition to above technologies, advanced powertrain vehicles which combine alternative fuels are receiving increased attention. Several HEVs and EVs have appeared in automotive market such as GM Volt and Toyota Prius. Besides the development of hybrid electric vehicles, some manufacturers still focus on development of fuel cell vehicles.

Both of the recent automotive development trends and ECOCAR 2 competition required our vehicle to be as efficient as possible. Thus, choosing the appropriate transmission for
Ohio State ECOCAR 2 was very important to improve fuel economy, promote customer acceptability, maintain driving pleasure and win the competition.

1.3 Purpose of Thesis

The existing research on the automated manual transmission was confined to the shift quality, clutch torque transient for conventional vehicles, or AMT system configuration selection for hybrid powertrain, etc. None of these studies provided the modeling method for the AMT actuation system, which is the guidance for engineers to better understand the system. Besides, these researches did not describe the control strategy of the AMT; shift lever position control, sensitivity of the system to load change and temperature were not reported in the technical papers. More importantly, it is obvious that during the installation and tests, the preset neutral position and gear positions are likely to change. And it is undeniable that even during the operating period the vibration of the vehicle will influence the preset positions. None of these studies had considered the possibility of the change of preset transmission lever positions. Therefore, this study aims at performing research and experiments that could overcome the shortcomings mentioned above. The model and control requirements in this thesis includes: 1) Modeling the automated manual transmission actuation system; 2) Adaptive reference algorithm of gear positions during start-up mode; 3) Position control for actuation system of AMT which could realize reliable gear shifting during normal operation mode; 4) Reasonable transmission gear shifting time; 5) On-board fault diagnosis of the AMT actuation system; 6) Communication with supervisory controller for speed matching and fault report; 7) Shut-
down mode development; 8) Robust and stable gear shifting under various ambient temperature and different load; 9) No overshoot, fast and no steady state error transmission shifting lever step response under various ambient temperature and loads; and 10) Durability test of control algorithm.

1.4 Thesis Outline

In this thesis, the content presented covers the model-based control of an automated manual transmission and some work of AMT system design during the second year of ECOCAR 2 competition. Using the model-based experimental tests and system analysis, an equivalent electric motor model was developed and model-based control was implemented in SIMULINK. The adaptive reference algorithm, on-board fault diagnosis, communication with supervisory controller, force-based and position-based control were implemented in SIMULINK as well. The adaptive reference and gear shifting control were validated in both software and hardware. The performance of control and accuracy of model were discussed based on the results of various tests. The Figure 2 below indicates the development and experiment process of this project, while the Figure 3 shows the connection between different components.
Figure 2: AMT development and experimental test procedure
The outline of this thesis is as follows:

Chapter 2: Literature Review

Chapter 2 presents background information on popular contemporary types of vehicle transmissions and, the advantages and disadvantages of each transmission. In addition to the transmission types, different control strategies for AMT has also been discussed and compared, which clarifies the purpose of this specific research.

Chapter 3: Experimental Test Setup
Chapter 3 provides the description of devices and the experimental setup to determine system parameters. The transmission neutral position recovery algorithm and gear shifting control were tested and validated on ECOCAR team’s dSPACE Hardware-in-the-Loop (HIL) simulator. Besides the HIL test, the climate chamber test rig was used to examine the controller sensitivity to temperature changes and durability tests. To fully validate the controller robustness, the load test was also performed. For speed tolerance test and real environment test, the transmission spinning test rig was developed.

Chapter 4: Mathematical Model of AMT Actuation System

Chapter 4 describes the mathematical model of the overall transmission actuation system. It starts from the overview and introduction of the total system, which contains different components and their layouts. After the overview of the system, the mathematical model for different components are listed and explained. At the end the equivalent system model is presented and explained, which includes the calculation of various parameters, the ratio between components. This model is the basis for the position control.

Chapter 5: Start-Up Mode of AMT

Chapter 5 details the adaptive neutral position recovery strategy. Before the vehicle runs, the transmission control will adaptively acquire the different gear positions and put the shifting lever back to neutral.

Chapter 6: Normal Operation Mode of AMT System

Chapter 6 presents normal operation gear shifting control method, which contains position-based and force-based control methods. For position-based control, the position control strategy was adopted to realize the zero steady-state, no overshoot, and
disturbance rejection. The gear shifting strategy also contains shifting retrial logic. Additionally, the communication between GCM and supervisory controller is presented.

Chapter 7: On-Board Diagnosis and Shut-Down Mode

Chapter 7 discusses the on-board diagnosis procedure about the transmission linear actuator assembly which uses the DFMEA as the analysis and optimization tool. The DFMEA covers fault diagnosis on system components and transmission shifting algorithm. The failure detection of system components covers the warm-up stage, normal operation stage and park stage.

Chapter 8: System Model and Control Validation

Chapter 8 performs both the software and hardware validation for system model and actuation system position control. In addition to validations, the robustness and durability of controller under temperature test and load test are described.

Chapter 9: Conclusion and Future Work

Chapter 9 summarizes the work described in this thesis and provides list of opportunities for the future work.
Chapter 2: Literature Review

2.1 Introduction

In order to better understand an automated manual transmission in hybrid vehicle, this chapter provides the background information for AMT system architecture, AMT system modeling, AMT position control and system DFMEA.

2.2 Transmission Comparison and Selection

Many manufacturers have been developing many different types of transmissions to achieve the optimal powertrain operating efficiency. Various kinds of multiple gear transmissions are available in the market. The most common and most utilized transmissions are manual transmission, automatic transmission, continuous variable transmission (CVT) and Dual clutch transmission (DCT). In order to shed light on the advantages and disadvantages of different transmissions, the following sections have been presented with the description of four common transmissions that could be implemented in hybrid electric vehicles.

2.2.1 Manual Transmission

The manual transmission usually contains two parallel gear shafts; one is the torque and power input shaft, another is the output shaft which provides the torque and power to wheels. The following Figure shows the configuration of a typical manual transmission:
Generally, the average efficiency for manual transmission is around 96.2 percent for current products [2]. In order to show the exact efficiency range for each gear and in gear time, Table 1 is cited here from Kluger, M. and Long, D. [2].

<table>
<thead>
<tr>
<th>Gear</th>
<th>Time in Gear</th>
<th>Representative Efficiency</th>
<th>Current Production Efficiency Variation</th>
</tr>
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<tbody>
<tr>
<td>1</td>
<td>8%</td>
<td>93.5%</td>
<td>92-96%</td>
</tr>
<tr>
<td>2</td>
<td>10%</td>
<td>92.0%</td>
<td>92-97%</td>
</tr>
<tr>
<td>3</td>
<td>21%</td>
<td>94.0%</td>
<td>93-97%</td>
</tr>
<tr>
<td>4</td>
<td>20%</td>
<td>97.4%</td>
<td>93-99%</td>
</tr>
<tr>
<td>5</td>
<td>41%</td>
<td>93.8%</td>
<td>92-97%</td>
</tr>
</tbody>
</table>

Table 1: Five-speed manual transmission percent time in gear and efficiency in each gear [2]

As it is shown in Table 1, no matter which gear the transmission is in, the efficiency will always above 92%. However newer technologies may improve manual transmission’s efficiency by about 0.5% [2]. Considering only the efficiency of the transmission system,
the high efficiency of manual transmission makes it the ideal choice for ECOCAR 2 competition. However, since the customers favor the automatic transmission, the manual transmission was not used for ECOCAR2.

### 2.2.2 Continuously Variable Transmission

A continuously variable transmission (CVT) is a transmission which could gradually and smoothly change infinite gear ratios between maximum and minimum values. In contrast to most transmissions which have 5-8 discrete gear ratios, a CVT could change the gear ratio continuously so that engine operates at the most efficient region. Based on the characteristics of CVTs, CVTs operates the vehicle without jerks or torque holes [3]. The most common CVTs consist of two sets of conical pulleys which allow the radius of contact point to be changed [3]. In addition to the two sets of pulleys, there is a V-belt or V-chain which connects the two pulley sets. The diagram for a chain-driven CVT is shown below:

![Figure 5: A chain-driven CVT][1]
Despite the advantages of CVTs which allows the engine to operate at the efficient region, there are several minor disadvantages. The pulleys are often actuated by hydraulic power, which needs low flow and high pressure. This could consume a lot of energy during the operation of CVTs; the manufacturing cost of driving belts or chains of CVTs is much higher compared to conventional transmissions. These disadvantages make the overall efficiency much lower than that of the manual transmissions, which is shown in the following Table [2]:

<table>
<thead>
<tr>
<th></th>
<th>Low-Speed Ratio High-Input Torque</th>
<th>Mid-Speed Ratio Mid-Input Torque</th>
<th>High-Speed Ratio Low-Input Torque</th>
</tr>
</thead>
<tbody>
<tr>
<td>Low-Speed</td>
<td>84%</td>
<td>86%</td>
<td>77%</td>
</tr>
<tr>
<td>Mid-Speed</td>
<td>86%</td>
<td>89%</td>
<td>80%</td>
</tr>
<tr>
<td>High-Speed</td>
<td>83%</td>
<td>85%</td>
<td>76%</td>
</tr>
</tbody>
</table>

Table 2: Belt type CVT efficiency values at various operating conditions [2]

In conclusion, although the CVTs could make the engine operate at a better efficiency region, the overall system efficiency of the CVTs is lower than manual transmissions. For PHEVs like ECOCAR 2, most efficient engine running region is guaranteed by supervisory controller and hybrid function, which makes this kind of transmission less preferable for ECOCAR2 competition.

2.2.3 Automatic Transmissions

An automatic transmission can change the transmission gear automatically during vehicle operation providing ease of driving and free the driver from gear shifting. Automatic
transmission is mainly made of torque converter, planetary gear and gearbox. The diagram of an automatic transmission is shown below:

![Automatic transmission configuration](image)

Figure 6: Automatic transmission configuration [1]

The three components in different transmissions have many differences in the configurations and layouts based on the manufacturer [2]. For automatic transmissions, the efficiency and running time for each gear under combined local/highway cycle are summarized below [2]:

14
Table 3: Five-speed automatic transmission percent time in gear and efficiency in each gear [2]

Although automatic transmission could free the driver from shifting gears, the low efficiency in low gears and the pumping loss make this type of transmission not suitable for PHEVs’ application. Recently, automatic transmissions are being designed with more gears, which make results in the improvement in efficiency; however, the negative effects of increasing gears are the increasing weight and larger volume. The additional weight caused by transmission is unacceptable for PHEVs which already has a large and heavy battery package. Therefore, automatic transmissions are not the first choice for ECOCAR 2.

2.2.4 Dual Clutch Transmissions

It is well known that the dual clutch transmissions can provide comfortable shifting similar to the traditional automatic transmissions with faster gear shifting, optimized fuel efficiency and performance [4]. A dual-clutch transmission (DCT) is a type of automated manual transmission which generally has two separate clutches for odd and even gears.
Dual clutch transmissions could be regarded as two parallel automated manual transmissions in which odd gear shifting is controlled by one clutch, even gear shifting is controlled by another clutch. Figure 7 describes the configuration of a typical 6 speed dual clutch transmission.

![6-Speed Basic Design](Image)

Figure 7: Dual clutch transmission [1]

The two clutches of DCT allow the transmission to be in two gears at the time and prepare for next gear to be engaged. The shifting time is significantly reduced and fuel economy is better. The following Figure indicates the comparison of fuel economy between various automatic transmissions.
Figure 8: Fuel efficiency vs. engine displacement comparison for automatic transmissions [4]

From the Figure 8, we could clearly see that the more gears the automatic transmissions have, the better the fuel economy. For small engine, the efficiency of CVTs is better than that of 5-speed step ATs; however, for large engines, the fuel efficiency of CVTs is not better than that of step ATs. But for DCTs, the fuel economy is always much better than the efficiency of other automatic transmissions. For large displacement engine, DCT’s fuel economy is even better than manual transmissions. Therefore, DCT is the best transmission system described so far.

2.2.5 Automated Manual Transmission

An automated manual transmission consists of a dry clutch, a gearbox, and an embedded actuation system which is controlled by a dedicated electronic control unit to realize gear shifts according to driver’s command. This system removes the clutch pedal, but the
clutch itself is actuated by a device that can synchronize the engine speed and transmission speed, and at the same time complete the gear shifting in the shortest possible time. The advantages of AMT are the efficiency, light-weight and small size [22].

The below Figure 9 shows the fuel efficiency of various transmission types where the tradition 5 speed automatic transmission is the baseline. From the Figure 9, the best fuel economy transmission is AMT which is even better than manual transmission. In addition to the AMTs, the wet DCTs and dry DCTs have the highest efficiency as well as good comfort. However, if passengers’ comfort and driver’s preference are considered, AMT is not the first option for conventional vehicles which only contain one power source. Based on the combined factors, the DCTs are the best choice for conventional vehicles.

![Figure 9: Transmission comfort and fuel efficiency comparison [4].](image)

2.2.6 ECOCAR 2 Transmission Selection

For the first three transmissions, it is obvious that the manual transmission possesses the highest efficiency. Although CVT could make the overall powertrain efficiency higher
than automatic transmission, the efficiency of CVT itself is lower than that of an automatic transmission. As to the dual clutch transmission, the fuel efficiency compared to ATs could be up to 15% better [4]. However, the manual transmission has 15% to 25% higher fuel efficiency compared to ATs [4]. Since the ultimate goal for ECOCAR 2 competition is to obtain high efficiency, a transmission which possesses both the stable high efficiency of a manual transmission and the automatic operation was preferred. Under this condition, an automated manual transmission (AMT) would be the first choice, which combines advantages of an automatic and a manual transmission [5]. Based on the European transmission market for passenger cars, AMT is adopted by many manufacturers and its use has been on steady increase from 2000[6]. According to the variation of cars and designs, AMT shift times vary from 0.25 to 1.5 seconds [8]. However, combined with the hybrid powertrain of ECOCAR, shifting time could be reduced to 0.6 seconds achieved by the improvement of gear synchronization [9]. This shifting time could compete with the time taken for shifting by a driver operating a manual transmission vehicle [8]. Besides, the automated manual transmission is more stable than common manual transmission whose shifting performance is totally determined by driver [10]. The overall advantages of automated manual transmission make it the first choice for Ohio State vehicle.

2.3 Automated Manual Transmission Architecture and Control

2.3.1 Automated Manual Transmission Architecture
It is undeniable that manual transmissions are being replaced by various automatic transmissions around the world. Because of the low efficiency of automatic transmissions and the high efficiency of manual transmission, the automated manual transmission is being noticed by more and more engineers and manufacturers. However, vast majority of production automated manual transmissions utilize a dry clutch between engine and transmission to synchronize the speed. For a hybrid electric vehicle which has the hybrid powertrain, the gear speed matching is performed by the electric motor without using the clutch. This could greatly simplify the position control strategy of the AMT because the speed matching phase of clutch is not in control consideration and there is no concern for comfort and smoothness optimization of gear shifting. However, the precise and adaptive position control of manual transmission shifting must be guaranteed.

There are many published studies available on the Automated Manual Transmission modeling and control. G. Lucente et al. [7] developed a series of high-order equations to describe the clutch dynamics and hydraulic actuators in order to realize shifting comfort and optimize the clutch engagement time; Other studies also investigated the shift quality and clutch dynamics such as X-Y Song et al. [8] and H. Dong lee et al. [9]. However, literature focusing on clutchless automated manual transmission for a hybrid electric vehicle is rather sparse.

Breen, J. and Bower, G [10] proposed a clutchless automated manual transmission for all-wheel drive hybrid powertrain. In order to realize the speed synchronization of engine speed and transmission, and at the same time make the system comparatively easier to
control, they used electric motor to complete the clutchless shifting of AMT, which is shown in Figure 10.

![Figure 10: A clutchless AMT in the front hybrid AWD powertrain [10]](image)

The clutch is still adopted in their implementation; however, the function of clutch has been totally changed. The clutch here is only used to connect or disconnect the power between engine and transmission, the speed synchronization is completed by speed control of electric motor. The clutchless shifting of AMT could be used in series mode, parallel mode and electric mode. The validation data was provided to show that clutchless gear shifting could realize the gear shift according to the transmission gear command. The speed synchronization electric motor is close loop controlled by a PI controller, in which the speed input is calculated from the wheel speed multiplied by the differential gear ratio, gear ratio of gear and the gear ratio of belt drive. The linear actuator which
realizes the gear shift of transmission shift lever is simply controlled by proportional controller. There is no modeling of the actuation system, no accurate position control which could eliminate the overshoot and guarantee the shifting time. However, the most important idea for this paper is proposed configuration and integration for clutchless automated manual transmission in hybrid powertrain, which could give some insight for Ohio State vehicle. As it has been discussed, there is no adaptive neutral position retrieving control, gear position adaptive control, DFMEA and detailed modeling and accurate position control for the actuators. Thus, this implementation could not be totally adopted for OSU ECOCAR 2.

Erickson, B. et al. [5] developed an automatic shifting Manual Six speed transmission which contains the speed synchronization clutch to match the engine and transmission input shaft speed for parallel hybrid electric vehicles. Before gear shifting, the clutch should be disengaged, and then the gear is actuated to desired position, at the end the clutch was engaged to deliver the engine power and match the speed. For the gear shifting, there are two actuators; one is the pinion actuator, another one is lifter actuator. The design and integration of manual transmission and actuators in this paper are interesting and worth learning. It gives us a true integration method and operation strategy for AMT system. However, no model had been proposed for this mechanism, no sophisticated and robust position controller had been developed for the two actuators. Further, in this paper, the HEV from MTU does not utilize the front electric motor to speed match the engine speed and transmission speed, which is the major difference from OSU ECOCAR 2.
Link M. et al. [11] discussed the application possibilities of ASTs and expected the future development of ASTs for three different type vehicles. In this study, they stated that ASTs were evolved based on the AMTs but integrated the concepts of dual-clutch transmissions and starter-generator. They believed this concept could reduce or totally eliminate the interruption of traction during gear shifting. As to different type of vehicles, Link M. et al. proposed the corresponding calibration focus and design emphasis for a specific type vehicle, and they also stated the compromises that could be made for specific market segments. Besides the statement of different vehicle segment, this paper also gives suggestions for future AST development regarding the integration of manual transmission and actuators, actuating force, AST with wet clutch, etc. However, all advices are statement, no model has been proposed and no practical design and control has been presented in this study.

2.3.2 Automated Manual Transmission Position Control

Based on the OSU ECOCAR2 AMT development and architecture, a simple close-loop control could not meet our requirement. The distance between actuators and transmission shift lever is much longer than any mechanisms shown in above papers; the inertia of other components in the actuation system could not be ignored because the linear actuator itself is small and its power level is low; some time-variant factors were also contained in this system. All of these reasons require us to model the system and design a better and more precise controller. Fortunately, the detailed components were known after the complete disassembly of constitutive devices. The fundamental control target is the
PMDC motor inside the linear actuator; therefore, related control background for PMDC will be given in the following paragraphs.

As to the permanent magnetic electric motor control, the first concept in mind is the PID controller which could reduce overshoot and guarantee no steady-state error. There are a lot of related papers discusses the PID controller design. G. Ghang et al. [13] developed an intelligent PID controller for PM DC motor position control using evolutionary programming method. The general electric motor model is also described. Through minimizing the integrated-absolute error (IAE), adaptive fast evolutionary programming (AFEP) algorithm is used to find global optimization solution for PID controller parameters. This algorithm allows the PID controller to find the optimal proportional gain, integral gain and derivative gain during the step response process. Comparison among different algorithm PID controllers is shown below [13]:

Figure 11: step response of AFEP tuned controller vs. other controllers [13]
It is easy to see that AFEP algorithm PID controller possesses shorter settling time and less overshoot than the step response under Z-N controller and EP controller. It is undeniable that the performance of AFEP regulated PID controller is better than that of Z-N and conventional PID controller. However, even the adaptive evolutionary programming could not eliminate the overshoot and this controller is based on continuous time, which is not suitable for discrete time controller used in transmission control.

J. Tang [13] proposed on-line parameter adjustment for DC motor speed and position control using PID controller. Since the transmission shifting only needs position control, the following discussion will be focused on position control part. This paper used digital signal processing starter kit, and the development process of PID controller is based on discrete time. The designed digital PID controller was implemented through C31 DSK and examined by specific system [13]. For position control, if there is no integration part in the PID controller, the overshoot was eliminated and the step response transition was very good.

Figure 12: Actual motor position for PD controller [13]
The actual test result shows the expected position controller curve, which means PD controller really plays a good role. However, the transmission lever position generally works under vibration, which indicates that derivative gain might produce a lot of noise and make position control collapse. And without integration part, the noise could not be rejected. Therefore, only PD control could not meet the control requirement, a more stable and noise insensitive controller was necessary.

Faa-Jeng Lin developed a good method to eliminate the overshoot, remove steady-state error and set the rising time of step response. The purpose of Faa-Jeng Lin’s paper is to control PM synchronous motor using a digital signal processor. For my research, in which the power source is a PM DC motor, this method could not be directly adopted; however, the simplicity and controllability of DC motor makes the control and modeling of linear actuator easier than PM AC synchronous motor. After the field orientation of PM synchronous motor is analyzed, the DSP-based PM synchronous motor control is designed using field orientation. The basic control block diagram for PM synchronous motor servo control is shown below:

![Control block diagram for PM synchronous motor](image)

**Figure 13:** Control block diagram for PM synchronous motor [14]
In order to control the PM synchronous motor to realize servo drive, the transfer function of the real output position to input position command is developed below [14]:

\[
\frac{\theta_r(s)}{\theta_r(s)}|_{r_k(s)=0} = \frac{K_p K_b}{S^3 + (a + K_p K_b)S^2 + K_t K_b S + K_s K_t K_b} = \frac{h_1}{S + \mu_1} + \frac{h_2}{S + \mu_2} + \frac{h_3}{S + \mu_3}
\]  

(2.1)

Based on the relationship between the six parameters in equation (2.1), the unit-step response of the system transfer function could be expressed in the following format [14]:

\[
y(t) = \frac{h_1}{u_1} (1 - e^{-u_1 t_r}) + \frac{h_2}{u_2} (1 - e^{-u_2 t_r}) + \frac{h_3}{u_3} (1 - e^{-u_3 t_r})
\]  

(2.2)

After you determine the control specifications you need, the rise time of step response could be expressed as

\[
0.9 - \left[ \frac{h_1}{u_1} (1 - e^{-u_1 t_r}) + \frac{h_2}{u_2} (1 - e^{-u_2 t_r}) + \frac{h_3}{u_3} (1 - e^{-u_3 t_r}) \right] = 0
\]  

(2.3)

The energy distribution according to C.M.Liaw et al. and described in Faa-Jeng Lin’s research [21], could be expressed as \(e_1, e_2, e_3\). These three energy distribution parameters could be expressed by \(h_1, h_2, h_3, \mu_1, \mu_2\) and \(\mu_3\).

The nonlinear functions which could guarantee the control specifications could be constructed by \(h_1, h_2, h_3, \mu_1, \mu_2, \mu_3\) and \(e_1, e_2, e_3\). Once the nonlinear equations were solved, the parameters of the controller could be derived as follows [14], which could realize on-line and real-time parameters determination.

\[
K_p = \frac{u_1 + u_2 + u_3 - a}{K_t b}
\]  

(2.4)
Using the above control method, the test result is shown below:

\[ K_I = \frac{u_1u_2 + u_1u_3 + u_3u_2}{K_I b} \]  
\[ K_S = \frac{u_1u_2u_3}{K_I b c K_I} \]  

Using the above control method, the test result is shown below:

Figure 14: Motor step command response using real-time IP and torque forward control

Based on the above Figure, we could see that the performance of under real-time IP controller well enough for position control. The step response has no steady-state error, no overshoot, and the disturbance rejection is very fast. But the problem of this method is that the controller development is based on continuous time model and the control parameters determination is also based on continuous model. Because of the limited sampling frequency of Woodward ECU, modeling error and mathematic inaccuracy, this method could not work as expected. Therefore, this method could not be directly used for
control of transmission shifting actuator. Although it could not be adopted for current research, it is meaningful and helpful for future controller development.

In conclusion, for AMT actuator control, the modeling of the actuation system needs to be developed in discrete time; the controller development should be constructed in discrete time and so that it can be implemented on a digital controller; the controller needs to be insensitive to modeling error, noise interference and must not need very high sampling frequency.

2.3.3 Adaptive Control for Automated Manual Transmission

For AMT in automobiles, the gear positions of transmission generally do not stay the same because of the installation error, tolerance of actuation system linkages and abrasion during transmission operation [15]. It is very reasonable to put position adaptive control to AMT controller to guarantee the accurate transmission shifting. Z. Zhang and W. Chen designed the control algorithm to memorize and store the current actual position. After several gear shifting operations, the controller will process the stored actual position data; calculate the mean value for different gears as well as the standard deviation. If the difference between the reference value and the average new value is bigger than the threshold, the controller will update the gear position using the new calculated average value. The experimental result of the error greater than the threshold is shown below:
Figure 15: The experimental result of self-learning control for error greater than the threshold [15].

It could be clearly seen that once the error between calculated average value and reference value is greater than threshold value, the controller will adaptively adjust the target value for specific gear.

This paper proposed a good idea to adaptively control gear shifting during transmission operation. However, it is still difficult to know whether the transmission lever is completely engaged or not. In order to imitate driver’s gear shifting behavior, the controller needs to know whether the actuator reached the end of the gears. Therefore, a new algorithm needs to be developed for AMT actuation. The adaptive shifting algorithm must ensure that the transmission shifting lever is engaged after every operation, regardless of the installation error, abrasion error or tolerance error.

2.4 Automated Manual Transmission DFMEA
In reliable design and estimating complex design risks, Design Failure Mode and Effect Analysis (DFMEA) is a useful and appropriate tool in evaluating potential issue, determining and mitigating failures and risks of the overall system [16]. DFEMA is not new in US military and aerospace areas, but this methodology is comparatively new to automotive engineering. For hybrid electric vehicles or fuel cell vehicles which contain complicated powertrain systems, the DFMEA evaluation is very important and necessary [16]. The DFMEA consists of different levels of analysis, in which the highest level is system level. For this level, the analysis observes the overall system and interfaces. Under the system level, DFMEA could reach subsystems like engine and transmission, interfaces, and component levels [16]. The DFMEA family tree could be explained in the following Figure:

![DFMEA Family Tree](image)

Figure 16: DFMEA family [16].

The key point for DFMEA is the analysis level determination. Once the analysis aspects are confirmed, the development procedures and forms are almost the same. The DFMEA form contains function area which indicates the system, subsystem or components
function, potential failure modes area, potential cause area, determination of severity number, determination of occurrence number, determination of detection number and recommended action area. The meanings of these steps are easy to understand from their names.
Chapter 3: Experimental Test Setup

The facilities for the research and tests presented in this thesis are provided by The Center of Automotive Research at The Ohio State University. The actuator parameters’ test setup and actuation system parameters’ test were built by transmission team, which will be described in section 3.1; different mode control strategies were validated by Hardware-in-the-loop (HIL) from dSPACE. For durability and robustness, tests were performed in climate chamber; additionally, the setup of load test was performed by transmission team. The setup of transmission spinning test was built and designed by ECOCAR2 transmission and mechanical team, the detailed control and setup will be shown in Section 3.5. To control the transmission actuation system, a 128-pin Woodward general control module (GCM) was connected to the HIL, control laptop and coupled with several potentiometers. The transmission control algorithm was designed in MATLAB and SIMULINK which contains the built-in Woodward Motohawk blocks. These blocks allow the MATLAB code to communicate with GCM through CAN messages and Mototron USB connector.

3.1 Parameter Tests Rig

Parameters tests were used to determine the system parameters which are necessary to model the system. The testing platform and devices were constructed as shown below:
Figure 17: Power supply and data acquisition for parameter test rig

Figure 18: Test frame setup for linear actuator tests
<table>
<thead>
<tr>
<th>Device number</th>
<th>Device name</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Fluke Current clamp</td>
</tr>
<tr>
<td>2</td>
<td>NI DAQ 6009</td>
</tr>
<tr>
<td>3</td>
<td>Agilent DC power supply</td>
</tr>
<tr>
<td>4</td>
<td>Actuator flame</td>
</tr>
<tr>
<td>5</td>
<td>Linear actuator</td>
</tr>
<tr>
<td>6</td>
<td>Load bucket</td>
</tr>
</tbody>
</table>

Table 4: Linear actuator parameters determination test

The linear actuator was powered by Agilent DC power supply (4V to 13V). The data acquisition is performed by the DAQ 6009 National Instruments. The recorded data included linear speed provided by the potentiometer embedded in the linear actuator housing, the electric motor current provided by the clamp current sensor and the voltage provided by power supply. For the load tests, a certain weight was added to the buckets shown in Figure 18 in order to realize load variation. In transmission actuation system parameter tests, the cables were installed to actuators along with the aluminum case.

3.2 Hardware-in-the-loop Test Rig

For verification and validation of the warm-up stage neutral position recovery algorithm, force-based and position based gear shifting logic, the HIL test was implemented. The control strategy was implemented in the ECOCAR team’s DSPACE Hardware-in-the-loop (HIL) simulator, and the communication between EcoGCM and computer was
through MotoTune control connector and MotoHawk software. The entire setup of the HIL, actuator, GCM and computer is shown below.

Figure 19: HIL and dSPACE setup for transmission linear actuators
<table>
<thead>
<tr>
<th>Device number</th>
<th>Device name</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>SKF linear actuator</td>
</tr>
<tr>
<td>2</td>
<td>HIL control laptop</td>
</tr>
<tr>
<td>3</td>
<td>EcoGCM control laptop</td>
</tr>
<tr>
<td>4</td>
<td>Med-size Hardware-in-the-Loop device</td>
</tr>
<tr>
<td>5</td>
<td>Woodward 128-pin ECU (EcoGCM)</td>
</tr>
<tr>
<td>6</td>
<td>MotoTron control connector</td>
</tr>
<tr>
<td>7</td>
<td>GM M-32 6-speed manual transmission</td>
</tr>
<tr>
<td>8</td>
<td>Actuator shifting cable</td>
</tr>
<tr>
<td>9</td>
<td>Fluke current clamp</td>
</tr>
<tr>
<td>10</td>
<td>National Instrument DAQ 6009</td>
</tr>
</tbody>
</table>

Table 5: Transmission adaptive control algorithm test device setup

In the above setup, the ECU and MicroAuto box were powered by the HI, MircoAuto box and EcoGCM could communicate with each other in HIL environment. Because the supervisory controller is implemented in the MicroAuto box, the gear shifting commands would be sent to EcoGCM through MicroAuto box and EcoGCM would control the shifting of transmission.

In addition to the devices shown in the above Figures, the specific communication between different system components and data acquisition are shown below:
In Figure 20, a Woodward 128-pin ECU was used to communicate with sensors and actuators. The ECU was connected with control laptop through MotoTron USB cable connector and wiring harness. The position information was provided by actuator built-in potentiometers and potentiometer installed on transmission. The transmission control code written in the ECU is built in MATLAB and SIMULINK with MotoHawk development library. The MotoHawk development library could build the interface between control code and Woodward ECU [20]. Combined with Mototune software package from Woodward, the new control code could be flashed into ECU and transmission operating condition could also be monitored. The MotoHawk software could also provide the current feedback of actuators. Once the control code was
implemented to the ECU, the controller could manipulate the movement of actuators. The data acquisition of actuators’ position is through a National Instruments DAQ 6009.

3.3 Climate Chamber Test Rig

The climate chamber test setup was developed to test the sensitivity of the proposed warm-up stage adaptive reference logic and the normal operation stage gear shifting control to temperature changes. The system was subjected to different temperatures. In addition to the temperature sensitivity test, durability test of the controlled system was conducted in the climate chamber. The test procedure was automated so that the transmission could continuously operate for prolonged time durations inside the climate chamber.

3.3.1 Climate chamber Temperature Test Setup

In order to further validate the adaptive control algorithm, robustness of the controller, durability and reliability of system’s mechanical design; the transmission actuation system was put in the climate chamber. The selected tests temperature range is from -15 Celsius degree to 60 Celsius degree.
As it shown in the above Figure, the large beige box is the climate chamber which contains the transmission and transmission actuation system. Other devices used in temperature tests are the same devices explained in HIL test. The chosen temperature for robustness tests are -15C, -5C, 5C, 10C, 20C, 30C, 40C, 50C and 60C. The actuator position control, transmission warm-up stage adaptive reference logic and transmission gear shifting control are all tested in the climate chamber.

3.3.2 Durability Test

In addition to the temperature sensitivity test, the durability tests had to be conducted in order to guarantee transmission operation performance. The proposed test method was to generate a series of repeat gear shifting commands, which would be implemented on the transmission ECU. The repeating sequence stair contained incremental signal value from
one to seven, in which one represents shifting to first gear and seven represents shifting to reverse gear. The below Figure shows one cycle of repeating sequence stairs:

![Figure 22: Designed durability test sequence](image)

The above developed durability and reliability methodology was used for different temperatures described in temperature test section. The actuation system was commanded to shift to every gear repetitively for long time duration.

### 3.4 Transmission Load Test Rig

The above several sections described different tests setup for control and model validation, temperature sensitivity tests, and durability test. In order to fully validate the robustness of the control method, the load tests also needed to be performed to know whether the position controller was sensitive to load variations. For the load test, the key point is to add load without of increasing the system inertia. In order to satisfy the load variation requirement, the weight could not be added because this method would increase
system inertia and change system parameters. Therefore, a special method was applied in this test.

![Figure 23: Transmission position control step response load test](image)

From the above Figure, the load test setup is shown and presented. The electronic scale was used to indicate load on the transmission push-and-pull cable. The elastic cord connects the push-pull cable and scale. Because the elastic cord could extend very long, the movement of shifting lever will only slightly change the load added on the push-pull cable; therefore, the near constant load could be guaranteed.

### 3.5 Transmission Spinning Test Rig

The test procedure discussed above concerns the transmission actuation system robustness and logic reliability; however, the tests conducted were under static conditions, which mean that there was no speed and torque input to the transmission input shaft and there was no output speed and torque. For ECOCAR2 PHEV, which uses the front electric motor to match the transmission input speed and output speed, determining the speed tolerance for this manual transmission is very important for
transmission control. Therefore, a realistic test platform needed to be developed and controlled to test the transmission in real shifting environment. The below Figure indicates the configuration of the transmission speed tolerance test platform. Details about these components are described in the Appendix.

![Platform for the automated transmission shifting test with speed feedback control and frequency command control of electric motors.](image)

In this test platform, three inverters were connected to three motors. Each inverter controlled one motor independently. One of the motors was used to provide the speed and torque to transmission input shaft, the other two motors were connected to the half axles of the transmission output shaft. Therefore, three motors were used to mimic the real transmission operation speed during vehicle driving conditions. For the connection between three inverters and Personal Computer, the RJ-45 connector and Multi RS-485 Converter were adopted to enable the use of the Frenic Loader software control. The
below Figures indicate the wiring of the RS-485 Converter, configuration of shaft encoder, PG card interface and the connection of the above components. The shaft encoders were installed into motor shaft, the acquired shaft speed data was sent to the OPC-E1-PG interface card. For frequency control, the signals wires of encoders should be connected to XA and XB terminals, while for speed feedback control, the signal wires of encoders should be connected to YA and YB terminals.

Figure 25: Communication between transmission test platform components

After the hardware of the test platform have been completed, the control of the three induction motors was very important for transmission test. In order to get the speed
tolerance of transmission gear shifting, constant speed and accurate speed should be guaranteed. Based on the characteristics of the three inverters, only the factory Frenic Loader software could be used to control the three inverters and three motors. The Frenic loader motor control interface is shown in the Figure below.

![Frenic Loader control interface](image)

**Figure 26: Frenic Loader control interface**

Based on different connection ports of PG card and different function codes in Frenic Loader, both the speed feedback control and frequency command control which would allow slave motor to follow the speed of master motor were realized. In order to realize speed feedback control by personal computer, the following codes need to be used.
<table>
<thead>
<tr>
<th>Function code</th>
<th>Setting</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>H30</td>
<td>3</td>
<td>Frequency and Run command through RS-485</td>
</tr>
<tr>
<td>Y01 (inverter 1)</td>
<td>1</td>
<td>Station address for inverter 1</td>
</tr>
<tr>
<td>Y01 (inverter 2)</td>
<td>2</td>
<td>Station address for inverter 2</td>
</tr>
<tr>
<td>Y01 (inverter 3)</td>
<td>3</td>
<td>Station address for inverter 3</td>
</tr>
<tr>
<td>Y04</td>
<td>3</td>
<td>19200 Baud rate</td>
</tr>
<tr>
<td>Y07</td>
<td>2</td>
<td>2 stop bits</td>
</tr>
<tr>
<td>Y10</td>
<td>1</td>
<td>Frenic Loader protocol</td>
</tr>
</tbody>
</table>
| H30          | 1       | Communications Link Function (frequency
and Run command through RS-485)                        |
| F11          | 8.6     | Electronic Thermal Overload Protect Level           |
| F12          | 1       | Electronic Thermal Overload Protect time             |
| F42          | 3       | Control Mode Selection (V/f control with optional PG card) |
| P03          | 8.6     | Rated current value                                  |
| O09          | 20      | Feedback Input (Encoder pulse resolution)            |

Table 6: Function code for Frenic Loader speed feedback control.
Chapter 4: Mathematical Model Development of the AMT Actuation System

4.1 Motivation

In order to realize model-based control, an accurate system model is needed. Therefore, this chapter mainly describes the modeling details of the equivalent transmission actuation system.

4.2 Background and Overview of the AMT System

4.2.1 AMT Shifting Background

In order to fully understand how to automate the manual transmission and change it to automated manual transmission, the shifting process and principle of manual transmission should be completely understood. When shifting a manual transmission, the first step done by driver is to depress the clutch pedal. This action unloads the torque added on the transmission and permits the engaged gear to be disengaged. Once the current gear is disengaged, the driver needs to move the shift lever to the desired gear in order to match the input shaft speed and output shaft speed for certain gear. At this time, the friction synchronizers are utilized until the certain speed ratio matched and the shift dogs meshed. At the end, the driver releases the clutch pedal and the clutch synchronizes the speed of engine and the speed of transmission input shaft [9]. In this process, the
clutch cannot be omitted because, it plays an important role in engine speed and transmission input shaft speed synchronization.

The first problem that needed to be solved was the actuation of gear shifts. The most viable and convenient way to automate the manual transmission was to actuate the synchronizer and cut off the power while shifting the transmission. In this way, the transmission would be operated in the same manner as with a driver. For ECOCAR2 vehicle, the speed match between engine and transmission input shaft is accomplished by the electric motor. The electric motor was controlled to cut off the engine power during shifting and match the transmission input speed after the shift completed. Thus there was no need to consider the disconnection of engine load for this manual transmission automation. The Figure 27 shown below is the concept diagram of the proposed automated manual transmission system. The clutch was not used for the speed synchronization. However, it was used just for disconnection and connection of engine power under different operating modes.

4.2.2 AMT Actuation System Structure
The proposed automated manual transmission design contains a six-speed manual transmission, electronic actuators, an electromagnetic clutch as well as a silent chain. The selected components were the Goodyear Eagle Pd belt system, the Eaton Airflex electromagnetic clutch, the M32 six-speed transmission from General Motors, and the SKF CAHB-10 linear actuators. As it shown in Figure 27, the electric motor uses the belt to match the speed between engine and transmission input shaft. The clutch here was just used for disconnection when the vehicle is in electric mode in which the engine does not supply power. The actuators were connected with transmission shift lever with wire cable, which were not shown in the CAD Figure. With the wire cable connection, the actuators could be installed at a different location instead of on the top of the
transmission. Installation and fixation for these components required the housing which could hold the linear actuators and the cable connected with transmission shift lever. This particular actuators and cable frame was built by 1/8 inch 6061-T6 aluminum sheet metal [11], which is shown in Figure 28 [11]. One end of cable was attached to linear actuator box to get force from actuator, another end was attached to the designed cable frame to deliver the force to transmission shifting lever, which is shown in Figure 29 [11].

Figure 28: Actuator housing assembly [11].
After the real system design was discussed and system components were analyzed, the simplified structure Figure of the linear actuator system has been shown below. This specific system contains PM-DC brushed electric motor as power source. In addition to the electric motor, this system also contains gear trains, ACME lead screw drive, extension tube, actuator frame lever, cable, L-shaped lever and transmission shift lever. The main purpose of the gear trains was to increase the torque, which would amplify the output torque of the electric motor.
The ACME lead screw was used here to convert the gear rotation into linear movement, and also to increase the output force of the tube. The goal of actuator frame lever was to increase the velocity of transmission shaft by choosing the appropriate lever ratio. In the following section, the specific modeling process for this system will be described.

4.3. Mathematic Models of Different Components

4.3.1 Permanent Magnetic DC Motor Dynamics
Figure 31: Schematic diagram of the PM DC motor

The schematic circuit diagram for PM DC motor is shown in Figure 31. The armature of the PM DC motor could be represented by an electric circuit which contains the resistance $R_a$, an inductance $L_a$, and a voltage source $e_b$. $e_b$ represents the back electromotive force (back-emf), which is produced by the rotation of the rotor in the magnetic field produced by permanent magnet [17].

According to Farid and Benjamin [17], it is appropriate to make the assumption that the torque produced by the motor is proportional to the armature current since magnetic field is produced by permanent magnet and the air-gap flux in PM motor is constant. Thus, the motor torque equation is described in the following equation:

$$T_m(t) = K_t i_a(t)$$ (4.1)

Beginning in the input voltage $e_a(t)$, which is the control input for the whole system, the equations for the motor circuit are [17]:
\[ L_a \frac{di_a(t)}{dt} = e_a(t) - R_a i_a(t) - e_b(t) \quad (4.2) \]

\[ e_b(t) = K_b \frac{d\theta_m(t)}{dt} = K_b \omega_m(t) \quad (4.3) \]

\[ T_m(t) = K_t i_a(t) \quad (4.4) \]

\[ J_m \frac{d^2\theta_m(t)}{dt^2} = T_m(t) - T_L(t) - B_m \frac{d\theta_m(t)}{dt} \quad (4.5) \]

Where:

- \( e_a(t) \) is the applied voltage for armature;
- \( L_a \) is the armature inductance;
- \( R_a \) is the armature resistance;
- \( i_a(t) \) is the armature current;
- \( e_b(t) \) is the back emf (V);
- \( K_b \) is the back electromotive force (V/rad);
- \( \omega_m(t) \) is the rotor angular speed (rad/sec);
- \( J_m \) is the electric motor inertia (kg.m\(^2\));
- \( T_m(t) \) is the motor torque (N.m);
- \( T_L(t) \) is the load torque (N.m);
- \( B_m \) is the viscous-friction coefficient (N.m/rad/sec);
- \( \theta_m(t) \) is the rotor displacement (rad);
- \( K_t \) is the torque constant (N-m/A).

From the above equations, the state variables of the system could be defined as \( i_a(t) \), \( \omega_m(t) \), and the displacement \( \theta_m(t) \). Through some arrangement and manipulations of the above equations, the state-variable equations of the PM DC motor could be written in vector-matrix form [1]:

\[
\begin{bmatrix}
\frac{di_a(t)}{dt} \\
\frac{d\omega_m(t)}{dt} \\
\frac{d\theta_m(t)}{dt}
\end{bmatrix} =
\begin{bmatrix}
\frac{-R_a}{L_a} & -\frac{K_b}{L_a} & 0 \\
K_t & \frac{-B_m}{J_m} & 0 \\
\frac{1}{J_m} & 0 & 0
\end{bmatrix}
\begin{bmatrix}
i_a(t) \\
\omega_m(t) \\
\theta_m(t)
\end{bmatrix} +
\begin{bmatrix}
\frac{1}{L_a} \\
0 \\
0
\end{bmatrix} e_a(t) +
\begin{bmatrix}
0 \\
-\frac{1}{J_m} \\
0
\end{bmatrix} T_L(t) \quad (4.6)
\]
The transfer function between the motor displacement and the input voltage could be derived by setting the load torque to be zero.

\[
\frac{\theta_m(s)}{E_a(s)} = \frac{K_i}{L_a J_m s^3 + (R_a J_m + B_m L_a) s^2 + (K_b K_i + R_a B_m) s}
\] (4.7)

Relation between \(K_i\) and \(K_b\):

Even though the torque constant \(K_i\) and back-emf constant \(K_b\) are two separate parameters, their values are closely related to each other.

If the \(K_i\) and \(K_b\) are both in SI units, the following relationship between these two parameters can be derived [17]:

\[
K_b(V/\text{rad/sec}) = K_i((N.m)/A)
\] (4.8)

Therefore, it is obvious that the values of \(K_b\) and \(K_i\) are the same in SI units, which makes the actuator test easier. For the actuator parameter tests, we only need to test the back-emf constant, then the torque constant could be calculated by the above relationship.

4.3.2 Gear Train Dynamics

![Figure 32: Gear trains of linear actuator diagram](image)

Figure 32: Gear trains of linear actuator diagram
Based on the characteristics of the gear train, the relationship between the torques of different gears could be expressed as below:

\[ T_1 = \frac{N_1}{N_2} T_2, \quad T_3 = \frac{N_3}{N_4} T_4 \]  (4.24)

The relationship between the angles, angular speed of gears and the angular acceleration is shown in the following equations:

\[ \theta_1 = \frac{N_2}{N_1} \theta_2, \quad \theta_3 = \frac{N_4}{N_3} \theta_4 \]

\[ \omega_1 = \frac{N_2}{N_1} \omega_2, \quad \omega_3 = \frac{N_4}{N_3} \omega_4 \]

\[ \alpha_1 = \frac{N_2}{N_1} \alpha_2, \quad \alpha_3 = \frac{N_4}{N_3} \alpha_4 \]  (4.25)

4.3.3 ACME Lead Screw Drive Mechanisms

Figure 33: Lead screw drive diagram
In this system, the lead screw is directly connected with the 4th gear, thus the inertia of lead screw is part of the total inertia for 4th gear. The dynamics of lead screw could be described in the following equations:

\[ T_{in}(t) = J_{ls} \frac{d^2 \theta_{ls}(t)}{dt^2} + B_{ls} \frac{d\theta_{ls}(t)}{dt} + F_{ls} \omega_{ls} + T_{out}(t) \]  

(4.9)

Where:

- \( J_{ls} \) is total equivalent inertia of lead screw system;
- \( B_{ls} \) is the viscous coefficient of lead screw system;
- \( F_{ls} \) is the static friction produced by lead screw system;
- \( T_{in}(t) \) represents the input torque;
- \( T_{out}(t) \) represents the output torque.

Compared to the inertia of the lead screw itself, the equivalent inertia of the weight on the lead screw bar is more important.

For the weight on the lead screw bar, the equivalent inertia could be expressed in the following equation [17]:

\[ J_{eq} = \frac{W \times \left(\frac{1}{2\pi}\right)^2 \times S^2}{\eta^2} \]  

(4.10)

Where:

- \( W \) is the weight on the lead screw bar;
- \( S \) is the screw lead;
- \( \eta \) is the efficiency of the lead screw bar;

And for the force applied on the lead screw bar, we could transfer the load force to load torque for the electric motor.

\[ T_{eq} = F_T \frac{L}{2\pi e} \]  

(4.11)
Where:

\( F_T \) is the total load force on the lead screw bar; \( L \) is the screw lead; \( \eta \) is the efficiency of lead screw.

As to the total force calculation, \( F_T \) could be written in the following format [4]:

\[
F_T = F_A + F_E + F_F
\]  
\( (4.12) \)

Where: \( F_T \) is the total force; \( F_A \) refers to the acceleration force; \( F_E \) represents the external force; \( F_F \) is defined to be the friction force;

For the acceleration force, it can be calculated by the following equation [4]:

\[
F_A = \frac{W}{g} \cdot \frac{a}{12}
\]  
\( (4.13) \)

Where: \( W \) is the total weight to accelerate (kg); \( a \) is the linear acceleration (m/sec\(^2\)); \( g \) refers to the acceleration from gravity (m/sec\(^2\)).

External force is the force which is needed to produce certain amount of work; friction force is the force needed to overcome the system internal resistance.

4.3.3.1 Inertia of Lead Screw

Based on the mass and radius of the lead screw, the inertia of the lead screw bar could be calculated with the following equation:

\[
J_{\text{leadscrew}} = \frac{mR^2}{2}
\]  
\( (4.14) \)

Another form of calculating the inertia of the lead screw bar could be related to the material density and length of screw bar, which is shown below.
Based on the density of materials, length of the lead screw bar and the radius, the following equation could be used for inertia calculation:

\[ J_{\text{leadscrew}} = \frac{\pi L \rho R^4}{2} \]  \hspace{1cm} (4.15)

Where:

\( L \) is the lead screw bar length; \( \rho \) is the steel material density; \( R \) is the radius of the lead screw bar.

4.3.4 Linear Actuator Box Lever Equivalent Inertia and Dynamics

![Figure 34: Actuator frame lever diagram](image)

For this drive line, the lever is mounted on the actuator frame, and the pivot is not on the mass center. The connectors of actuator and cable could be seen as a part of the lever and do not need separate calculations for inertia. Then, the equivalent mass of this system for the actuator tube could be calculated.

Beginning with the input force from linear actuator and the load force from cable, the equation of lever motion could be derived as shown below:
For the actuation system, the torque could be replaced by force of actuator tube and cable. Therefore, the equation will be shown in below format:

\[
J_{\text{lever}} \frac{d^2 \theta_{\text{lever}}(t)}{dt^2} = T_{\text{actuator}}(t) - B_{\text{lever}} \frac{d \theta_{\text{lever}}(t)}{dt} - \frac{F_{\text{cl}}}{|\omega_1|} \omega_1 - T_{\text{cable}}(t) \quad (4.16)
\]

For the actuation system, the torque could be replaced by force of actuator tube and cable. Therefore, the equation will be shown in below format:

\[
J_{\text{lever}} \frac{d^2 \theta_{\text{lever}}(t)}{dt^2} = F_{\text{actuator}}(t)L_1 - B_{\text{lever}} \frac{d \theta_{\text{lever}}(t)}{dt} - \frac{F_{\text{cl}}}{|\omega_1|} \omega_1 - F_{\text{cable}}(t)(L_2 + L_1) \quad (4.17)
\]

Where:

- \(J_{\text{lever}}\) is the housing lever inertia;
- \(F_{\text{actuator}}(t)\) represents the force produced by the actuator tube;
- \(F_{\text{cl}}\) is the static friction generated from the motion of the lever;
- \(F_{\text{cable}}(t)\) is the load force from cable;
- \(L_1\) is the distance between actuator connection point and lever pivot;
- \(L_2 + L_1\) denotes the length between cable connection point and lever pivot.

4.3.4.1 Equivalent Mass and Inertia of Frame Lever

For the lever rotation not about the center of mass, the parallel axis theorem could be used here:

\[
J_{\text{lever}} = \frac{1}{12} ML^2 + M \times d1^2 \quad (4.18)
\]

Where:

- \(J_{\text{lever}}\) is defined as the lever inertia about the rotation point;
- \(L\) is the length of the lever;
- \(M\) is the mass of the lever;
- \(d1\) is the distance between mass center and rotation center.
Based on the configuration of Figure 34, the equivalent mass of lever converted to actuator tube end could be expressed in below format:

\[ m_{eq, \text{lever}} = \frac{J_{\text{lever}}}{L_1^2} \]  \hfill (4.19)

Where:

\( L_1 \) is distance between the rotation center and actuator tube connection point;

### 4.3.4.2 Equivalent Mass of Cable and Bolt

For the actuator drive line system, the mass of cable could not be neglected and as the same time, in this paper, the mass of bolt and shim are also considered. Therefore, their equivalent mass converted to actuator tube part (lead screw weight) could be expressed below:

\[ m_{eq, \text{cable}} = \frac{m_{\text{cable}}L_2^2}{L_1^2}, \quad m_{eq, \text{bolt}} = \frac{m_{\text{bolt}}L_2^2}{L_1^2} \]  \hfill (4.20)

Where:

\( L_1 \) is distance between the rotation center and actuator tube connection point; \( L_2 \) is the distance between the rotation center and cable connection point; \( m_{\text{cable}} \) is the mass of the cable; \( m_{\text{bolt}} \) is the mass of bolt.

### 4.3.5 Transmission Shift Lever Equivalent Inertia

When considering the equivalent inertia of the transmission lever, it is necessary to separate the equivalent inertia into two parts. One condition is the axial rotation of the transmission lever, and another condition is the linear movement of the transmission lever.
4.3.5.1 Transmission Lever Rotation Equivalent Inertia

The transmission lever rotation inertia could be calculated by the AutoCAD or SolidWorks once the structure of the transmission lever is determined. Therefore, based on the calculated inertia, the equivalent inertia could be described below:

\[
J_{eq_{\text{trans\_inertia}}} = \left(\frac{L_2}{L_1}\right)^2 \frac{J_{\text{trans\_lever}}}{r_{\text{trans\_lever}}} \left(\frac{1}{2\pi}\right)^2 \times S^2 \left(\frac{N_1N_3}{N_2N_4}\right)^2
\]  

(4.29)

Where:

\(L_2\) and \(L_1\) are shown in Figure 34; \(J_{\text{trans\_lever}}\) is the transmission lever inertia; \(S\) is the lead of ACME screw drive; \(r_{\text{trans\_lever}}\) is the distance between the cable connection point and transmission lever axis; \(N_1, N_2, N_3, N_4\) are the teeth of the four gears.

Figure 35: Inertia and mass of the transmission lever

4.3.5.2 Transmission Mass Linear Movement Equivalent Inertia
The second movement method of the transmission lever is the linear motion to realize the gear shifting which is actuated by cable through L-shaped lever. The moving mechanism is shown in Figure 36, and shift lever equivalent inertia is also described by Eq. (4.30).

![Diagram of transmission lever motion mechanism](image)

Figure 36: Transmission lever motion mechanism

\[
J_{eq\_trans\_linear} = \left( \frac{L_2 + L_1}{L_1} \right)^2 \frac{r_1^2}{r_2^2} m_{trans\_lever} \left( \frac{1}{2\pi} \right)^2 \times S^2 \left( \frac{N_1 N_2}{N_2 N_4} \right)^2 \left( \frac{1}{\eta_{\text{cable}}} \right)^2 \left( \frac{1}{\eta_{\text{actuator}} \eta_{\text{gear}}} \right)^2
\]

(4.30)

Where:

- \(L_2\) and \(L_1\) are shown in Figure 34; \(m_{\text{trans\_lever}}\) is the transmission lever mass; \(S\) is the lead of ACME screw drive; \(r_2\) is the distance between the transmission lever and transmission lever fix point; \(r_1\) is the distance between the cable connection point and transmission lever fix point; \(N_1, N_2, N_3, N_4\) are the teeth of the four gears.

4.3.6 Actuation System Dynamic Equations
From the above dynamic analysis for different components, the transmission actuation system’s dynamics could be described with the following equations:

\[ L_a \frac{d^2 i_a(t)}{dt^2} = e_a(t) - R_a i_a(t) - e_b(t) \]  
(4.2)

\[ e_b(t) = K_b \frac{d\theta_m(t)}{dt} = K_b \omega_m(t) \]  
(4.3)

\[ T_m(t) = K_t i_a(t) \]  
(4.4)

\[ J_{eq_{total}} \frac{d^2 \theta_m(t)}{dt^2} = T_m(t) - F_{eq_{total}}(t) - B_{eq_{total}} \frac{d\theta_m(t)}{dt} - T_i(t) \]  
(4.31)

The value determination for the parameters in the above equations will be explained in the following paragraphs.

4.3.6.1 Total Equivalent System Inertia

Total equivalent inertia for actuation 1 system

In conclusion of the above statement, the total equivalent inertia converted to the motor end for actuator 1 system could be expressed in the following equation:

\[ J_{eq_{total}} = J_{motor} + J_{gear1} + \left( \frac{N_1}{N_2} \right)^2 J_{gear23} + \left( \frac{N_3}{N_2N_4} \right)^2 \left[ J_{gear4} + J_{leadscrew} + (M_{nut} + M_{tube} + M_{eq_{lever}} + M_{eq_{boil}}) \left( \frac{S}{2\pi} \right)^2 \left( \frac{1}{\eta_{ac}} \right)^2 \right] + J_{eq_{trans\_linear}} + J_{eq_{cable}} \]  
(4.32)

Where:

\[ J_{eq_{trans\_linear}} = \left( \frac{L_2 + L_1}{L_1} \right)^2 \frac{r_2^2}{r_1^2} m_{trans\_leaver} \left( \frac{1}{2\pi} \right)^2 \times S^2 \left( \frac{N_1}{N_2} \right)^2 \left( \frac{1}{\eta_{cable}} \right)^2 \left( \frac{1}{\eta_{head\_motor}} \right)^2 \]

\[ J_{eq_{cable}} = \left( \frac{L_2 + L_1}{L_1} \right)^2 \frac{r_2^2}{r_1^2} m_{cable} \left( \frac{1}{2\pi} \right)^2 \times S^2 \left( \frac{N_1}{N_2} \right)^2 \left( \frac{1}{\eta_{cable}} \right)^2 \left( \frac{1}{\eta_{head\_motor}} \right)^2 \]

Total equivalent inertia for actuator 2 system
In actuation 2 system, the actuator is connected to push and pull cable through the frame lever just like actuator 1. Therefore, the calculation of the second actuation system is similar to that of actuator 1.

\[
J_{eq\text{total}} = J_{motor} + J_{gear1} + \left( \frac{N_1}{N_2} \right)^2 J_{gear23} + \left( \frac{N_1N_3}{N_2N_4} \right)^2 J_{gear4} + J_{leadscrew} + (M_{nut} + M_{tube}) \left( \frac{S}{2\pi} \right)^2 \left( \frac{1}{\eta_{\text{lead}}} \right)^2 + J_{eq\text{trans2}} + J_{eq\text{cable2}}
\]

(4.33)

Where:

\[
J_{eq\text{trans2}} = \frac{J_{eq\text{trans}}}{r^2} \left( \frac{1}{2\pi} \right)^2 S^2 \left( \frac{N_1N_3}{N_2N_4} \right)^2 \left( \frac{1}{\eta_{\text{cable}}} \right)^2 \left( \frac{1}{\eta_{\text{lead}}\eta_{\text{gear}}} \right)^2
\]

\[
J_{eq\text{cable2}} = m_{\text{cable}} \left( \frac{1}{2\pi} \right)^2 \times S^2 \left( \frac{N_1N_3}{N_2N_4} \right)^2 \left( \frac{1}{\eta_{\text{cable}}} \right)^2 \left( \frac{1}{\eta_{\text{lead}}\eta_{\text{gear}}} \right)^2
\]

\[J_{motor}\] is the inertia of the electric motor rotor; \[J_{gear1}\] is the inertia of the first gear; \[J_{gear23}\] is the inertia of the second and third gear; \[J_{gear4}\] is the inertia of the fourth gear; \[J_{leadscrew}\] is the inertia of lead screw rod; \[W\] is the total weight applied on the lead screw rod; \[S\] is the lead of the lead screw bar; \[N_1, N_2, N_3, N_4\] are the teeth of the four gears; \[M_{nut}\] is the mass of nut; \[M_{tube}\] is the mass of the tube in actuator; \[M_{eq\text{lever}}\] is the equivalent mass of box lever; \[\eta_{\text{cable}}\] is the cable efficiency; \[\eta_{\text{lead}}\] is the lead screw efficiency; \[\eta_{\text{gear}}\] is the efficiency of gear trains. \[L_2\] is the distance between actuator connection point and actuator frame lever pivot point; \[L_1\] is the distance between cable connection point and lever pivot point; \[L_1\] and \[L_2\] are shown in Figure 34; \[r_2\] and \[r_1\] are shown in Figure 36; \[r\] represents the sum of \[r_2\] and \[r_1\].

4.3.6.2 Total Equivalent Viscous Coefficient
Based on the fact that the actuation 2 system is very similar to actuation 1 system, the expression of equivalent viscous coefficient will not be discussed separately. The viscous-friction coefficient of the assembly transferred to electric motor end could be expressed as the following equation:

\[
B_{eq_{\text{total}}} = B_m + B_1 + \left(\frac{N_1}{N_2}\right)^2 B_2 + \left(\frac{N_1 N_3}{N_2 N_4}\right)^2 \left[B_3 + B_{\text{leadscrew}} + \left(\frac{S}{2\pi}\right)^2 \eta_{ac} + \left(\frac{S}{2\pi}\right)^2 \eta_{cable} \eta_{ac}\right]
\]

Where:

\[
B_{eq_{\text{cable}}} = \left(\frac{L_2 + L_1}{L_1}\right)^2 \frac{r_1^2}{r_2^2} B_{\text{cable}}, \quad B_{eq_{\text{trans}}} = \left(\frac{L_2 + L_1}{L_1}\right)^2 \frac{r_1^2}{r_2^2} B_{\text{trans}}
\]

\(B_m\) is the electric motor viscous coefficient; \(B_1\) is the viscous coefficient of gear 1; \(B_2\) is the viscous coefficient of gear 2 and gear 3 shaft; \(B_3\) is the viscous coefficient of the gear 4 shaft; \(B_{\text{leadscrew}}\) is the viscous coefficient of lead screw; \(B_{\text{nut}}\) is the viscous coefficient of nut, \(B_{\text{bolt}}\) refers to the viscous coefficient produced by bolts and \(B_{\text{tube}}\) is the viscous coefficient of tube; Similarly, the \(B_{eq_{\text{cable}}}\) refers to the equivalent cable viscous coefficient; \(B_{eq_{\text{trans}}}\) is the equivalent transmission shaft viscous coefficient. \(S\) is the lead of the lead screw bar; \(N_1, N_2, N_3, N_4\) are the teeth of the four gears; \(\eta_{cable}\) is the cable and actuator frame efficiency; \(\eta_{ac}\) represents the efficiency of actuator.

4.3.6.3 Total Equivalent Coulomb Friction Torque

If we consider the coulomb friction among all of the actuator system components, and combine the efficiency of lead screw, the total equivalent coulomb friction could be derived as the following:
Where:

\[
F_{\text{eq, total}} = F_{c1} \frac{\omega_1}{|\omega_1|} + F_{c2} \frac{\omega_2}{|\omega_2|} N_2 + F_{c4} \frac{\omega_4}{|\omega_4|} N_1 N_3 + \left( F_{\text{cnut}} \frac{v_{\text{nut}}}{|v_{\text{nut}}|} + F_{\text{c\_tube}} \frac{v_{\text{tube}}}{|v_{\text{tube}}|} \right) \]

\[
F_{\text{c\_lever}} \frac{\omega_{\text{c\_lever}}}{|\omega_{\text{c\_lever}}|} + F_{\text{eq\_cable}} \frac{v_{\text{cable}}}{|v_{\text{cable}}|} + F_{\text{eq\_bolt}} \frac{\omega_{\text{bolt}}}{|\omega_{\text{bolt}}|} + F_{\text{eq\_trans}} \frac{v_{\text{nut}}}{|v_{\text{nut}}|} S \frac{N_1 N_3}{2\pi \eta_{\text{ac}} N_2 N_4} \]

(4.35)

Where:

\[
F_{\text{eq\_cable}} = \left( \frac{L_2 + L_1}{L_1} \right) \frac{1}{\eta_{\text{cable}}} F_{\text{cable}}, \quad F_{\text{eq\_trans}} = \left( \frac{L_2 + L_1}{L_1} \right) \frac{r_1}{r_2} \frac{1}{\eta_{\text{cable}}} F_{\text{trans}}
\]

\[
F_{c1} \text{ is the first gear coulomb friction torque coefficient; } F_{c2} \text{ is the second and third gear}
\]
coulomb friction torque coefficient; \( F_{c4} \) is the fourth gear coulomb friction torque
coefficient; \( F_{\text{cnut}}, F_{\text{c\_tube}}, F_{\text{c\_lever}}, F_{\text{eq\_cable}}, F_{\text{eq\_bolt}} \) and \( F_{\text{eq\_trans}} \) represent the nut static
friction coefficient, tube static friction coefficient, lever friction coefficient, equivalent
cable static friction coefficient, equivalent bolt static friction and equivalent transmission
shifting lever static friction coefficient; \( S \) is the lead of the lead screw bar; \( N_1, N_2, N_3, N_4 \)
are the teeth of the four gears; \( \eta_{\text{ac}} \) is the actuator efficiency; \( \eta_{\text{cable}} \) represents the cable
efficiency.

### 4.4 Potentiometer in Control Systems

A potentiometer is the common device for position feedback in control systems. Since the
voltage of the variable terminal is proportional to the shaft displacement of the
potentiometer, this kind of sensor could give us the position of a system. The relationship
between the output voltage of the variable terminal and the shaft position could be
expressed as the following equation:

\[
e(t) = K_s \theta_c(t) \quad (4.36)
\]

Where:
\( K_s \) is the proportional constant of potentiometer

For an \( N \) turn potentiometer, this coefficient could be calculated in the following format

\[
K_s = \frac{E}{2\pi N} \quad (V/\text{rad})
\]  

(4.37)

Where:

\( E \) is the applied reference voltage for the fixed terminals.

4.5 Viscous Coefficient and Static Friction Value Determination

From the above statement of the linear actuator modeling procedure, it is clear that if we know the mass, material and shape of different gears, electric motor, lead screw, tube, and cable, etc., the total equivalent inertia at the electric motor end could be derived by some manipulation. However, the viscous coefficient and static coefficient are difficult to get just by calculation. The linear actuation system efficiency could be calculated just by equations. Thus, the test should be designed in order to get the parameters we need for this model.

In order to get the parameters of the linear actuator, the same series tests were designed to get the viscous coefficient, static friction torque and actuator efficiency. All the tests used to determine the parameters of transmission actuation system and linear actuator were described in first test rig described in Chapter 3. For viscous coefficient and coulomb friction test, different speeds were applied to linear actuator. As to the linear actuator efficiency and linear actuation system efficiency, various loads were attached to the baskets shown in Section 3.1. The following several paragraphs will show the tested parameter values.
4.5.1 Linear Actuator Parameters Test

Based on this kind of tests, the mean value of the back-emf was derived. Thus, the torque constant of the permanent magnet electric DC motor was also known since the values of the two parameters are the same.

4.5.1.1 Linear Actuator Viscous Coefficient and Static Coefficient Determination

Through linear actuator test, both the viscous coefficient value and static coefficient were obtained by first order curve fitting shown in the below Figure.

![Figure 37: Curve fitting for linear actuator damping coefficient and static friction.](image)

The obtained two parameters are shown in below Table:
4.5.1.2 Linear Actuator Efficiency Test

In order to get the accurate model of the transmission actuation system, the linear actuator efficiency is a parameter that could not be ignored. The linear actuator efficiency determines the system equivalent inertia. Thus, various loads have been added to the basket shown in Chapter 3. Through some manipulations, the linear actuator speed, load’s speed could be obtained from tested data. Combing with the damping coefficient and static friction obtained from section 4.5.1.1, the efficiencies under different loads could be derived.

The linear actuator was added with various loads ranging from 3.125 pounds to 26.875 pounds. The tested data and the fitted curve are shown in the below Figures.

<table>
<thead>
<tr>
<th></th>
<th>Extension test</th>
<th>Retraction test</th>
</tr>
</thead>
<tbody>
<tr>
<td>Viscous coefficient $B_m$</td>
<td>$1.41758e-05$N.m/(rad/s)</td>
<td>$1.08058116e-05$N.m/(rad/s)</td>
</tr>
<tr>
<td>Static friction torque $F_m$</td>
<td>$0.0127$N.m</td>
<td>$0.01216$N.m</td>
</tr>
<tr>
<td>$K_b/K_t$</td>
<td></td>
<td>$0.0190$</td>
</tr>
</tbody>
</table>

Table 7: Viscous coefficient and static friction of linear actuator
Based on the Figure 38, the linear actuator efficiency could be fitted comparatively well with quadratic curve. For the transmission shifting operation, the load added on linear actuator ranges from 11.68 pounds to 14.92 pounds. In this load range, the efficiency of retraction stroke is among 31.68% to 33.324%. And the efficiency range of extension stroke is between 37.7245% and 36.537%. From the tests, there is some minor difference of actuator efficiency for retraction and extension movement of linear actuator. However, linear actuator efficiency only influences the actuator tube, nut and box lever, which are small components and have very small inertia values. In order to simplify the position controller, it is reasonable to assume the linear actuator efficiency is constant. For linear actuator, the retraction efficiency could be regarded as 33% because at the beginning of lever movement, the load is higher than normal value. As to the extension efficiency, the efficiency could be regarded as 37%. For transmission shifting control, the difference
between extension and retraction could be ignored because the linear actuator efficiency only affects some small components. In the future work, a more detailed controller should consider the small difference.

The coefficients of quadratic curves are shown in the below Table:

<table>
<thead>
<tr>
<th></th>
<th>Quadratic polynomials coefficients</th>
</tr>
</thead>
<tbody>
<tr>
<td>Retraction test</td>
<td>-0.029489 1.29157 20.61793</td>
</tr>
<tr>
<td>Extension test</td>
<td>0.0134552 -0.72441 44.35457</td>
</tr>
</tbody>
</table>

Table 8: Curve fitting quadratic coefficients for linear actuator

4.5.2 Transmission Actuation System Parameters Test

4.5.2.1 Transmission Actuation System Viscous Coefficient and Static Friction Determination

Under the same power supply and the same recording devices, the tested linear actuation system contains the linear actuator, the linear actuator housing and the cable. The only item that was not and could not be contained in the test was the transmission shifting lever. The reason being the nonlinear nature of the disturbance load of this lever and also the movement range of the shifting lever is too limited to make the electric motor operate under the steady state condition. After some manipulation and calculation of the test results, the below Figures and Table show the curve fit and original test data which could give the viscous coefficient and the constant static friction for the linear actuation system.
Figure 39: Curve fitting for linear actuation system viscous coefficient and static coefficient

From the above Figure, it is obvious that first order polynomial equation could precisely represent the relationship between motor speed and system friction. The calculated parameters of the viscous coefficient and constant static coefficient both for extraction and retraction tests are shown in the below Table.

<table>
<thead>
<tr>
<th></th>
<th>Extension test</th>
<th>Retraction test</th>
</tr>
</thead>
<tbody>
<tr>
<td>Viscous coefficient $B_m$</td>
<td>1.7243-05N.m/(rad/s)</td>
<td>1.2867-05N.m/(rad/s)</td>
</tr>
<tr>
<td>Static friction torque $F_m$</td>
<td>0.0240N.m</td>
<td>0.0223N.m</td>
</tr>
</tbody>
</table>

Table 9: Parameters for extension and retraction movement of linear actuation system

4.5.2.2 Linear Actuation System Efficiency Test
The actuation system efficiency is very important for transmission modeling because the key parameter of equivalent inertia calculation is the system efficiency. In order to get the system efficiency, a series of tests were developed. The actuation system was added with various loads. Once the load speed, electric motor angular speed and electric motor torque were known, the efficiencies under different loads could be derived based on zero load system parameters.

These tests still utilize the same devices and setup in actuator parameters test, instead of direct adding the loads to actuator tube, the loads were added to the end of cable, which was connected to transmission shifting lever in vehicle. Different loads were added to the basket.

The efficiency fitted curve for extension test and retraction test are shown in below Figure. It is difficult to explain why the efficiency curves have the trends in the below Figure, but curve fitting could give system efficiency under one specific load.
Figure 40: Curve fitting for transmission actuation system efficiency

<table>
<thead>
<tr>
<th></th>
<th>Fourth order polynomials coefficients</th>
</tr>
</thead>
<tbody>
<tr>
<td>Extension test</td>
<td>-3.94e-05  0.00143  -0.0183  0.0955  0.0803</td>
</tr>
<tr>
<td>Retraction test</td>
<td>-3.42e-05  0.0011   -0.0129  0.0711  0.0756</td>
</tr>
</tbody>
</table>

Table 10: Curve fitting polynomial’s coefficients for transmission actuation system

From the efficiency fitted curve, the system efficiency under one specific load could be obtained based on the fourth order polynomials. From the measurement, in order to move the transmission lever, the resistance needed to overcome is among 3.6lb and 4.6lb. The corresponding efficiency for extension ranges from 0.2466 to 0.2533, and for retraction movement the efficiency changes from 0.2101 to 0.2117. In the working load range, the retraction efficiency almost has no change, while the extension efficiency changes in a small range. Based on the fact that there is no much change of system efficiency, the
system efficiency could be regarded as constant. In order to guarantee the stable and no overshoot position control, system efficiency was chosen to be 0.2101.

### 4.6 Actuation System Parameter Values

Based on the modeling process of the transmission actuation system, the final control-based modeling is the equivalent electric motor model. The indispensable parameters of the system are summarized in below Table:

<table>
<thead>
<tr>
<th>Parameter name</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Electric motor inductance ($L_a$)</td>
<td>514.4e-06 H</td>
</tr>
<tr>
<td>Electric motor resistance ($R_a$)</td>
<td>0.88397 ohm</td>
</tr>
<tr>
<td>Back-emf constant ($K_b$)</td>
<td>0.0190</td>
</tr>
<tr>
<td>Torque constant ($K_t$)</td>
<td>0.0190</td>
</tr>
<tr>
<td>System equivalent inertia ($J_m$)</td>
<td>1.111e-5 kg. m$^2$</td>
</tr>
<tr>
<td>System equivalent viscous coefficient ($B_m$)</td>
<td>1.2867e-5 N.m/rad</td>
</tr>
<tr>
<td>System efficiency</td>
<td>0.2101</td>
</tr>
<tr>
<td>Linear actuator efficiency</td>
<td>0.33</td>
</tr>
<tr>
<td>System coulomb friction</td>
<td>0.0240N.m</td>
</tr>
</tbody>
</table>

Table 11: Modeling values of actuation system parameters
Chapter 5: Adaptive Reference Algorithm of Transmission Start-Up

5.1 Motivation

For transmission shifting control, the adaptive reference logic of transmission shifting is indispensable. Adaptive reference method could effectively guarantee the completion of gear shifting and inform the supervisory controller of the shifting information. Adaptive reference logic happens at the warm-up stage. After the different gear positions were obtained through adaptive reference logic, the obtained data will be provided to the normal operation block and the controller will adopt the adaptively obtained positions to shift gears. The advantage of developing this algorithm is that it could guarantee the precise acquisition of every gear position and normal operation of the transmission. Under the vehicle driving conditions and transmission installations, the preset components’ positions are very likely to change based on installation error and vehicle vibration. If the components’ positions changed, the overall calibrated gear positions would change as well, which will lead to the shift failure. In addition to gear shifting requirement, the DFMEA of the system also requires the adaptive gear positions acquisition, which are crucial for the overall performance of the powertrain system. Referring back to Figure 2 shown in Chapter 1, this chapter focuses on the first part of the overall control strategy.
5.2 Adaptive Reference Logic in Warm-Up Stage

The warm-up stage is defined by the period during at key on, but the vehicle is not running. During this stage, the power supply allows the transmission actuators to detect whether the transmission has some failures or not, and allow the actuators to get the instantaneous gear positions every time the vehicle is in warm-up stage. The logic flow chart of this control algorithm is shown in the below Figure, and the simplified schematic diagram of transmission lever movement path is also shown below.
As is shown in Figure 42, based on the detection of occurrence of locked-rotor current and time duration of locked-rotor current, the controller could determine whether the linear actuator hits the boundary or runs normally. If there is a huge current increase in the electric motor during operation, two causes could be found. One cause could be due to the starting of the electric motor while another could be that the electric motor was stuck. For the starting moment of the linear actuator, the current spike will only last for 0.05 seconds, while the locked-rotor current will not disappear until the rotor is released. Therefore, counting the high value current duration, we could distinguish the locked-rotor from starting current. When the transmission controller is woken up during warm-up stage, the controller will automatically actuate the transmission lever to diagnose the transmission and acquire the gear positions. The first step of the control algorithm is to try to move transmission lever in X direction. If this step could work, then the controller
would go to maximum X direction and minimum X direction to acquire the two extreme positions. Then, based on the calibration of the transmission configuration, the X axis positions of different gears will be generated. If the tested X direction values are out of range or these values could not be corrected, the controller will report the failure to supervisory controller and make driver to take action. Once the X direction values are tested to be correct, next step commanded by controller is to go to the third and fourth gears’ X direction position, and then obtain the maximum and minimum Y axis positions of third and fourth gears. If the acquired Y direction extreme positions could meet the expected range, the final step of this controller is to generate the Y axis neutral position and update the neutral position to the controller. However, if the detected Y direction positions were out of range, the controller will report the error to supervisory controller as well. During the transmission operation period, the controller will use the new generated gear positions to complete shifting. On the other hand, if the transmission lever could not move in the X direction in the first step, then it will try to move in the Y direction. If the Y direction could not move as well, the transmission controller will report this failure to supervisory controller to check transmission. Generally, transmission mechanical failure will not occur. When the controller knows that transmission lever could only move in Y direction, it will command the lever to move up and down and acquire the Y direction maximum and minimum positions. Based on the two extreme positions, controller will move the lever to Y axis neutral position and redo the examination as stated in the first possibility. During this operation, any failures that stop the moving of transmission lever will be reported to supervisory controller. Figure 43
shows the neutral position recovery path of transmission lever movement under one stuck condition-the lever is stuck in the first gear.

Figure 43: Example for transmission lever path under warm-up stage
Figure 44: Flow chart diagram of adaptive control during warm-up stage
For this adaptive neutral position recovery algorithm, the implementation of the algorithm in SIMULINK is shown below:

Figure 45: Warm up adaptive control algorithm implemented in SIMULINK

Figure 46: Implementation of adaptive neutral position retrieval algorithm
The above two Figures clearly show the logic described in this section. In addition to the automatic searching of the neutral gear position, the X direction positions for all of the transmission gears could be calculated automatically by control algorithm. This function is very useful in preventing the gear shifting caused by installation error and position drifting error. Because system installation is very likely to change the initial calibration position, the calibrated positions before installation are meaningless. Due to the inconvenience of transmission recalibration when the transmission was integrated into the vehicle and the time consumed for recalibration, it is necessary to develop a control logic which could modify gear positions automatically. Therefore, this algorithm has been designed to automatically obtain the different gears positions.
Chapter 6: Normal Operation of AMT System

6.1 Control Strategy Requirements

In Chapter 5, the start-up mode has been discussed and implemented. The reference obtained from the warm-up stage will be sent to gear shifting during vehicle normal operation. Normal gear shifting requires stable position control to realize different gear shifting commands; therefore, a position controller is needed. In addition to the position controller, this strategy also should guarantee the completion of gear shifting command and notify the supervisory controller shifting completion. Thus, normal operation controller should also meet the above two requirements. Satisfying a highly accurate position control of linear actuator during transmission shifting is not a simple problem. For automated manual transmission gear shifting, the X direction total range is around 15mm, and the tolerance for moving into gear is around 1.5mm for X direction. The 1.5 mm range refers to the 0.75 mm tolerance radius for specific gears. Therefore, in order to guarantee no shifting failure, the overshoot of step response should be less than 5%. Besides the overshoot, in order to guarantee the shifting time and reach the desired position, it is reasonable to require the system has no steady state error. In order to overcome the resistance in different temperature and variable load, the controller should also reject the disturbance. The below Figure could better indicate the position of normal operation stage in the total AMT control strategy.
Figure 47: Transmission normal operation stage position in overall control strategy

6.2 Two Control Method in Normal Vehicle Operation Stage

In Section 5.2, the adaptive reference logic was described, and the next step for gear shifting is the reference algorithm for normal vehicle operation period. In order to realize the robust gear shifting, the controller should recognize whether the shift lever is already engaged into gear or not. Therefore, two gear shifting method were proposed for operation stage, which are shown in the below Figure.
Figure 48: Two method for gear shifting during vehicle operation

The first gear shifting method is position-based method, in which the set point for position control is inherited from start-up stage. Position is the only measured parameter in this method, and the shifting performance is totally determined by step response position control. As to the force-based shifting method, both the position feedback and current feedback were utilized to achieve gear shifting which does not need to know the exact calibrated position. The force-based reference logic is mainly for Y direction position control because shift lever needs to be guaranteed that it is engaged. It is very easy and clear to understand position-based gear shifting which is basically position control. For force-based method, the logic is the same as the neutral position recovery process presented in Section 5.2. The current feedback here was used to determine
whether the shift lever hits the boundary of gear slots and position window feedback was used to determine whether the shift lever was stuck in the correct position range. This particular method could guarantee the shift lever reaches the desired gear position even if the transmission or actuation characteristics changed. However, for first gear to fourth gear, only position-based control could be implemented because there is no boundary for these four gears. In this project, all gears’ Y direction control was selected to be force-based since the controller needs to determine the shifting completion. For fifth and sixth gear’s X direction control, both methods were performed and validated. For the gear shifting process, if the first shifting trial has failed, the control strategy will allow three times gear shifting retrial to mitigate the possible shifting failure. The following Figure shows the proposed control logic developed by State Flow chart and the implementation of the gears shifting algorithm.

Figure 49: Adaptive gear shifting algorithm implementation in Simulink
Figure 50: Control Algorithm of different gear shifting
6.3 Compensated PID Controller Development

As discussed before, some particular gears have to implement position-based control method. In this section, a position controller was developed in order to meet the requirements in Section 6.1.

6.3.1 Linear Actuator System Transfer Function

For the linear actuator system, the input is the voltage duty cycle which changes from 15% to 100%. The system block diagram is shown below.

![System Block Diagram](image)

Figure 51: AMT actuation system transfer function block diagram

Which:

- $G_{motor\_pot}$ is the angular displacement ratio between the electric motor and transmission potentiometer;
- $J_m$ is the equivalent electric motor inertia; $B_m$ is the electric motor equivalent viscous coefficient;
- $K_t$ is the electric motor torque constant; $K_b$ is the back-emf;
$L_a$ is the electric motor armature inductance, which is $514.4 \, \mu H$;

$R_a$ is the electric motor armature resistance, which is $0.88397 \, \Omega$;

$T_L$ is the disturbance for the system;

Based on the above block diagram of the linear actuator system, the transfer function for the system could be derived below:

\[
\frac{K_t}{S[(L_aS + R_a)(J_mS + B_m) + K_tK_b]} = \frac{D_{pot}}{E_a} \tag{6.1}
\]

### 6.3.2 Discrete Compensator Development

In order to optimize the linear actuator performance, the discrete time controller development is preferred because the ECU is a digital controller unit which deals with the sampling data. Inserting all the parameters to equation 6.1, the transfer function of the linear actuator system is shown below:

\[
\frac{0.0190}{5.7150e-09s^3 + 9.82752e-06 s^2 + 0.000372362 s} = \frac{D_{pot}}{E_a} = G_p(s) \tag{6.2}
\]

The control of the transmission shifting is realized by an ECU interfaced to a continuous time plant. Therefore, the ECU must sample the plant output via the Analog to Digital Converter (ADC), calculate the required control effort and apply this to the process through the Digital to Analog Converter (DAC). Since the computer is considered here, the discrete time signal will be sent out and the discrete position signal will be received. So, in order to model the sampled-data system and discrete time signal, the modeling theory of the sampled data system should be applied. For this system, there are three real poles which are: 0, 38.76, 1.6803e+03. Because the third pole is generated by electric
circuit and far away from the origin point, for the discrete time design, it could be neglected. Therefore, the dominant poles are 0 and 38.76. So, the sampling rate should be chosen as 250 Hz, which means the sampling time is 0.004s. At this sampling time, the controller could capture the system dynamics and will not generate too many step delays. Based on the modeling of zero order hold, the below z transfer function could be derived with the sampling time of 0.004 s:

\[
\frac{0.011289z^2 + 0.017401z + 0.00059042}{z^3 - 1.85757z^2 + 0.8586z - 0.0010281} = \frac{D_{pot}}{E_a} = G_p(z) \quad (6.3)
\]

From the above equation, the root locus of the transfer function of the linear actuator system could be drawn below to show the stability.

![Root Locus](image)

Figure 52: Discrete time root locus of the AMT actuation system transfer function

From the Figure shown above, if the proportional gain is under the 0.0117, the system is over-damped and the overshoot is zero. However, at this proportional gain value, the system could not meet the rising time requirement because it needs too much time to
reach the desired position and could not even reach the desired position because of the system load. If the proportional gain is larger than 8.22, the system will not be stable. Based on the requirement of the rising time, the proportional gain should be around 1. If the position feedback is from the potentiometer installed on the L-shaped lever, the system step response will be unstable with only proportional control. Although using the feedback from the transmission shift lever will make the system unstable, the transmission potentiometer was chosen to be the feedback because only the feedback from this location could represent the true position of the shift lever. The difference in step response between actuator built-in feedback and transmission feedback will be shown in the below Figures, illustrating the impact of step response produced by the non-colocated system.

![Figure 53: Simulation step response of linear actuator using different potentiometer feedback](image)

As is shown in Figure 53, even if we adopt the actuator potentiometer to provide feedback, there is almost 10% overshoot. If the transmission potentiometer is chosen, the system step response will be continuously oscillating. Based on these situations, a
discrete-time compensator is desired. Combined with a PID controller, the desired controller should eliminate the overshoot or reduce the overshoot to be below 5% and guarantee no steady state error.

It is easier to analyze the system and design the discrete time controller if the transfer function could be expressed as poles and zeros.

Thus, the z transfer function could also be expressed below:

\[
\frac{0.011289 (z + 1.50498)(z + 0.034712)}{(z - 1)(z - 0.8564)(z - 0.0012024)} = G_p(z) \tag{6.4}
\]

Based on the above discrete transfer function, a direct discrete controller could be designed. The direct design methods are based on the principle of finding a controller which could give the closed-loop a desired transfer function. Denoting the desired transfer function, which is also the closed-loop transfer function, to be \( F(z) \), and the plant transfer function to be \( H(z) \), the block diagram is shown below:

![Figure 54: Block diagram for system plant](image)

Rearranging the transfer function of the closed-loop function above, the expression for the controller \( D(z) \) in terms of the closed-loop transfer function \( F(z) \), and the plant \( H(z) \) could be shown below:
The controller derived with the above equation could firstly cancel the plant dynamics and then incorporate extra poles and zeros necessary to give the desired closed-loop transfer function.

For this system, the time constant should be 0.08 second so that the transmission lever could have a settling time less than 0.4 seconds. In addition to the above specification, the compensator should guarantee a zero steady state error.

For time constant less than 0.08 second, the \( F(z) \) must have a pole at \( Z < e^{-T/\tau} \), in this linear actuator control system, \( \tau = 0.08 \) and \( T = 0.004 \) second

\[
e^{-T/\tau} = 0.9512294
\]

Therefore a pole at \( z = 0.9 \) could satisfy the design requirement.

From the \( z \) transform of the system dynamics, there is one zero that is outside the unit cycle. Thus, according to the constraint of the discrete direct design, this zero, \( z = -1.50446 \), should be included as the zero of \( F(z) \). Since the system has a pole/zero of excess 1, \( F(z) \) needs at least have one extra pole at \( z = 0 \).

Based on the above statement, the closed-loop transfer function \( F(z) \) expression could be shown as below:

\[
F(z) = \frac{b_0(z + 1.50498)}{z(z - 0.9)}
\]  \hspace{1cm} (6.6)

Since this controller should achieve zero steady state error, the parameter \( b_0 \) could be found. Thus, the overall transfer function could be shown as below:
The direct design formula then gave the controller,

$$D(z) = \frac{1}{H(z) \left(1 - F(z)\right)} = \frac{3.495214(z - 0.8564)(z - 0.0011289)}{(z + 0.034712)(z + 0.06054576)}$$

(6.8)

The Simulink model of the designed controller is shown in below Figure, in which the universal PID controller is used to reject disturbance.

![Simulink model of the designed controller](image)

Figure 55: Transmission actuation system position controller SIMULINK implementation

### 6.4 Communication between Supervisory Controller and CAN

In order to realize the successful gear shifting and good speed matching between engine speed and transmission output speed, the communication between supervisory controller and transmission GCM is very important. The communication logic between the ECOGCM and supervisory controller is shown below. Based on the below algorithm, the supervisory controller could know the status of gear shifting and give the transmission
GCM controller the command to reach the desired gear. Thus, the desired operating condition will be realized.

Figure 56: Communication with supervisory controller and CAN message algorithm
In the above algorithm, under a no gear shift command, the supervisory controller will send the zero gear shift request to the GCM to keep the shifting lever in the current position and the gear shift status as zero. When the shifting lever is requested to shift to neutral position, the gear shifting request is 1. If the shifting lever is in neutral position, gear shift status will change to 1. Once a specific gear shift is needed, the GCM would receive a gear shift request of 3 and, during the shifting process, the gear shift status would be 3. Once the gear shifting is completed, the gear shift status will change to 4. In addition to the gear shift request which represents the gear shifting request, another parameter, which is the gear command, will indicate which particular gear needs to be engaged.
Chapter 7: On-Board Diagnosis and Shut-Down Mode

7.1 Motivation

For this specific transmission actuation system, the fault diagnosis is very important since this system contains many components and every possible component should be monitored in order to report failure to the supervisory controller. In addition to the components’ failure, gear shifting failure and warm-up stage failure need to be reported to the supervisory controller. Only one failure among electric motor, gear trains, lead screw, frame lever, cable and transmission lever will lead to severe results, that is, shift failure during vehicle operation. Therefore, the fault diagnosis is the post-processing method to alert the driver and allow the driver to take actions to fix the failure. As is shown in the previous chapters, the below Figure could represent the relationship of on-board diagnosis and park mode to normal operation stage and indicate the position of on-board diagnosis.
7.2 Design Failure Mode and Effect Analysis Procedure

In order to obtain the possible failure mode and design fault diagnosis strategy, Design Failure Mode and Effect Analysis (DFMEA) was adopted in this project. Failure Mode and Effects Analysis is the system function evaluation and reasoning of the potential failures and causes of these failures [16]. In the fault diagnosis of systems, the FMEA could be used to analyze the effect of system faults and failures of subsystems. This method is very important for system reliability and could report the failures in time in order to take related action. Thus, FMEA could improve and optimize both the product design and test design. According to the statement of Zhang Zhong et al. [15], the fundamental procedure of FMEA could be shown below:
7.3 DFMEA Results

In this paper, the automated manual transmission actuation system was divided into the following part: power supplier, linear actuator, transmission and actuator connection components, and the control strategy design. The linear actuator contains the electric motor circuit, gear trains, and lead-screw drive. The transmission-actuator connection contains the linear actuator housing frame, housing lever, and push-pull cable. Part of the potential components failure, failure detection method and recommended actions are shown in the below Table.
<table>
<thead>
<tr>
<th>Item/Function</th>
<th>Potential Failure Mode</th>
<th>Potential Effect(s) of Failure</th>
<th>Severity</th>
<th>Potential Cause/ Mechanism of Failure</th>
<th>Current Design Controls Prevention Controls</th>
<th>Detection Controls</th>
<th>RPN</th>
<th>Recommended Action</th>
</tr>
</thead>
<tbody>
<tr>
<td>Voltage source supplies 12V voltage to actuator assembly</td>
<td>Supplies no voltage</td>
<td>Actuator unable to work</td>
<td>5</td>
<td>Short circuit (to ground or battery)</td>
<td>Robust connection</td>
<td>Supervisor controller recognizes loss of communication</td>
<td>1 15</td>
<td>Alert to driver</td>
</tr>
<tr>
<td>Wrong or inconsistent voltage</td>
<td>Battery failure</td>
<td>Battery diagnosis</td>
<td>7</td>
<td>Robust Wiring</td>
<td>Supervisor controller recognizes the loss of communication</td>
<td>4 20</td>
<td>Report to supervisory controller</td>
<td></td>
</tr>
<tr>
<td>Electric motor in the actuator drives the actuator drive</td>
<td>Electric motor failure</td>
<td>Actuator failure</td>
<td>8</td>
<td>Electric motor circuit mechanical damage</td>
<td>None</td>
<td>Fault detection control on sensor</td>
<td>4 28</td>
<td>Report to supervisory controller</td>
</tr>
<tr>
<td>Electric motor armature overheated</td>
<td>Electric motor damage</td>
<td>Actuator damage</td>
<td>7</td>
<td>Fault detection control on sensor</td>
<td>Long time large current of the armature input</td>
<td>Detection of large current and disable the power</td>
<td>6</td>
<td>Fault detection on software and position sensor</td>
</tr>
</tbody>
</table>

Table 12: DFMEA table of transmission actuation system
<table>
<thead>
<tr>
<th>Item/Function</th>
<th>Potential Failure Mode</th>
<th>Potential Effect(s) of Failure</th>
<th>Current Design Controls</th>
<th>Recommended Action</th>
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<tr>
<td>Actuator output the force to housing lever</td>
<td>Gear trains or lead screw yield</td>
<td>Actuator failure Shift incomplete</td>
<td>Current Design Controls</td>
<td>Prevention Controls</td>
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<td>Frame lever and cable drive the shift lever</td>
<td>Lever yield or cable connection failure</td>
<td>Shift failure</td>
<td>Current Design Controls</td>
<td>Prevention Controls</td>
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<tr>
<td>Actuator built in position sensor measures the movement of tube</td>
<td>No signal</td>
<td>Actuator failure Shift failure</td>
<td>Current Design Controls</td>
<td>Prevention Controls</td>
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<th>Potential Cause/Mechanism of Failure</th>
<th>Current Design Controls</th>
<th>Prevention Controls</th>
<th>Detection Controls</th>
<th>RPN</th>
<th>Recommended Action</th>
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<td>Signal out of range</td>
<td>Wrong position control signal</td>
<td>5</td>
<td>Sensor damage</td>
<td>installed correctly</td>
<td>1</td>
<td>Compare signals from transmission lever sensors</td>
<td>5 25</td>
<td>Report the supervisory controller, shift to neutral position</td>
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<td></td>
<td>Mechanic al failure</td>
<td>check the actuator system</td>
<td>2</td>
<td>Compare signals from transmission lever sensors</td>
<td>5 50</td>
<td>Report the supervisory controller, shift to neutral position</td>
</tr>
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<td></td>
<td></td>
<td></td>
<td></td>
<td>Transmiss ion shaft sensor</td>
<td>measures the movement of shaft</td>
<td>8</td>
<td>Alert the driver, disable the actuator</td>
<td></td>
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<td></td>
<td></td>
<td>No signal</td>
<td></td>
<td>Incorrect wiring</td>
<td>Wiring correctly</td>
<td>2</td>
<td>Compare the signal from actuator built in sensor</td>
<td>3 48</td>
<td>Alert the driver, disable the actuator</td>
</tr>
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<td></td>
<td></td>
<td>Shift failure</td>
<td></td>
<td>Potentio meter damage</td>
<td>Installed correctly Actuator sensor</td>
<td>1</td>
<td>Compare the signal from actuator built in sensor</td>
<td>3 24</td>
<td>Alert the driver, disable the actuator</td>
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<td>Mechanic al failure</td>
<td>Installed correctly Actuator sensor</td>
<td>1</td>
<td>Compare the signal from actuator built in sensor</td>
<td>3 24</td>
<td>Alert the driver, disable the actuator</td>
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<th>Potential Effect(s) of Failure</th>
<th>Sensor Damage</th>
<th>Potential Cause/ Mechanism of Failure</th>
<th>Current Design Controls Prevention Controls</th>
<th>Detection Controls</th>
<th>RPN</th>
<th>Recommended Action</th>
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<td>[6.2] Signal out of range</td>
<td>Wrong position control signal</td>
<td>Sensor damage</td>
<td>installed correctly</td>
<td>1</td>
<td>Compare the signal from the actuator sensor</td>
<td>5 35</td>
<td>Report to the supervisory controller, and shift transmission to neutral</td>
<td></td>
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<tr>
<td></td>
<td></td>
<td>Mechanical failure</td>
<td>Check the drive system</td>
<td>2</td>
<td>Compare the signals from actuator sensor</td>
<td>5 70</td>
<td>Report to the supervisory controller, and shift the lever to neutral</td>
<td></td>
</tr>
<tr>
<td>Warm-up Control algorithm acquire the new gears positions</td>
<td>Failure in adaptive gear positions acquisition</td>
<td>Algorithm design defect</td>
<td>Test the algorithm in HIL</td>
<td>6</td>
<td>Analysis of the movement of transmission lever</td>
<td>6 25 2</td>
<td>Report to supervisory controller, shift the transmission to neutral</td>
<td></td>
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<tr>
<td></td>
<td></td>
<td>Connectors mechanical failure</td>
<td>Install every components correctly</td>
<td>3</td>
<td>Fault detection control on sensor and software</td>
<td>4 84</td>
<td>Report to supervisory controller, shift the transmission to neutral</td>
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<th>Item/Function</th>
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<td>Operation Tests on HIL</td>
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<td>Sensor and Software</td>
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<td></td>
<td>Fault Detection Control</td>
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<td></td>
<td>Report to Supervisory Control, Shift to Neutral</td>
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<tr>
<td>Warm-up control algorithm predict the failure of transmission</td>
<td>Inconsistent data</td>
<td>Wrong report to supervisory controller</td>
<td>6 42 Report to supervisory controller, Shift to neutral</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Sensor Damage or Mechanical Failure</td>
<td>Install every component correctly</td>
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<td></td>
<td></td>
<td>Control Algorithm Defect</td>
<td>Compare signal from actuator sensor</td>
</tr>
<tr>
<td>Adaptive position control guarantee every shift in gear</td>
<td>Actuator does not move when requested</td>
<td>Shift incomplete</td>
<td>Control Algorithm Defect</td>
</tr>
<tr>
<td></td>
<td>Actuator moves when not requested</td>
<td>Shift incomplete</td>
<td>Control Algorithm Defect</td>
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<td>Prevention Controls</td>
<td>Detection Controls</td>
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<td>Actuator moves to incorrect distance</td>
<td>Shift incomplete</td>
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<td>Control algorithm defect</td>
<td>Controller test on HIL</td>
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<td></td>
<td></td>
<td>Transmission damage</td>
<td>Operation tests on HIL</td>
<td>3 24</td>
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<td></td>
<td></td>
<td></td>
<td></td>
<td>Connecter mechanical failure</td>
<td>Install every components correctly</td>
<td>4 64</td>
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</table>

Based on the recommended actions from the DFMEA table, the fault diagnosis controller was developed in order to identify different failure modes and report the failure to the
supervisory controller. Once the GCM has determined that the failure is not repairable, the controller will automatically move the shifting lever to neutral position in order to guarantee the safety of vehicle subsystems. The fault diagnosis was performed in both the vehicle warm-up stage and normal operation stage. With the implemented fault diagnosis strategy, the supervisory controller could know the input power voltage failure, electric motor failure, gear trains or lead screw failure, housing lever or push-pull cable failure, failure of the two potentiometers, transmission shifting pattern failure during warm-up stage and normal operation stage. The implementation of on-board diagnosis of the actuation system is shown in the following Figure.

1.2.2 Fault Diagnosis
EcoGCM / Model / Controller / Fault Diagnosis

Figure 59: Implementation of actuation system on-board fault diagnosis
7.4 Shut-Down Mode

When the vehicle was shut down and when the vehicle was parked, the supervisory controller would send a command to the transmission GCM and requested the shifting lever to move to fifth gear. Once the park mode shifting was finished, the supervisory controller would receive the completion notice and shut down the vehicle.
8.1 Motivation

In order to validate the modeling of the actuation system and the control of transmission actuation system, the NI DAQ, MotoTune and MotoHawk mentioned in Chapter 3 will be used. For actuation system position control, the equivalent system model could provide simulation validation before the controller is implemented in the transmission’s ECU. Due to the unpredictability and the requirement for parameter feedback, the adaptive control algorithm could not be adapted into a software-in-the-loop model to test the gear shifting process and neutral position retrieving process. Although the software tests for the adaptive shifting algorithm are not available, both the position controller and adaptive control algorithm will be validated and transmission shifting data will also be collected, through the setup below, and processed in MATLAB. This will actually validate the accuracy of modeling and effectiveness of the designed controller. Additionally, only the validation of control strategy could not meet the control development requirements. As discussed in Chapter 1, the ambient temperature test and load test for both the shifting lever step response and gear shifting were performed; temperature tests were also performed for start-up mode. A durability test for normal operation mode under different temperatures was developed to fully validate the
robustness and durability of the control strategy. The influence of ambient temperature and load to a step response and gear shifting time were analyzed.

8.2 Software Validation

The transmission shifting lever position controller described in Chapter 6 and the modeling of actuation system presented in Chapter 4 were validated in simulation. A simple simulator of the actuation system was developed and used to verify the functionality of the position controller. The simple simulator could also accelerate the development of a controller and make the system better understood because the control results could be observed in a much shorter time compared to implementation in hardware devices. What is more, the simulation could easily see the influence to the system response of one specific parameter change. The following system equivalent electric model described in Chapter 4 will be used as the simulator for modeling software validation, while the controller presented in Chapter 6 will be utilized as the simulator for position control validation.

8.2.1 Actuation System Modeling Software Validation

In order to validate and verify the feasibility of system modeling methodology, a simple simulator has been developed to simulate the step response of the actuation system only based on feedback and the proportional controller. The simulator diagram is shown below:
Figure 60: Transmission actuation system proportional feedback control SIMULINK implementation

The actuation system step response simulation results are shown in Chapter 6, section 6.3.1. The difference of step response based on different locations of potentiometers indicates the influence of distance between actuator and feedback devices.

**8.2.2 Actuation System Controller Software Validation**

The discrete controller simulator shown in Chapter 6, section 6.3.2 could verify the performance of the designed controller in SIMULINK and effectively improve the controller.

The step input simulation results of the designed controller under a no disturbance condition are shown below:
In Figure 61, the performance of the designed controller was indicated by the red curve. Compared to the performance of the proportional only controller, the developed compensator combined with a PI controller shows satisfactory position control results. The system oscillation will not occur for step input, both rising time and settling time are guaranteed in desired time, and no overshoot will occur during position command. In conclusion, the developed controller could satisfy the requirement of transmission shift lever position control based on the simulation results.

8.2.3 Actuation System Controller Load Test Software Validation

In order to fully validate the robustness of the controller, the actuation system step response has been simulated under different loads. The selected load range is 20% load to 200% load. The reason for different load simulations is to examine whether the controller
could guarantee the shift lever step response performance. The simulation results are shown below:

Figure 62: Actuation system load test simulation results

Figure 63: Simulation step response specification results for load test
As can be seen in the above two Figures, the higher the load value, the longer the actuator needs to complete the step response. Both the settling time and rising time are proportional to the system load, which is reasonable for the system. Comparing the settling time trend to the rising time trend, it is clearly shown that system settling time is more sensitive to load changes than rising time. In Figure 63, it can be seen that within reasonable load range, the system step response will not contain overshoot oscillation. Therefore, the position controller developed for transmission actuation system is not sensitive to the load changes.

8.3 Hardware Validation

After the simulation of the controller, the actual data collected from transmission setup will be used to further validate the performance. For the following tests, different test procedures and data were recorded and processed in MATLAB. To validate the algorithm, tests of the system step response position control, gear shifting tests and neutral gear position retrieving tests were conducted and all of the data was collected. The validation results will be described separately in the following paragraphs.

8.3.1 Actuation System Modeling and Position Controller Validation

8.3.1.1 Actuation System Modeling Validation

In order to validate the modeling of the actuation system, a system step response without the compensator is shown in the following Figures. Although the modeling parameters of the system are constant, the delay produced by the location of the potentiometers determines the stability of the system. Based on the fact that the X direction actuation
system has two potentiometers, one of which was installed in the linear actuator and another one is installed on the transmission shifting lever, the stability of the system is largely determined by the X direction actuation system. The built-in potentiometer is very close to the electric motor of the actuator, which means using the signal collected from this potentiometer could maintain the system’s stability. However, the received signal from the potentiometer which was installed far away from the actuation point (electric motor) makes the system inclined towards instability. In order to fully validate the model of this system and indicate the influence produced by the non-colocated feedback, both the simulation and test results under the two kinds of feedback are shown below.

Figure 64: Actuator potentiometer feedback proportional control system step response
From the above Figure 64 and Figure 65, the influence of the non-colocated feedback system was clearly validated through hardware tests. The above two Figures also indicate that the modeling of this system is reliable. The developed system model could present and indicate the step response of the actual system. In addition to the verification of the system model, the comparison between the above two Figures proves that even though the system itself is stable, non-colocated feedback has the possibility to make the system unstable. The backlash and yielding between system components make the system difficult to control. Therefore, for this kind of system, a compensator is required to regulate the control signal.

8.3.1.2 Actuation System Controller Validation
Figure 66: Implementation of actuator discrete compensator with PID controller

Figure 67: Actuation system discrete controller step response validation
In Figure 67, the actual test data of the actuation system step response is almost the same as the simulation result. It is clearly shown that the 10mm step response of the system could be completed in less than 0.5 seconds and no overshoot has occurred in the actuator step response process. The small steady state difference is due to the effective control range set. Instead of a specific X direction position value of one specific gear, the desired gear X direction position is a range, during which the shifting lever could be engaged into gear. Therefore, once the shifting lever is in this range, no control effort would be sent to the actuator. The reason for this kind of controller is to eliminate the motor noise once the transmission is engaged, without which the motor will try to eliminate position error and produce operating noise. Combining the compensator and PID controller could guarantee no overshoot, rejection of disturbance, and a less than 0.5 seconds shifting time.

In conclusion, the designed actuator position controller could satisfy system shifting specifications which require no overshoot, smooth position transition, 0.3 second rising time and rejected disturbance.

8.3.2 Adaptive Neutral Gear Position Recovery Algorithm Validation

From the content discussed in Section 5.2, after the control algorithm was implemented in SIMULINK and with the experimental setup described in Chapter 3, the adaptive control algorithm validation data was collected and processed in MATLAB. For the validation of the algorithm, the chosen stuck position of the shifting lever was in the 4th gear slot, in which the initial X direction value is 32.5 and Y direction value is 50. The
The following three Figures indicate the function of the adaptive neutral position recovery algorithm.

Figure 68: Neutral position recovery validation for shifting lever stuck in 4th gear

Figure 69: Neutral position recovery X direction current validation for 4th gear
Figure 70: Neutral position recovery Y direction current validation for 4<sup>th</sup> gear

From the above three Figures, the shifting process is clearly shown. In the beginning, the X direction actuator attempted to move, however, it could not move because the lever was stuck in the 4<sup>th</sup> gear slot. After the controller detects that the X direction moving was stuck, it will give the command to move the shifting lever in Y direction. Since the Y direction allows the lever to move, the controller will let the lever go through the Y direction path and calculate the neutral range of Y direction. Once the shifting lever reached the neutral range, the X direction actuator would start moving and get the accurate positions for all of the gears. After the X direction gear positions are obtained, the controller will make the lever go to the neutral position. The last step is to double check the Y direction range under the neutral X direction position. In this case, the X direction neutral position does not change because the initial position is in 3<sup>rd</sup> / 4<sup>th</sup> gear.
From Figure 70, it can be seen that the Y direction goes into neutral range which is 37.86 mm instead of stuck in the 50mm position.

In validation tests of this algorithm, all possible stuck positions of the transmission lever were tested and evaluated. In this thesis, only the 4th and 2nd gear stuck positions will be shown. The following figures will indicate the neutral position retrieving when the shifting lever was stuck in the 2nd gear slot.

![Neutral position retrieving algorithm validation for actuator stuck in 2nd gear](image)

**Figure 71: Neutral position recovery validation for shifting lever stuck in 2nd gear**

Looking at the above Figure, the proposed adaptive control algorithm was validated through different possible position retrieving tests. For every possible non-neutral actuator stuck position, the controller could find the neutral position in less than 8 seconds. Neutral position retrieving time depends on where the transmission shifting lever is stuck; if the lever was not stuck in the X direction, the required shifting time to
find the neutral position will be less than 4 seconds. Even if the required shift time is 8 seconds, it is acceptable for the vehicle warm-up stage and will not affect customer acceptability.

8.3.3 Normal Operation Algorithm Validation

As mentioned in Section 6.2, the proposed gear shifting algorithm could guarantee and notify the supervisory controller of the completion of gear shifting based on the value of the potentiometer and electric motor current. The Simulink diagram of the adaptive gear shifting algorithm subsystem is shown in Figure 72:

Figure 72: Normal operation stage subsystem SIMULINK diagram

In the algorithm diagram shown above, the adaptive algorithm will notify the supervisory controller of the shifting status when the specific gear request has been received, allow three times shifting retrial if shifting failure occurs, and report failure of different gears
during the shifting period. Besides the above functions, the algorithm could also detect shifting retrial times for each gear, which could provide shifting quality information.

Seven validation tests were performed on this control algorithm, which contains six forward speed shifting tests and one reverse gear shifting tests. For detailed analysis, the fifth gear shifting is chosen because it contains the detection of both the X direction boundary and the Y direction boundary.

Figure 73: Force-based control validation of fifth gear test with X direction current

Figure 74: Force-based control validation of fifth gear test with Y direction current
Based on the above marked two Figures, it is clearly shown that the controller could recognize the boundary of the transmission. Once the lever position was in a certain range and the current value was higher than 5.8 amperes, the controller could generate backward voltage to eliminate the tension between system components. Once the position of the shifting lever is in the X direction range, the Y direction actuator will start moving to the end of the desired gear gate. The Y direction actuator will also release the tension force between the system components.

Besides the fifth gear shifting test, all other gears have been tested over hundreds of times. The following Figures present the validation of the other gears.

![Adaptive first gear shifting validation](image)

Figure 75: Normal operation algorithm validation test for first gear
Figure 76: Normal operation algorithm validation test for second gear

Figure 77: Normal operation algorithm validation test for third gear

Figure 78: Normal operation algorithm validation test for fourth gear
Figure 79: Normal operation validation for sixth gear with X direction current

Figure 80: Normal operation validation for sixth gear under Y direction current

Figure 81: Normal operation validation test for reverse gear under X direction current
The adaptive gear shifting algorithm has been tested for every possible gear. Each gear shift uses the current feedback and position feedback to detect the shifting status. For transmission gear shifting, each gear’s shifting time also needs to be considered, tested and evaluated. Therefore, the below Figure summarizes the shift time for every gear.

![Force-based and position-based shifting time](image)

**Figure 82: AMT different gear shifting time**

In the above figure, the red bars are the gear shifting times which use the force-based method to determine 5th, 6th and reverse gear X direction position, while the blue bars are the shifting times using the position-based method. It can be seen that using the position-based control, the shifting time could be controlled in less than 1 second. If the force-based control was adopted to shift 5th and 6th gears, the shifting time will be between 1.1-1.3 seconds. With the position-based control, the shifting time for all gears is less than 0.85 second. However, for a PHEV which utilizes the front motor to speed match the engine and transmission speed, shifting time is not the most important factor.
8.3.4 Actuation System Controller Load Test Hardware Validation

In the software validation section, the actuation system load test has been simulated based on the simulator. In this section, the load test has been implemented in a hardware validation test. The load range in the hardware test is from 10 % to 200%. The step response results under different loads are shown in the below Figure:

![Graph showing step response test results under different loads](image)

Figure 83: Actuation system controller load test hardware validation

As can be seen in the above Figure, the hardware load test results are similar to the simulation test results. The heavier the load, the more time needed to complete the step response. These results are consistent with the simulation results, no matter what the load is, and the performance of the step response could satisfy the transmission shifting requirement. The trend of settling time and rising time under various temperatures are shown in the following Figure:
Figure 84: Load test step response specification simulation and real tests comparison

It can be seen that the results of settling time and rising time from actual tests are similar to the simulation results shown in Figure 84. With the increase of the load, the settling time and rising time also increase. Compared with the simulation results, the settling time in the actual system is more sensitive to load than the simulated results of the system. The difference of settling time between 200% load and 20% load in real tests is around 0.133 seconds, while the difference in simulation results is only around 0.6 seconds. Although the system is a little bit more sensitive in real tests, the rising time has the same sensitivity for both tests under load variation. In conclusion, based on the software validation results and hardware validation results for the system step response and system load tests, the developed system model could fully represent transmission actuation.
system dynamics. The system simulator could provide very useful information for this system under various inputs.

8.3.5 Actuation System Step Response Temperature Test

In addition to the load test implemented in software and hardware tests, the temperature hardware tests of transmission actuation is also very important for system step response robustness and reliability. Because there is no method to simulate the temperature influence on the actuation system, the hardware temperature tests seem to be more crucial to validate controller sensitivity, robustness and durability. In the temperature test, the system was kept as original, that is, no additional load had been applied on this system. The whole system was put into the climate chamber which could change the environment temperature. The actuation system step response is shown below:

![Actuation system step response under different temperature](image)

Figure 85: Actuation system step response ambient temperature test results
It can be seen in Figure 85 that ambient temperature would influence the step response of the actuation system. Low ambient temperature will slow down the system step response speed, while higher temperature will make the step response faster. The reason for this situation is that low temperature will increase the system internal resistance and increase the system viscous coefficient value, which could slow down the response speed. To our satisfaction, the ambient temperature will not lead to overshoot. Despite some minor influence produced by ambient temperature, the overall controller performance could satisfy the requirement. Therefore, both the designed controller and the system are robust, durable and reliable in their work conditions. The step response performance specifications under the temperature tests are shown in Figure 86.

![System step response specifications temperature test](image)

**Figure 86: Actuation system temperature test step response specification**

It can be seen that settling time and rising time are shorter under high temperature, while these two specifications are longer under low temperature. This conclusion indicates that
even in high ambient temperature, the overshoot of system step response is still limited to 4%, which satisfies the less than 5% overshoot requirement. In low temperature, the system is seen to be over-damped because the viscous coefficient increased. In conclusion, the step response controller is robust and reliable under the tested temperature range.

8.3.6 Transmission Gear Shifting Temperature Test

In order to validate the robustness and reliability of gear shifting strategy, the transmission gear shifting algorithm was examined under a different temperature range. The selected temperature range was between -15 Celsius degree and 60 Celsius degree. The shifting time of all gears under various temperatures are shown below:

![Gear shifting time temperature test](image)

Figure 87: Gear shifting time tests under various ambient temperatures
It can be seen in the above two Figures that ambient temperature has some influence for gear shifting time. For the first and second gear shifting time, a shifting time under -15°C degrees needs 20% more time than the shifting time under 60°C degree. As to the third, fourth and reverse gear shifting, the influence to shifting time produced by ambient temperature is not serious. The extra time needed for these three gears’ shifting under low temperature is less than 10%.

However, the impact of ambient temperature to the shifting time of 5th and 6th gears could not be ignored, especially for the force-based X direction gear shifting strategy. Because the force-based Y direction actuator is always active and could not be replaced, Y direction shifting method will not influence shifting time. Once the 5th and 6th gears
adopted the force-based control strategy, the shifting time under low temperature will increase to more than 200% of the shifting time under high temperature, which is clearly shown in Figure 88. If the position-based control was utilized, the shifting time was not seriously influenced by ambient temperature. Therefore, whether the X direction force-based control strategy will be chosen or not should be based on the system requirement. If fast gear shifting is preferred, the position-based control is the first choice. If intelligence is preferred, the force-based control needs to be considered. Despite the unsatisfactory shifting time of the force-based 5th and 6th gear control, the shifting time of other gears was satisfying and robust enough for AMTs in PHEVs.

8.3.7 Transmission Gear Shifting Durability Test

In addition to the validation test and robustness test, the durability test was also performed in this project. The durability test was performed in the climate chamber as the experimental setup described previously. The selected durability tests’ ambient temperatures were -15 Celsius degree, 5 Celsius degree and 60 Celsius degree. The durability test results are shown in the below Figure.
Figure 89: Climate chamber different temperature durability test

In the above Figure, the durability test shift number of times is around 8000-10000 times for every ambient temperature. During the whole durability test process, no shifting failure occurred and no shift retrial occurred. The durability test verified that the system and control strategy could satisfy our requirements.
Chapter 9: Conclusion and Future Work

9.1 Conclusions

The model of the ECOCAR 2 AMT actuation system was successfully built and validated in this thesis. In order to create the model, a parameter test was also designed. The setup of the parameter test, which contains NI DAQ and a linear actuator frame, was completed by the transmission team. A curve fitting method was adopted in order to get the system and linear actuator efficiency. The actuation system model combined with a simple proportional feedback controller was validated in a hardware-in-the-loop testing device which was coupled with a real GM 6-speed manual transmission. The dynamic process of a step input response showed overshoot and small oscillation.

Based on the actuation system model, the compensator of the system was designed and validated in both simulation and the hardware-in-the-loop system. Combined with a conventional PID controller, this controller could realize no overshoot, desired settling time and no steady-state error while the simple proportional feedback control had oscillation. In addition to the position control, an adaptive reference algorithm which could retrieve the shifting lever neutral position and obtain gear positions was developed as well. After the adaptive reference algorithm and neutral position recovery were completed, the position-based and force-based gear shifting methods were described. The above adaptive control strategies were validated in both a hardware-in-the-loop system.
and self-made transmission spinning test platform. On-board diagnosis and adaptive
control algorithm could also mitigate the impact of shifting failure and inform the driver
of irreparable shifting failure. Besides the mitigation of the shifting failure, on-board
diagnosis could detect system failure in real time and report the failures to the
supervisory controller in order to take actions. Additionally, the control strategies were
further validated through load tests, temperature tests and durability tests. Through all the
tests mentioned above, the step response of the position control could guarantee stability,
no overshoot and fast movement; the adaptive reference logic worked in all temperatures;
proposed gear shifting methods satisfied our requirements in all tests.

The transmission spinning test platform was built and controlled in this research to test
gear shifting quality when the transmission has input and output speed. This test platform
could provide speed matching tolerance for different gears which is needed for speed
matching between engine and transmission. From hardware-in-the-loop tests and
transmission spinning tests, the gear shifting control and position control were proven to
be competitive with a production automated manual transmission and adaptive reference
algorithm unique in AMT control.

9.2 Future Work

Although the validation of the AMT position control algorithm indicated satisfactory
results, there are some improvements that could be made to the modeling and control of
the actuation system. One improvement is the modeling of the system. The parameters of
the equivalent model of the system are not absolutely constant over the entire service life,
thus, the controller, which is based on constant system parameters, is not perfect. Although the parameter variation is small, the accurate controller needs to consider this variation. Therefore, if it is possible, an estimator which could estimate the inertia and damping coefficient on-line should be developed in discrete time format.

In addition, if it is possible, the linear actuation system should change the controller automatically corresponding to the changing system parameters. The best way to control the linear actuator is to develop a controller which could adjust the compensator or find a new controller to adjust the control gain according to the model parameters. Therefore, an adjustable on-line discrete time controller is preferred.

In the end, future work could include placing a new potentiometer on the transmission for actuator 2, with the actuator built in potentiometer and the potentiometer on the transmission, the DFMEA could also be applied on the actuator2 driveline. Once the DFMEA could be applied to both the actuator1 and actuator2, all kinds of failure and analysis could be reported to the supervisory controller. In short term, the transmission spinning test needs to be completed to get every gear speed matching tolerance, and combine the actuation system tests with the spinning test platform to further validate AMT control strategy in transmission rotating conditions.
Bibliography


Appendix A: SKF Linear Actuator Data Sheet

This appendix shows the data sheet of linear actuator
Figure 90: SKF CAHB-10 linear actuator datasheet
Figure 91: SKF CAHB-10 linear actuator datasheet
Appendix B: Transmission Spinning Test Platform Electric Motor Encoder Data Sheet

This appendix shows the transmission spinning test platform components design specifications
PU-E Series Hall-Effect Pick-up

The PU Series pick-up is an economical and reliable way to monitor motor speed. Its unique design provides ease of installation in otherwise difficult to reach areas. The PU pick-up operates at a 5 to 24 volt level producing a sharp square wave output which may be fed into Dutt's field-programmable autotuner, closed loop control, counters, or any other digital device.

The PU pick-up series also includes a quadrature model to monitor both motor speed and direction by providing two square wave output signals 90° out of phase.

**STANDARD FEATURES**
- PU Series pick-up mounts directly on shaft being monitored using a single 10-32 screw.
- Maximum speed: 5,000 RPM or 50,000 pulses per minute.
- Supply voltage: +5VDC to +24VDC.
- NPN open collector output signal with built-in pull-up resistor.
- Supply voltage output signal voltage equals supply voltage: +5VDC to +24VDC, supply voltage: Current sink 5mA, 250mA maximum.
- Operating temperature: 0°C to +100°C.
- Stainless steel ball bearing.
- Compact housing of molded “Sanitoxene” plastic rubber.
- Output cable: 3 rubber jacketed, 3 wire 18AWG conductors; red wire: +5VDC supply input; black wire: Common; white wire: Signal A; brown wire: Signal B (model PU-20EUQAD only).

**PU SERIES SELECTION GUIDE**

<table>
<thead>
<tr>
<th>MODEL</th>
<th>PU SERIES SOLUTION</th>
</tr>
</thead>
<tbody>
<tr>
<td>FU-3E</td>
<td>FU-3R</td>
</tr>
<tr>
<td>FU-4E</td>
<td>FU-4R</td>
</tr>
<tr>
<td>PU-3E</td>
<td>PU-3R</td>
</tr>
<tr>
<td>PU-4E</td>
<td>PU-4R</td>
</tr>
<tr>
<td>FU-RO7</td>
<td>FU-40</td>
</tr>
</tbody>
</table>
| PU-20EUQAD | PU-20RUQAD | (10 pair directions)

**DIMENSIONAL SPECIFICATIONS**

**INSTALLATION AND WIRING**

No other mounting brackets or sensors are necessary as the cord will keep the unit from rotating. The PU gives a high signal when the south pole of the magnet passes across the Hall-effect transistor. The signal is switched low when the north pole crosses this same transducer.

CAUTION: DO NOT OVER TIGHTEN MOUNTING SCREWS!!

CAUTION: The PU cord should not be grouped with any other wires or cables. For applications with PU wiring over 6 feet long, particularly in an environment, a SHIELDED CABLE is recommended. Connect the shield to the COMMON terminal of the control drive, leaving the shield at the pickup and loading.

Figure 92: Electric motor encoder datasheet

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Figure 93: Marathon electric motor dimension for spinning test platform
CERTIFICATION DATA SHEET

Model: GSK1705038

TYPICAL MOTOR PERFORMANCE DATA

<table>
<thead>
<tr>
<th>HP</th>
<th>KW</th>
<th>SYNCH RPM</th>
<th>FL RPM</th>
<th>FRAME</th>
<th>ENCLOSURE</th>
<th>KVA CODE</th>
<th>DESIGN</th>
</tr>
</thead>
<tbody>
<tr>
<td>1/2</td>
<td>0.93</td>
<td>1470</td>
<td>1750</td>
<td>112S60S2</td>
<td>65</td>
<td>SP</td>
<td>20</td>
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<tr>
<th>PH</th>
<th>VOLT</th>
<th>START TYPE</th>
<th>DUTY</th>
<th>H.P.</th>
<th>AMP</th>
<th>ELEVATION</th>
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<tbody>
<tr>
<td>3</td>
<td>230</td>
<td>60</td>
<td>2.5</td>
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<td>40</td>
<td>32</td>
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<table>
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<tr>
<th>LOAD EFF.</th>
<th>0.9264</th>
<th>LOAD EFF.</th>
<th>0.9383</th>
<th>LOAD EFF.</th>
<th>0.9383</th>
<th>ELECT TYPE</th>
<th>NO LOAD AMPS</th>
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<tbody>
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<td>0.1497</td>
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<td>0.1497</td>
<td>0.1497</td>
<td>0.1497</td>
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<table>
<thead>
<tr>
<th>FL. TORQUE</th>
<th>LOCKED ROTOR AMP</th>
<th>L.R. TORQUE</th>
<th>E.D. TORQUE</th>
<th>FL. RPM</th>
</tr>
</thead>
<tbody>
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<td>62/31</td>
<td>25</td>
<td>31</td>
<td>70</td>
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</table>

**SUPPLEMENTAL INFORMATION**

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<tr>
<th>BRACKET TYPE</th>
<th>CAGE BRACKET TYPE</th>
<th>MOUNT TYPE</th>
<th>ORIENTATION</th>
<th>SHAFT DUTY</th>
<th>SHAFT MATERIAL</th>
<th>FRAME MATERIAL</th>
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</thead>
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<tr>
<td>D1</td>
<td>01</td>
<td>L1</td>
<td>HORIZONTAL</td>
<td>FALSE</td>
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<thead>
<tr>
<th>BEARINGS</th>
<th>GREASE</th>
<th>SHAFT TYPE</th>
<th>SPECIAL CODE</th>
<th>Shaft Material</th>
<th>FRAME MATERIAL</th>
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<td>DE</td>
<td>OPE</td>
<td>01</td>
<td>NONE</td>
<td>NONE</td>
<td>STANDARD</td>
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<table>
<thead>
<tr>
<th>THERMOCOUPLE PROTECTORS</th>
<th>THERMISTORS</th>
<th>CONTROL</th>
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</thead>
<tbody>
<tr>
<td>NONE</td>
<td>NONE</td>
<td>NONE</td>
</tr>
</tbody>
</table>

**INVERTER TORQUE: NONE**

**Encoder: NONE**

**Brake: NONE**

**Date: 07/15/2018**

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Figure 94: Marathon electric motor datasheet
Appendix C: Actuation System Components Parameters

This appendix shows the parameters for different components of AMT actuation system, which were necessary for system modeling.
Figure 95: CAD plot for smallest gear in linear actuator

Figure 96: CAD plot for medium gear in linear actuator
Figure 97: CAD plot for largest gear in linear actuator

Figure 98: CAD plot of lead screw bar in linear actuator
Figure 99: Bolt plot of actuator 2

Figure 100: Configuration and dimension of L-shaped bracket lever
Figure 101: PM DC motor configuration and dimension

Parameters of components in transmission actuation system
<table>
<thead>
<tr>
<th>Different components</th>
<th>Parameters and values for actuation system</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>First gear in Actuator</strong></td>
<td>weight</td>
</tr>
<tr>
<td></td>
<td>N1(number of teeth)</td>
</tr>
<tr>
<td></td>
<td>Inner diameter</td>
</tr>
<tr>
<td></td>
<td>Exterior diameter</td>
</tr>
<tr>
<td></td>
<td>Depth</td>
</tr>
<tr>
<td><strong>Second and third gear in Actuator</strong></td>
<td>weight</td>
</tr>
<tr>
<td></td>
<td>N2(number of teeth connecting with N1)</td>
</tr>
<tr>
<td></td>
<td>N3(number of teeth connecting with largest gear N4)</td>
</tr>
<tr>
<td></td>
<td>Inner diameter</td>
</tr>
<tr>
<td></td>
<td>Exterior diameter for N2</td>
</tr>
<tr>
<td></td>
<td>Exterior diameter for N3</td>
</tr>
<tr>
<td></td>
<td>Depth for N2</td>
</tr>
<tr>
<td></td>
<td>Depth for N3</td>
</tr>
<tr>
<td></td>
<td>Small projection in the back</td>
</tr>
<tr>
<td></td>
<td>Diameter of small projection</td>
</tr>
<tr>
<td><strong>Fourth gear in Actuator</strong></td>
<td>weight</td>
</tr>
<tr>
<td></td>
<td>N4(number of teeth)</td>
</tr>
<tr>
<td></td>
<td>Inner diameter</td>
</tr>
<tr>
<td></td>
<td>Exterior diameter</td>
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<td></td>
<td>Length of rectangular hole</td>
</tr>
<tr>
<td></td>
<td>Width of rectangular hole</td>
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Table 13: Masses and dimensions of actuation system components
<table>
<thead>
<tr>
<th>Different components</th>
<th>Parameters and values for actuation system</th>
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</thead>
<tbody>
<tr>
<td></td>
<td>Diameter of Projection of the gear 0.45 inch</td>
</tr>
<tr>
<td></td>
<td>Length of projection 0.433 inch</td>
</tr>
<tr>
<td></td>
<td>Depth of the largest gear 0.1595 inch</td>
</tr>
<tr>
<td></td>
<td>Depth of the projection 0.1715 inch</td>
</tr>
<tr>
<td></td>
<td>Shallow pit of gear 0.0075 inch</td>
</tr>
<tr>
<td></td>
<td>Diameter of shallow pit 0.736 inch</td>
</tr>
<tr>
<td>Rotor of electric motor</td>
<td>Diameter of rotor 0.984 inch</td>
</tr>
<tr>
<td></td>
<td>Length of shaft 1 3.193 inch</td>
</tr>
<tr>
<td></td>
<td>Length of shaft 2 0.385 inch</td>
</tr>
<tr>
<td></td>
<td>Diameter of shaft 1 0.197 inch</td>
</tr>
<tr>
<td></td>
<td>Diameter of shaft 2 0.124 inch</td>
</tr>
<tr>
<td></td>
<td>Total weight 107.18g</td>
</tr>
<tr>
<td></td>
<td>Shaft weight 13.1177g(calculation from density)</td>
</tr>
<tr>
<td></td>
<td>Coil and rotor plate 94.0623 g</td>
</tr>
<tr>
<td>Screw-nut system</td>
<td>Screw weight 97.29g</td>
</tr>
<tr>
<td></td>
<td>Plastic nut weight 9.89g</td>
</tr>
<tr>
<td></td>
<td>Screw lead 3.071667mm/lead</td>
</tr>
<tr>
<td></td>
<td>Exterior diameter of screw 0.418 inch</td>
</tr>
<tr>
<td></td>
<td>Screw length 5.488inch</td>
</tr>
<tr>
<td></td>
<td>Cylinder length 1.569 inch</td>
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</table>

Continued
Table 13 Continued

<table>
<thead>
<tr>
<th>Different components</th>
<th>Parameters and values for actuation system</th>
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</thead>
<tbody>
<tr>
<td>Diameter of cylinder</td>
<td>0.315 inch</td>
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<tr>
<td>Tube weight</td>
<td>50.67g</td>
</tr>
<tr>
<td>Cable 1 weight</td>
<td>784.058g</td>
</tr>
<tr>
<td>Cable 2 weight</td>
<td>786.328g</td>
</tr>
<tr>
<td>Pin weight with three washer</td>
<td>21.3g</td>
</tr>
<tr>
<td>Pin weight with two washer</td>
<td>18.55g</td>
</tr>
<tr>
<td>Actuator 1 housing lever weight</td>
<td>52.065g</td>
</tr>
<tr>
<td>Name</td>
<td>Inertia</td>
</tr>
<tr>
<td>-----------------------------</td>
<td>----------------------------------------------</td>
</tr>
<tr>
<td>Gear set</td>
<td>First gear  $2.539 \times 10^{-5} g \cdot m^2$</td>
</tr>
<tr>
<td></td>
<td>Second and third gear  $6.5908 \times 10^{-4} g \cdot m^2$</td>
</tr>
<tr>
<td></td>
<td>Fourth gear  $1.42258 \times 10^{-3} g \cdot m^2$</td>
</tr>
<tr>
<td>Screw drive inertia</td>
<td>$1.28783 \times 10^{-3} g \cdot m^2$</td>
</tr>
<tr>
<td>Screw nut equivalent inertia</td>
<td>$2.361 \times 10^{-6} g \cdot m^2 / 0.296637 = 3.63245 \times 10^{-6} g \cdot m^2$</td>
</tr>
<tr>
<td>Tube equivalent inertia</td>
<td>$1.2097 \times 10^{-5} g \cdot m^2 / 0.296637 = 1.8611 \times 10^{-5} g \cdot m^2$</td>
</tr>
<tr>
<td>Electric motor rotor inertia</td>
<td>$7.64628 \times 10^{-3} g \cdot m^2$</td>
</tr>
<tr>
<td>Total leads of screw drive</td>
<td>42.7 circles</td>
</tr>
<tr>
<td>Total lead length</td>
<td>5.139 inch</td>
</tr>
<tr>
<td>Screw-drive efficiency</td>
<td>Acme-plastic nut: 0.296637</td>
</tr>
<tr>
<td>Inertia of box lever</td>
<td>$0.502617 g \cdot m^2$</td>
</tr>
<tr>
<td>Cable 1 equivalent inertia</td>
<td>$3.898 \times 10^{-4} g \cdot m^2$</td>
</tr>
<tr>
<td>to electric motor</td>
<td></td>
</tr>
<tr>
<td>Transmission shifting lever mass</td>
<td>$1.86444 kg$</td>
</tr>
<tr>
<td>Transmission shifting lever mass equivalent inertia to actuator 1</td>
<td>$1.07 \times 10^{-5} kg \cdot m^2$</td>
</tr>
<tr>
<td>Transmission shifting lever inertia</td>
<td>$5.822 \times 10^{-3} kg \cdot m^2$</td>
</tr>
</tbody>
</table>

Table 14: Inertias of actuation system components