國立交通大學

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碩士論文

自手排傳動系統之動態模擬及最佳化

Dynamic Modeling and Optimization of Automated Manual Transmission System

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摘要

自手排傳動系統(Automated Manual Transmission System, AMT)近年來由於低成本、高效率、及方便性的優點，已開始迅速的發展並被廣泛的運用。目前市售車輛中，從BMW M3、Ferrari 355、Alfa Romeo 156之類的高級跑車，到Opel Corsa、Renault Twingo之類的平價車款，都已採用此類傳動系統。然而自手排傳動系統換檔時離合器的控制較難達到與傳統自排系統(Automated Transmission, AT)扭力轉換器相比擬的舒適性，是目前自手排傳動系統尚無法完全取代傳統自排系統的主因。

為了改善自手排傳動系統換檔時的不舒適性，本文以自手排傳動系統中離合器的控制為主要研究目標。本論文主要包含三個部分: 動態模型的建立、控制方程式的設計、及最佳化設計。在動態模型的建立上，主要是建立AMT傳動系統由引擎、離合器、離合器致動器、變速箱、到車輛負載的動態模型，藉以模擬AMT系統在換檔時的動態表現。由於本研究與工研院(ITRI)機械所先進車輛動力組合作，協同進行工研院所開發離合器致動器原型的修改，因此整個動態模型的建立特別著重於離合器致動器。在控制方程式的设计中，則針對所建立出動態模型的特性，再藉由控制法則及參數調整理論，設計出整個離合器致動器的控制器的數學模型，藉以模擬整個傳動系統由控制指令到動態表現的完整行為。在最佳化的部分之中，則藉由最佳化原理更改原本離合器致動器部分零件的設計及控制方程式中的參數，使換檔時離合器的離合與分離有更符合設計需求的表現。
Dynamic Modeling and Optimization of Automated Manual Transmission System

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ABSTRACT

Automated Manual Transmission (AMT) system is gradually prevailed in recent years since it has high transmission efficiency, auto shifting capability, and low cost. Vehicles from high quality sports cars like BMW M3, Ferrari 355, Alfa Romeo 156, etc., to general sedans like Opel Corsa, Renault Twingo, etc. have introduced such system. However, the key point that AMT can’t completely substitute Automated Transmission (AT) system is the smoothness that a auto clutch system on AMT can’t transmit power as smooth as a torque converter on AT.

Clutch control of AMT system is the main subject in this study, which includes three objectives: dynamic modeling, control function design, and optimization. The first objective is to create a dynamic model of AMT system, which includes engine, clutch, clutch actuator, gear box, and vehicle loading. Clutch actuator is focused in this model, since this study is a project cooperating with Industrial Technology Research Institute (ITRI) to modify a prototype of clutch actuator. The second objective uses the dynamic model created before to design a control function for the clutch actuator, some control methods and parameter turning methods are introduced here. According to the control model and dynamic model, a complete
shifting process from controller to vehicle dynamic is simulated in this study. The third objective is optimization, some parts in the clutch actuator and some parameters of the control function are optimized to give a better performance on clutch engagement and disengagement.
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NOTATIONS

\( a_x \) \hspace{1cm} \text{longitudinal car acceleration}

\( b \) \hspace{1cm} \text{setpoint weighting of proportional term in PID control}

\( B \) \hspace{1cm} \text{magnetic flux density}

\( c \) \hspace{1cm} \text{setpoint weighting of derivative term in PID control}

\( C_{BL} \) \hspace{1cm} \text{damping coefficient from release bearing to ball and socket joint}

\( C_c \) \hspace{1cm} \text{damping coefficient of virtual spring on worm gear collision}

\( C_{CP} \) \hspace{1cm} \text{damping coefficient of clutch disc}

\( C_{DM} \) \hspace{1cm} \text{damping coefficient of identity shaft}

\( C_{\text{eng}} \) \hspace{1cm} \text{damping coefficient of engine}

\( C_G \) \hspace{1cm} \text{damping coefficient of worm gear}

\( C_{IG} \) \hspace{1cm} \text{damping coefficients on input shaft}

\( C_{OG} \) \hspace{1cm} \text{damping coefficients on output shaft of gear box}

\( C_{RG} \) \hspace{1cm} \text{damping coefficients on inverse shaft}

\( C_{ss} \) \hspace{1cm} \text{damping coefficient of stage springs}

\( C_{SP} \) \hspace{1cm} \text{damping coefficient of assist spring}

\( C_{wh} \) \hspace{1cm} \text{identity damping coefficient on wheels}

\( C_{WS} \) \hspace{1cm} \text{damping coefficient of worm shaft}

\( d_G \) \hspace{1cm} \text{pitch diameter of worm gear}
\( d_w \)  
pitch diameter of worm shaft

\( D_A \)  
aerodynamic drag force

\( D_{out} \)  
clutch actuator travel

\( f_r \)  
rolling resistance coefficient

\( F \)  
force caused by current and magnetism

\( F_x \)  
longitudinal forward force on car

\( g \)  
gravity

\( i \)  
current in conductor

\( I_{CP} \)  
inertia moment of clutch disc

\( I_{DM} \)  
identity inertia moment of final shafts

\( I_{eng} \)  
inertia moment of engine

\( I_G \)  
inertia moment of worm gear

\( I_{IG} \)  
inertia moments of input shaft

\( I_L \)  
inertia moment of linkage arm at mass center

\( I_{OG} \)  
inertia moments of output shaft of gear box

\( I_{RG} \)  
inertia moments of inverse shaft

\( I_{ss} \)  
inertia moment of stage springs

\( I_{WS} \)  
inertia of worm shaft

\( J_m \)  
inertia of motor rotor
$K$ proportional integer in PID controller

$K_s$ spring coefficient of virtual spring on worm gear collision

$K_e$ EMF (electromotive force) constant

$K_{ss}$ stiffness coefficient of stage springs

$K_{SP}$ stiffness of assist spring

$K_t$ torque constant on DC motor

$l$ length of conductor

$I_{wk}$ identity inertia moment of the total wheels

$La$ inductance of armature

$m_{BL}$ simplified mass of release bearing and clutch lever

$m_G$ mass of worm gear

$M_{car}$ car mass

$M_{SP}$ mass of assist spring and fix plate

$N$ derivative limit constant in PID control

$N_w$ thread number of worm shaft

$p_x$ axial pitch of worm shaft

$r_{GG}$ distance from mass center to axle on worm gear

$R_a$ resistance of armature

$R_{DMR}$ final gear ratio

xix
\( R_G \quad \text{gear box transmission ratio} \)
\( R_{hs} \quad \text{hitch forces} \)
\( Ri \quad \text{inner radius of clutch disc} \)
\( Ro \quad \text{outer radius of clutch disc} \)
\( R_r \quad \text{clutch lever ratio} \)
\( R_s \quad \text{rolling resistance forces} \)
\( s_{ins} \quad \text{pre-deformation of assist spring} \)
\( S_{LC} \quad \text{motion of ball and socket joint in } X \text{ direction} \)
\( S_{Lx} \quad \text{displacement of mass center of the linkage arm in } x \text{-direction} \)
\( S_{Ly} \quad \text{displacement of mass center of the linkage arm in } y \text{-direction} \)
\( T \quad \text{torque caused by current and magnetism} \)
\( T_{cf} \quad \text{torque able to be transfer from clutch} \)
\( T_d \quad \text{derivative integer in PID controller} \)
\( T_{DM} \quad \text{torque transmitted to the drive wheels} \)
\( T_{engR} \quad \text{engine output torque} \)
\( T_{GD} \quad \text{torque from output shaft of gear box to output shaft of differential mechanism} \)
\( T_{Gf} \quad \text{torque caused by friction on the axle of worm gear} \)
\( T_i \quad \text{integral integer in PID controller} \)
\( T_{IG} \)  
\text{torque transmitted from clutch}

\( T_{IR} \)  
\text{torque transmitted to inverse shaft in gear box}

\( T_I \)  
\text{extra torque loading on motor}

\( T_{LCf} \)  
\text{friction torque on ball and socket joint}

\( T_{LGf} \)  
\text{torque caused by friction on the joint of worm gear}

\( T_{OR} \)  
\text{reacting torque from output shaft to the inverse shaft in gear box}

\( T_{OG} \)  
\text{torque transmitted to differential mechanism}

\( T_{RG} \)  
\text{torque from inverse shaft to output shaft in gear box}

\( T_{\text{syn}} \)  
\text{torque generated by synchronizer}

\( v_a \)  
\text{supplied voltage on armature}

\( W \)  
\text{force exerted from worm gear to worm shaft}

\( W_c \)  
\text{reacting force of clutch}

\( W_{\text{car}} \)  
\text{car force}

\( W_{Ga} \)  
\text{axial force on worm gear}

\( W_{Gr} \)  
\text{radial force on worm gear}

\( W_{Gi} \)  
\text{tangential force on worm gear}

\( W_{Gx} \)  
\text{axle force in X direction on worm gear}

\( W_{Gy} \)  
\text{axle force in Y direction on worm gear}
\( W_{LCx} \) force on ball and socket joint in x-direction

\( W_{LCy} \) force on ball and socket joint in y-direction

\( W_{LF} \) force of assist spring

\( W_{LGx} \) force from linkage to worm gear on x-direction

\( W_{LGy} \) force from linkage to worm gear on y-direction

\( W_{Wa} \) axial force on worm shaft

\( W_{Wyf} \) tangential friction force on worm shaft

\( W_{Wyr} \) radial force on worm shaft

\( W_{Wy} \) tangential force on worm shaft

\( \eta_{eng} \) transmission efficiency of powertrain

\( \theta_c \) overtaking angle of lower or upper limit on worm gear

\( \theta_{DM} \) rotation angle of identity shaft

\( \theta_{eng} \) engine rotation angle

\( \theta_G \) rotation angle of worm gear

\( \theta_{IG} \) rotation angle of input shaft

\( \theta_L \) rotation angle of linkage arm

\( \theta_m \) rotation angle of rotor

\( \theta_{OG} \) rotation angle of output shaft of gear box
\(\theta_{RG}\)  
rotation angle of inverse shaft

\(\theta_{ws}\)  
rotation angle of worm shaft

\(\lambda\)  
lead angle of worm gear

\(\mu\)  
friction coefficient between worm shaft and worm gear

\(\mu_{clutch}\)  
clutch disc friction coefficient

\(\mu_{Ga}\)  
friction coefficient on axle of worm gear

\(\mu_{Gj}\)  
friction coefficient on joint of worm gear

\(\mu_{LC}\)  
friction coefficient of ball and socket joint

\(\xi\)  
damping ratio in control function

\(\rho\)  
air density

\(\phi_n\)  
pressure angle of worm shaft
CHAPTER 1
INTRODUCTION

1.1 Automated Manual Transmission

Vehicle transmission systems are typically sorted into three types: manual transmission (MT), continuously variable transmission (CVT), and automated transmission (AT). Manual transmission is the most historical transmission system which has very high reliability, high transmission efficiency, and low cost. However, the operation of manual transmission is more complicated comparing to continuously variable transmission and automated transmission, that driver should control throttle pedal, clutch pedal, and shifting stick. Continuously variable transmission offers a continuum of infinite gear ratio and adjusts it automatically. In which ratio spectrum, it can increase the overall powertrain efficiency and eliminate the unknown jerks associated with manual and automated transmissions. On the other hand, it has difficulty with high torque and low speed transmission requirements [1]. Automated transmission system, likes the name it is, operates with no requirement for driver to control the shifting stick or clutch pedal, and has the ability to transmit larger torque than CVT. Torque converter on automated transmission system provides a smooth coupling between engine and gear box. However, the slippage within the torque converter, which is an inherent character, results in plastic losses, thereby decreases the efficiency of transmission. Furthermore, the shifting operation in automated transmission requires hydraulic pumps which pressurize fluid for gears to shift and torque converter to engage. The power required to pressurize the fluid introduces an additionally parasitic losses in powertrain [2][3].

Automated manual transmission system is gradually developed in recent years. It combines both the advantages of manual transmission and automated transmission. An automated manual transmission is essentially a manual transmission. Engine power is transmitted from engine shaft to gear box by a clutch coupling with the two shafts between
engine and gear box, and synchronizers are utilized to shift gear ratios. However, the control of clutch and synchronizer are substituted by clutch actuator and shifting actuator in automated manual transmission. The actuators are controlled by electrical control unit (ECU). In such way, automated manual transmission works like automated transmission, that driver only needs to decide the working conditions of park, reverse, neutral, or drive. Besides, automated manual transmission is able to transmit power with higher torque, higher efficiency, and less fuel cost comparing to automated transmission since no hydraulic component like torque converter and hydraulic pumps are used. And the transmission efficiency is even higher than manual transmission. Because it always works in a condition with higher system efficiency controlled by a computer. Furthermore, the cost is lower than both CVT and AT. A comparison of auto-shifting transmission systems is shown in Table 1.1-1, and structure of automated manual transmission system is shown in Figure 1.1-1.

<table>
<thead>
<tr>
<th></th>
<th></th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>Comfort</td>
<td>good</td>
<td>very good</td>
<td>middle</td>
</tr>
<tr>
<td>Reliability</td>
<td>good</td>
<td>bad</td>
<td>good</td>
</tr>
<tr>
<td>Max. Torque</td>
<td>very high</td>
<td>&lt;250 Nm</td>
<td>very high</td>
</tr>
<tr>
<td>Transmission Efficiency</td>
<td>85%</td>
<td>90%</td>
<td>&gt;93%</td>
</tr>
<tr>
<td>Mechanical Structure</td>
<td>high</td>
<td>middle high</td>
<td>middle</td>
</tr>
<tr>
<td>Control System</td>
<td>Middle high</td>
<td>high</td>
<td>high</td>
</tr>
<tr>
<td>Starting Years</td>
<td>1930’s</td>
<td>1970’s</td>
<td>1990’s</td>
</tr>
<tr>
<td>Design Cost</td>
<td>1</td>
<td>1</td>
<td>&lt;0.75</td>
</tr>
<tr>
<td>Manufacturing Cost</td>
<td>1</td>
<td>1</td>
<td>&lt;0.75</td>
</tr>
</tbody>
</table>

Table 1.1-1 Comparison of AT, CVT, and AMT [4]
In these years, automatic transmission prevails in Japan, North America, and many Asia countries, manual transmission cars are main stream in Europe. However, according to Montreal Protocol 1987, Rio de Janeiro Protocol 1992, and Kyoto Protocol on Global Warming 1998, 38 developed countries are to reduce greenhouse gas by approximately 95 percent of 1990 levels by 2008~2012, and the United States is to lower its discharge of carbon dioxide (CO₂) to 93 percent of 1990 emission [5]. An astringent CO₂ regulation is required on vehicle gradually in many countries. For example: Europe, as shown in Figure 1.1-2. Automated manual transmission which has higher transmission efficiency and lower gas emission has more superiority to grow up to meet this tendency in recent years than many others [7]. Many main factories like TOYOTA, DaimlerChrysler, BMW, ZF, GETRAG, SEIKI, LUK, VALEO, etc. have invested in such project. A forecast from Industrial Technology Research Institute (ITRI) is shown in Table 1.1-2. Where DCT is double clutch transmission
system, an evolution of automated manual transmission system.

<table>
<thead>
<tr>
<th>Transmission System in European Markets</th>
<th>2000</th>
<th>2010</th>
</tr>
</thead>
<tbody>
<tr>
<td>AT</td>
<td>11.9%</td>
<td>25%</td>
</tr>
<tr>
<td>CVT</td>
<td>0.5%</td>
<td>5%</td>
</tr>
<tr>
<td>MT</td>
<td>87%</td>
<td>45%</td>
</tr>
<tr>
<td>AMT</td>
<td>0.5%</td>
<td>12%</td>
</tr>
<tr>
<td>DCT</td>
<td>0%</td>
<td>13%</td>
</tr>
</tbody>
</table>

Table 1.1-2 Forecast of Transmission Systems in Europe

![](acea.png)

**ACEA voluntary agreement on CO₂**

CO₂ emissions (g/km)

<table>
<thead>
<tr>
<th>Year</th>
<th>ACEA fleet</th>
<th>ACEA:</th>
</tr>
</thead>
<tbody>
<tr>
<td>1990</td>
<td></td>
<td>- 9 % since 1995 (185 → 169 g/km)</td>
</tr>
<tr>
<td>1995</td>
<td></td>
<td>-10.8 % diesel (157 g/km)</td>
</tr>
<tr>
<td>2000</td>
<td></td>
<td>- 9.0 % petrol (171 g/km)</td>
</tr>
<tr>
<td>2005</td>
<td></td>
<td></td>
</tr>
<tr>
<td>2010</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

Figure 1.1-2 European Automotive Manufacturers Association (ACEA) voluntary agreement on vehicle CO2 emission [6]
1.2 Objectives

Automated manual transmission has high transmission efficiency, high torque capacity, low gas emission, and low cost; however, the main shortcoming is the amenity which is inferior to both automated transmission and continuously variable transmission. Continuously variable transmission is essentially a system transmits power continuously with no interrupt or shifting process, it has the best amenity than any others. Automated transmission transmits power through a torque converter. The torque converter is acted to interrupt the power transmission while shifting. But due to the property of hydraulic pressure applied, the power transmitted is not fully interrupted while shifting, thus shifting with low shocks. Automated manual transmission, like manual transmission, shifting process is introduced by clutch disengagement/engagement and synchronizer movement. Clutch disengagement is to provide an occasion of power interruption for synchronizer to shift to next effective gear ratio. And clutch engages after the shifting process is completed. In general, the processes of clutch disengagement and engagement cause the power transmitted to car a termination and a restoration. This interruption sometimes produces uncomfortable driving condition if the control of clutch is not skilled. Such condition can be seen obviously when a freshman drives a manual transmission vehicle.

Since the clutch control is handled by a clutch actuator mastered by computer algorithm in automated manual transmission, the amenity can be modified better than manual transmission. Clutch control dominates the amenity of a car while shifting, which is the crucial point of automated manual transmission. There are many strategies to increase the smoothness of automated manual transmission while shifting, like double clutches [8] used on mechanical structures and torque tracking method [9] for control algorithm. However, all these strategies, which improve the performance of automated manual transmission, should base on a clutch actuator and a controller which work well as our expectation. Moreover, the
vehicle dynamic characters influenced by shifting process should be understood clearly.

In this study, dynamic analysis of a clutch actuator prototype designed by Industrial Technology Research Institute is the main objective. According to this model, an expanded model of automated manual transmission system, including engine, clutch, gear box, and car loading, is built up to simulate complete vehicle dynamic characters while shifting. Besides dynamic modeling, control functions based on system dynamic characters is proposed.

Another objective in this study is optimization. Both mechanical parts on clutch actuator and control parameters for clutch control function are do be modified. In this part, using optimization methods and models created before, a new evolution of clutch control system which displays speedier and more stable is obtained. These solutions of optimizations give the prototype guidelines of modification.

With the objectives in this study, dynamic models of both the clutch actuator and the whole transmission system should be created in modules. In such way, the models not only give simulation results of the current job, but also provide a basic frame for simulation of many other transmission systems. The model should be able to be modified easily to simulate future developments like double clutches system, torque tracking controller, hybrid vehicle transmission, and some other advance transmission systems.

1.3 Thesis Outlines

There are two main components in this study according to the automated manual transmission system: mechanical component and control component. Besides model creation, computer simulation, and analysis of these two components, optimizations are introduced after discussions of both components are satisfied to provide improvements of the prototype. Both these studies are described from CHAPTER 2 to CHAPTER 5.
CHAPTER 2 and CHAPTER 3 are parts of mechanical component. CHAPTER 2 proposes the dynamic analyses and dynamic model creations of the clutch actuator and the powertrain. With free body analyses and experiment data, the dynamic characters are obtained into system equations. Finally, using Matlab Simulink, the dynamic models according to the system equations are created into modulus.

The dynamic models created in CHAPTER 2 are simulated in CHAPTER 3. The results according to the simulations are analyzed with tendency forecasting to check the exactitude. Finally in this chapter, some parts of the clutch actuator are modified by optimization method. In this section, the parts which are easy to be modified practically and capable of increasing performance are chosen to be design variables. Target and constraints of the optimization are defined according to the design requirements.

CHAPTER 4 and CHAPTER 5 are parts of control component. CHAPTER 4 is the development of control function. According to CHAPTER 2 and CHAPTER 3, which introduce the dynamic characters of the clutch actuator, the control function is created and the parameters are defined according to some control turning methods.

CHAPTER 5 optimizes the parameters of the control function created in CHAPTER 4. In this chapter, the parameters are optimized according to the dynamic models created before. Since the parameters have had been defined in CHAPTER 4 with some control turning methods, this chapter uses these parameters as an initial condition and expects a better performance after optimization. Optimization in this chapter deals with a complete automated manual transmission system model, which combines the dynamic models and the control models created before. With the integrated system model, the control functions and dynamic characters of the transmission system are merged with optimized combination. Finally in CHAPTER 5, the simulation results of the complete merged models are presented.
As a final point, CHAPTER 6 outlines the conclusions and direction of the future works.
CHAPTER 2
SYSTEM DYNAMIC MODELING

2.1 Introduction

In this chapter, the dynamic characters of automated manual transmission system are analyzed, and the dynamic models from system equations are created in Matlab® Simulink.

Since one of the objectives in this study is to assist Industrial Technology Research Institute (ITRI) to analyze and modify the prototype of a new clutch actuator, the dynamic analysis on clutch actuator is focused with more details.

There are two main methods for dynamic model creation in this chapter. With components that should be concerned with details or dynamic characters are commonplace, free-body analysis is applied. For example: all the structures in clutch actuator are analyzed with free body analysis. On the other hand, components that are more complicated and free body analyses are complex that are not worth for analyzing, or components that are not the emphasis of this study, are seen as black boxes, such as engine, gear box, clutch, etc.. In such black boxes, experiments data and curve fitting methods are used to create system equations that stand for dynamic characters of these components, and dynamic models are similarly created in Simulink® according to these system equations.

In the free body analyses, in order to simplify the complexity of the models, two main assumptions are supposed. First, all the components are seen as rigid bodies besides springs, since it is well approximated in most cases on vehicle [19]. Second, the gravity effects are ignored. These assumptions are hold in general cases. However, if the hypotheses are not held that errors caused by these assumptions are out of our capable range after a comparison with experiments data, the hypotheses should be modified in future works.
In this chapter, clutch actuator is specially analyzed in section 2.2, and the powertrain of automated manual transmission is analyzed in section 2.3, finally in section 2.4, the system dynamic equations developed in section 2.2 and section 2.3 are built up into Matlab® Simulink models.

2.2 Dynamic Analysis of Clutch Actuator

The clutch actuator which is a prototype of Industrial Technology Research Institute is analyzed with dynamic theories in this section. At first, structures of the clutch actuator are introduced and decomposed into several parts. The analyses and system equation creation are progressed according to these parts.

2.2.1 Structure of Clutch Actuator

The structure of the clutch actuator is shown in Figure 2.2-1. The deeply blue part is a DC electrical motor with a worm gear shaft in red assembled on the armature. The worm gear shaft transmits power from electrical motor to worm gear, which is drawn in brown. The worm gear shaft and the worm gear work like a deceleration system and torque amplify system for the electrical motor. On the worm gear, a joint structure is used to connect the worm gear with a linkage structure drawn in black. The linkage is driven by the worm gear throw the joint structure. At the other end of the linkage is a ball and socket joint coupling with a lever in yellow used to drive the clutch with a lever ratio through the clutch lever in yellow. Near the ball and socket joint is a fix plate in green connected with the linkage through a circular furrow. On the left of the fix plate is a spring in blue, which is usually in compressed status to assist the movement of the fix plate to drive the linkage to disengage clutch. Therefore, the worm gear, linkage, and fix plate can work almost like a linkage-slider structure since the travel of the actuator output is very small.
The clutch actuator drives the clutch through the clutch lever as shown in Figure 2.2-2. Clutch actuator has the maximum movement of 8mm in the design. The lever has a length of 65mm. According to Figure 2.2-2, the rotation angle $\theta$ is no more than 0.1077 degree in general operation. Since $\theta$ is small, the travel of clutch actuator can be seen as the horizontal motion and the displacement is the same as the clutch travel according to the lever ratio of one.
According to the structure, the clutch actuator is decomposed into four subsystems: electrical motor, worm shaft, worm gear, and linkage system. In the following sections, dynamic analyses of each part are introduced according to these subsystems.

### 2.2.2 Electrical Motor

The electrical motor here is used to drive the clutch actuator on AMT vehicle, since most vehicles use direct current (DC) as their electric power, the model here should be created as a DC motor as used in the prototype. Figure 2.2-3 shows the common structure of a DC motor. In addition to housing and bearings, the nonturning part (stator) has magnets, which establish
a magnetic field across the turning part (rotor). The brushes force current through the wire wound around the rotor. The (rotating) commutator causes the current always to be sent through the armature.

**Figure 2.2-3 DC Motor [10]**

In mechanical component, the torque developed by DC electrical motor is introduced by the force caused by magnetic flux. The relation of force and current is known as “law of motor”, which can be expressed as [11]:

\[ F = Bli \ \text{Newton} \quad (2.2-1) \]

Where \( B \) is the magnetic field strength, \( l \) is the length of conductor, \( i \) is the current in conductor, and \( F \) is the force caused by current and magnetism. Hence the torque cause by \( F \) is:

\[ T = Bli \ \text{Newton} \cdot \text{meter} \quad (2.2-2) \]

Where \( T \) is the torque caused by current and magnetism.
Applying to free body diagram of motor rotor, as shown in Figure 2.2-4, the relation between current and torque can be obtained as shown in Eq.(2.2-3):

\[ J_m \ddot{\theta}_m + b \dot{\theta}_m + T_l = K_i i = T \]  \hspace{1cm} (2.2-3)

Where \( J_m \) is the inertia of rotor, \( b \) is the damping coefficient effects on the rotor, \( \theta_m \) is the rotation angle of rotor, \( T_l \) is the extra loading torque on motor, and \( K_i \) is called torque constant, which can be expressed as:

\[ K_i = B l r \]  \hspace{1cm} (2.2-4)

In electrical component, Figure 2.2-5 shows electrical circuit of the armature. According to Kirchoff Voltage Law (KVL) loop analysis, the relationship between current and voltage on DC motor can be stated as Eq.(2.2-5).
Where $L_a, R_a,$ and $v_a$ indicate inductance, resistance and source voltage of the electrical circuit in DC motor. $K_e$ is called “EMF” (Electro Motive Force) constant which is decided by motor character. $K_e \theta_m$ is the term of counter EMF introduced by armature rotation that causes motor working simultaneously like a generator generating voltage opposing the supplied voltage $v_a$.

According to Eq.(2.2-3) and Eq.(2.2-5), the system equations of the DC electrical motor on clutch actuator are obtained, which has an system input $v_o$, an system output $\theta_o$, an outside loading $T_l$, and other system parameter terms.

### 2.2.3 Worm Shaft

The structure of worm shaft is shown in Figure 2.2-6. Worm shaft is connected with electrical motor armature at one end and transmits torque from motor to worm gear, as shown
in Figure 2.2-7.

Figure 2.2-6 Structure of Worm Shaft [13]
Figure 2.2-8 is free body diagram of the worm shaft coupling with a worm gear not shown. \( \phi_n \) is pressure angle, and \( \lambda \) is lead angle of the worm shaft, which has a relation with pitch diameter \( d_w \): \( \tan \lambda = \frac{L}{\pi d_w} \), \( L = p_x N_w \). Where \( p_x \) is axial pitch of worm shaft and \( N_w \) is thread number of the worm shaft [13]. The force exerted by the worm gear is \( W \), and the torque transmitted from motor is \( T \). \( W \) has three orthogonal components \( W^x \), \( W^y \), and \( W^z \). From geometry of Figure 2.2-8, the three terms can be expressed as Eqs.(2.2-6).

\[
\begin{align*}
W^x &= W \cos \phi_n \sin \lambda \\
W^y &= W \sin \phi_n \\
W^z &= W \cos \phi_n \cos \lambda
\end{align*}
\]  
\text{(2.2-6)}
Figure 2.2-8 Free Body Diagram of Worm Shaft [13]

\( W^y \) is the separating, or radial, force for both worm shaft and worm gear, which is known as \( W_{w_r} \) and \( W_{gr} \). The axial force on worm shaft is \( W^z \) known as \( W_{wa} \), and tangential force \( W_{ga} \) on worm gear, since the shaft angle here is 90°. The tangential force which is a workable force to be transmitted on the worm shaft is \( W^z \) known as \( W_{wy} \), and on worm gear is an axial force known as \( W_{ga} \).

From Newton’s third law, worm gear forces are opposite to the worm shaft forces, which can be summarized as Eqs.(2.2-7).

\[
\begin{align*}
W_{w_z} &= -W_{ga} = W^x \\
W_{w_r} &= -W_{gr} = W^y \\
W_{wa} &= -W_{gi} = W^z
\end{align*}
\]  
\( (2.2-7) \)
Since the relative motion between worm shaft and worm gear teeth are pure sliding, friction plays and important role in the transmission of worm gearing. By introducing a coefficient of friction $\mu$, from Figure 2.2-8, the force $W$ acting normal to the worm gear tooth produces a friction force $W_f = \mu W$, having a component $\mu W \cos \lambda$ in the negative $x$ direction and another component $\mu W \sin \lambda$ in the positive $z$ direction. Eqs.(2.2-6) therefore becomes Eqs.(2.2-8) which are considered with friction affection.

\[
W^x = W (\cos \phi_n \sin \lambda + \mu \cos \lambda) \\
W^y = W \sin \phi_n \\
W^z = (W \cos \phi_n \cos \lambda - \mu \sin \lambda)
\]  
(2.2-8)

From Eqs.(2.2-7) and Eqs.(2.2-8), the relation between pure tangential forces on worm shaft and worm gear, which is effective term on torque transmission, can be summarized as Eq.(2.2-9).

\[
W_{wt} = W_{GI} \frac{\cos \phi_n \sin \lambda}{\mu \sin \lambda - \cos \phi_n \cos \lambda}
\]  
(2.2-9)

And the friction term affects on tangential direction can be expressed as:

\[
W_{wtf} = \mu W_{GI} \frac{\cos \lambda}{\mu \sin \lambda - \cos \phi_n \cos \lambda}
\]  
(2.2-10)

Were $W_{wt}$ is the tangential force caused by friction between worm shaft tooth and worm gear tooth.

The terms of transmission torque and friction effect are separated here because of the convenience for further model creation, which in general can be expressed as [13]:

\[
W_{st} = W_{GI} \frac{\cos \phi_n \sin \lambda + \mu \cos \lambda}{\mu \sin \lambda - \cos \phi_n \cos \lambda}
\]  
(2.2-11)

For a simplified free body diagram as shown in Figure 2.2-9, the dynamic characters can
be expressed as a system equation as shown in Eq.(2.2-12), which has an input torque $T$ transmitted from electrical motor that the same with motor load $T_l$, an input loading torque from worm gear $W_{Gt}$, and an output of worm shaft rotation angle $\theta_{ws}$.

$$ T = T_l = W_{wt} \frac{d_w}{2} + W_{wt} \frac{d_w}{2} + I_{ws} a_{ws} + C_{ws} \theta_{ws} $$  \hspace{1cm} (2.2-12)

Where $d_w$ is pitch diameter of the worm shaft, $\theta_{ws}$ is rotation angle of the worm shaft, $I_{ws}$ is inertia moment of the worm shaft, $C_{ws}$ is the damping coefficient of the worm shaft, and $a = \dot{\theta}_{ws}$ as in Figure 2.2-9.

![Figure 2.2-9 Simplified Free Body Diagram of Worm Shaft](image)

**2.2.4 Worm Gear**

In the clutch actuator, worm gear couples with the worm shaft discussed in above subsection, which decelerates rotation speed and amplifies transmission torque of worm shaft. Besides, it also transforms rotation torque to a force which drives the linkage through a joint. As shown in Figure 2.2-10.
Free body diagram of worm gear is shown in Figure 2.2-11. Since structures here are supposed to be rigid bodies as mentioned before, free body analyses here are considered as a 2-D problem, where no deformation, which may change forces direction out of original plan, occurs. The moments on X and Y directions caused by the worm shaft and the uncommon plan force from linkage joint thus are assumed to be influence-less.
From Figure 2.2-11, considering the influences of inertia moment, damping, friction and forces that act on the worm gear, the system dynamic equations according to the equilibrium of forces and moments are obtained as Eqs.(2.2-13).

\[
\sum F_x = 0; \\
W_{LGx} + W_{Gr} + m_G \theta_G r_{GG} \cos \theta_{GL} = W_{Gr}
\]

\[
\sum F_y = 0; \\
W_{Gx} + W_{LGy} + W_{Gr} + m_G \theta_G r_{GG} \sin \theta_{GL} = 0
\]

\[
\sum M = 0; \\
W_{Gx} R_G + W_{LGy} r_G \cos \theta_{GL} = W_{LGx} r_G \sin \theta_{GL} + T_{LGF} + T_{Gf} + I_G \ddot{\theta}_G + C_G \dot{\theta}_G
\]

Where \( m_G \) is the mass, \( I_G \) is the inertia moment, \( C_G \) is the damping coefficient, \( \theta_G \) is the rotation angle, and \( r_{GG} \) is the distance from mass center to axle of the worm gear. \( W_{LGx} \) and \( W_{LGy} \) are reacting forces from linkage in X and Y directions. \( W_{Gx} \) and \( W_{Gr} \) are forces on the axle of the worm gear in the directions of X and Y. \( W_{Gt} \) and \( W_{Gr} \) are tangential force and radial force on worm gear as mentioned in Eqs.(2.2-7). By rearrangement of Eqs.(2.2-7), Eqs.(2.2-8), and Eq.(2.2-11), \( W_{Gr} \) can be expressed by \( W_{Gt} \), as shown in Eq.(2.2-14).

\[
W_{Gr} = \frac{W_{Gt} \sin \phi_n}{\cos \phi_n \cos \lambda - \mu \sin \lambda}
\]

\( T_{Gf} \) and \( T_{LGF} \) are torques that caused by frictions on axle of worm gear and joint used to connect linkage.

Let the friction coefficient on axle of the worm gear be \( \mu_{Ga} \), on the joint be \( \mu_{Gj} \). The torques \( T_{Gf} \) and \( T_{LGF} \) caused by friction can be expressed as:
\[ T_{Gj} = \sqrt{W_{Gx}^2 + W_{Gy}^2 \mu_{Ga} R_{GP}} \]
\[ T_{LGj} = \sqrt{W_{LGx}^2 + W_{LGy}^2 \mu_{Gj} R_{GJ}} \]  

(2.2-15)

From Eqs.(2.2-13), Eq.(2.2-14) and Eqs.(2.2-15), the system dynamic equation of the worm gear can be rearranged as Eqs.(2.2-16). In which equation, outside influences are known as forces \( W_{LGx}, W_{LGy}, \) and \( W_{Gz} \), which cause an output system of rotation angle known as \( \theta_G \). And parameters \( I_G, C_G, R_G, r_G, \theta_{GL}, \phi_n, \lambda, \mu_{Ga}, \mu_{Gj}, R_{GP}, r_{GG}, m_G \) and \( R_{GJ} \) are system characteristics defined by original design as shown in Figure 2.2-11.

\[
I_G \ddot{\theta}_G + C_G \dot{\theta}_G = (W_{Gx} R_G + W_{LGy} r_G \cos \theta_{GL} - W_{LGx} r_G \sin \theta_{GL}) - \\
\mu_{Ga} R_{GP} \sqrt{\left( \frac{W_{LGx} \sin \phi_n}{\cos \lambda \cos \phi_n - \mu \sin \lambda} - W_{LGx} - m_G \dot{\theta}_G r_{GG} \cos \theta_{GL} \right)^2 + \left( W_{Gz} + W_{LGy} + m_G \dot{\theta}_G r_{GG} \sin \theta_{GL} \right)^2} - \\
\mu_{Gj} R_{GJ} \sqrt{W_{LGx}^2 + W_{LGy}^2} 
\]  

(2.2-16)

On the other hand, there are lower and upper bounds in the rotation of the worm gear. The limits are caused by the design of mechanical interferences as shown in Figure 2.2-12 and Figure 2.2-13.

![Figure 2.2-12 Lower End Collision](image)
Since collision effect is not the emphasis of the simulation, dynamic characters of collision here are seen as an impact to a hard spring as shown in Figure 2.2-14.

Thus the system equation of Eqs.(2.2-16) is modified to be Eq.(2.2-17) while the upper or lower collision occurs.
\[
I_G \ddot{\theta}_G + C_G \dot{\theta}_G + (K_c \dot{\theta}_c + C_c \dot{\theta}_c) + \theta_c = W_{G} R_G + W_{LG} r_G \cos \theta_G - W_{LGx} r_G \sin \theta_G - \\
\mu_G R_G \sqrt{\left( \frac{W_G \sin \phi_n}{\cos \lambda \cos \phi_n} - W_{LGx} - m_G \dot{\theta}_G r_G \cos \theta_G \right)^2 + (W_G + W_{LGx} + m_G \dot{\theta}_G r_G \sin \theta_G)^2} \\
- \mu_G R_G \sqrt{W_{LGx}^2 + W_{LGy}^2}
\]

(2.2-17)

Where \( K_c \) and \( C_c \) are stiffness and damping coefficient of the virtual spring. \( \theta_c \) is the angle degree that overtakes the lower or upper limit.

Also notes that the rotation angle between worm shaft and worm gear is \( \theta_G = \frac{d_G \pi}{\lambda} \theta_{WS} \) as shown in Figure 2.2-11.

### 2.2.5 Linkage System

Components of the linkage system of the clutch actuator are shown in Figure 2.2-15, which includes a linkage, a fix plate, and a spring.
The linkage is driven by the worm gear discussed in previous subsection, by which produces a movement to drive the clutch lever through a ball and socket joint as shown. As mentioned in section 2.2.1, the movement on the right end of the linkage, which is a ball and socket joint, can be seen as a motion in pure X direction.

The mutual motion from clutch actuator to clutch is shown in Figure 2.2-16, where structure of the clutch lever is simplified. The actuating force is transmitted from clutch actuator to the clutch lever, through a release bearing on the clutch, finally set clutch to a desired position.

Figure 2.2-16 Structures from Clutch Actuator to Clutch

Figure 2.2-17 shows free body diagram of the linkage.
Figure 2.2-17 Free Body Diagram of the Linkage

From Figure 2.2-17, system equations according to equilibrium of forces and moments are shown in Eqs.(2.2-18).

\[ \sum F_x = 0; \]
\[ W_{LGx} + m_l s_{Lx} + W_{LF} = W_{LCx} \]
\[ \sum F_y = 0; \]
\[ W_{LGy} = m_l s_{Ly} + W_{LCy} \]
\[ \sum M_{xy} = 0; \]
\[ W_{LGx} L_{La} \sin \theta_L + W_{LCx} L_{Lb} \sin \theta_L + T_{LGf} + T_{LCf} + I_L \dot{\theta}_L = W_{LGx} L_{La} \cos \theta_L + W_{LF} L_{LC} \sin \theta_L + W_{LCy} L_{Lb} \cos \theta_L \]

(2.2-18)

Where the x-y axis system in Figure 2.2-17 is at mass center of the linkage. \( W_{LCx} \) and \( W_{LCy} \) are forces that act on the ball and socket joint from the clutch lever. \( W_{LGx} \) and \( W_{LGy} \) are forces from worm gear as mentioned in Eq.(2.2-13). \( W_{LF} \) is force from the spring which assists motion of the linkage to disengage the clutch. \( S_{Lx} \) and \( S_{Ly} \) are displacements of mass center of the linkage in X and Y direction. \( \theta_L \) is rotation angle of the linkage. \( I_L \) is inertia moment of the linkage in direction of Z. \( T_{LCf} \) and \( T_{LGf} \) are torques that caused by
friction at two ends of the linkage. $T_{LGf}$ has mentioned in Eqs.(2.2-15), and $T_{LCf}$ can be expressed as Eq.(2.2-19), where $\mu_{LC}$ is friction coefficient on the ball and socket joint.

$$T_{LCf} = \mu_{LC} R_{LC} \sqrt{W_{LCx}^2 + W_{LCy}^2}$$

(2.2-19)

From Figure 2.2-16, reacting force from clutch, transmitting through a release bearing and clutch lever, is know as $W_{LCx}$. To be simplification, the affection of masses on release bearing and clutch lever are simplified as $m_{BL}$. Thus, let the reacting force from clutch to be $W_c$, clutch lever ratio to be $R_c$, the relation between $W_{LCx}$, and $W_c$ can be expressed as Eq.(2.2-20).

$$W_{LCx} = m_{BL} s_{LC} + C_{BL} s_{LC} + R_c W_c$$

(2.2-20)

Where $C_{BL}$ is the identity of damping coefficients from release bearing to ball and socket joint. $s_{LC}$ is output of the clutch actuator, which is the same with motion of the ball and socket joint in X direction as mentioned in Figure 2.2-2. From Figure 2.2-18, $s_{LC}$ can be expressed as Eq.(2.2-21), where $\theta_L$ is zero at initial condition according to original design of the prototype.

$$s_{LC} = s_{Lx} + L_{Lb}(\cos \theta_L - 1)$$
$$s_{Lx} = L_{Lb} \sin \theta_L$$

(2.2-21)

[Figure 2.2-18 Linkage Arm Motion]
Since rotation angle $\theta_s$ is always small during operation, the displacement of the furrow where $W_{LF}$ acts can be seen as the displacement $S_{LC}$. From the basic dynamic model of spring, force from the spring $W_{LF}$ can be expressed as:

$$-W_{LF} = M_{SP} \dddot{s}_{LC} + C_{SP} \ddot{s}_{LC} + K_{SP} (s_{\text{ins}} - s_{LC})$$  \hspace{1cm} (2.2-22)

where $s_{\text{ins}}$ is pre-deformation of the spring, $M_{SP}$ is identity mass of the spring and the fix plate, $C_{SP}$ is damping coefficient of the spring, and $K_{SP}$ is stiffness coefficient of the spring.

According to Eqs.(2.2-15), Eq.(2.2-19), Eq.(2.2-20), Eq.(2.2-21) and Eq.(2.2-22), Eqs.(2.2-18) can be modified into Eqs.(2.2-23), which are system equations of the linkage system.

$$\sum F_x = 0;\hspace{1cm} W_{LGx} = m_b (S_{la} + L_{Lb} (\cos \theta_L - 1)) + \mu_{Lb} R_{Lb} + \mu_{Lc} R_{Lc} \sqrt{W_{Lx}^2 + W_{Ly}^2} + I_1 \dot{\theta}_L =$$

$$W_{LF} L_a \sin \theta_L + (M_{SP} (S_{la} + L_{Lb} (\cos \theta_L - 1)) + C_{SP} (S_{la} + L_{Lb} (\cos \theta_L - 1))' + K_{SP} (s_{\text{ins}} - s_{LC})) L_a \sin \theta_L + W_{Lc} L_{Lb} \cos \theta_L$$  \hspace{1cm} (2.2-23)

Besides the dynamic properties of the clutch actuator discussed in subsections 2.2.2, 2.2.3, 2.2.4, and 2.2.5, the relative motion from the electrical motor rotation angle $\theta_{WS}$ to output of the linkage tip $D_{out}$, from Figure 2.2-19, can be expressed as Eq.(2.2-24).
In Figure 2.2-19, $\theta_{GL} = \theta_{org} + \theta_{SG}$, where $\theta_{org}$ is an angle from joint to axle of worm gear, $\theta_{SG}$ is rotation angle of worm gear driven by worm shaft, which has a relation with worm shaft rotation angle:

$$\frac{\theta_G}{\theta_{WS}} = \frac{d_g}{d_w \tan \lambda}.$$ 

2.3 Dynamic Analyses of AMT Transmission System

Dynamic analyses of AMT powertrain are introduced in this section. Not like clutch actuator on AMT which has just been developed in recent years; AMT powertrain is almost the same as MT powertrain which has been well developed with maturity for many years. Therefore, not like clutch actuator considered with detail on every element since it is a prototype needed to be modified, dynamic analyses of powertrain are more focus on the dynamic performance obtained from system overview and experiments data.

2.3.1 Structure of AMT Transmission System

Mechanical structure of AMT powertrain is shown in Figure 2.3-1.
Figure 2.3-1 Structure of Automated Manual Transmission

The torque generated by engine is transmitted from engine shaft to clutch, where clutch couples output shaft of engine and input shaft of gear box. In the gear box, several gear ratios are available for input shaft to output shaft. During shifting process, synchronizer release the original engaged gear ratio, and then provides a torque to synchronize the next desired gear ratio to couple with output shaft. The desired transmission torque and rotation speed are transmitted from output shaft of gear box, throw a differential mechanism, to drive wheels.

A diagram of automated manual transmission powertrain is shown in Figure 2.3-2. System blocks of engine, clutch, gear box, and car loading are to be analyzed in the following subsections.
2.3.2 Engine

Engine is power source of the transmission system, as shown in Figure 2.3-3.
Since structure of engine is complex, which includes many mechanical and electrical components like piston, crank, cooling system, speed control, temperature control, firing control, etc. The analyses of engine with detail components are neither practical nor worthful here. Furthermore, study of engine is not the emphases of this study which focuses on AMT shifting and clutch control. Thus, dynamic model of engine is created into a black box, as mentioned in section 2.1.

The character of engine to be concerned is torque, which has correlations with engine speed, throttle position, engine temperature (TPS), and other operating conditions. Even engine temperature or other operation conditions are variables of different driving condition which can’t be controlled in initial. Thus, control parameters of engine torque are chosen to be engine speed and throttle position. Engine speed influences torque be generated, and depends on the dynamic characters of loading condition. Throttle position controls engine power to be generated, which is controlled by accelerator dominated by driver and Transmission Control Unit (TCU) setup up in car.

Table 1.1-1 shows experiments data of engine torque measured by ITRI.

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Table 2.3-1 Engine Torque Data (Nm)

The experiments data is from engine set up on the AMT prototype car with 4-cylinders, 1200 c.c., maximum power 70 h.p. at 6000 r.p.m., and maximum torque 102 Nm at 4000 r.p.m..

From Table 2.3-1, where first column is throttle position (TPS), first row is engine speed, and others are torques generated in different conditions. System dynamic model of engine, which has two input variables: throttle position (TPS) and engine speed and one output of engine torque, can be obtained from these data.

There are two methods to obtain a dynamic model of the engine, which can provides output with any input values. The first method is curve fitting. Curve fitting uses a predetermined system equation obtained from predicting or analyzing system dynamic characters. For example, system character equation of spring are analyzed to be: $F(x)_{pre} = mx'' + cx' + kx$, where $(m, c, k)$ are unknown system parameters, $x$ is system input of spring deflection, and $F(x)_{pre}$ is system output of exerted force. Using optimization methods to solve optimum problem where design variables are unknown system parameters $(m, c, k)$ and cost function is to minimum square error between force from experiments data.
and force from predetermined system function $F(x)_{pre}$ at positions where experiments data exist. Thus, system equation $F(x)_{pre}$ can be obtained with optimized parameters $(m,c,k)$. The second method is to create a look-up table [17]. The experiments data is firstly set up into look-up table, and then uses interpolation-extrapolation method [15] to compute unknown data at points where no experiments data exists. Thus system output is obtainable with any system inputs from look-up table.

Since curve fitting method requires a predetermined system equation according to system characters. Engine is a compound of many subsystems and hence system equation is complicated that not easy to be predicted or analyzed. Thus, look-up table method is used to create a black box of engine dynamic model.

From Table 2.3-1, using interpolation-extrapolation method to compute output torques where no experiment data is provided, an engine torque map according to engine speed and TPS is obtained as shown in Figure 2.3-4.

![Engine Torque Map](image)

*Figure 2.3-4 Map of Engine Torque*
According to engine map shown in Figure 2.3-4, engine dynamic model is obtained from look-up table, which includes two inputs: throttle position and engine speed, and one output of engine torque.

2.3.3 Clutch

Clutch couples shafts of engine output and gear box input. The structure of clutch is shown in Figure 2.3-5.
The clutch cover (Figure 2.3-5) is assembled on a fly wheel which is connected with engine output shaft. The pressure plate couples with a clutch disc connected to gear box input shaft, as shown in Figure 2.3-6. Pressure plate and clutch disc is coupled by an initial threshold load from diaphragm spring after assembled. The couple force is controlled by position of a release bearing assembled beside the diaphragm spring as shown in Figure 2.2-16.

Diaphragm spring is generally used in clutch since it has many advantages comparing to coil spring. However, the dynamic character of diaphragm spring is not like coil spring which has a linear relation between force and deflection as shown in Figure 2.3-7. The system equation of clutch should be created into black box since the dynamic characters is not easy to be expressed theoretically.
There are two inputs for clutch: engine torque and release bearing travel, and one output: torque transferable to input shaft of gear box.

Let engine torque be $T_{eng}$, engine speed be $\dot{\theta}_{eng}$, engine acceleration be $\ddot{\theta}_{eng}$, inertia moment of engine be $I_{eng}$, damping coefficient of engine be $C_{eng}$, torque from normal force between clutch disc and pressure plate be $T_{cf}$, output torque of clutch be $T_{clu}$, and rotation speed of clutch disc be $\dot{\theta}_{clu}$. From Figure 2.3-8, the relations between engine torque and torque out of clutch can be expressed as Eq.(2.3-1).
Considering torque \( T_{cf} \), which is the torque able to be transferred from clutch caused by normal force \( N_{clu} \) between clutch disc and pressure plate. The relation between \( T_{cf} \) and \( N_{clu} \) can be expressed as Eq. (2.3-2) [24].

\[
T_{cf} = \frac{4}{3} \mu_{clutch} N_{clu} \left[ \frac{(R_o^3 - R_i^3)}{(R_o^2 - R_i^2)} \right]
\]

(2.3-2)

Where \( \mu_{clutch} \) is friction coefficient between clutch disc and pressure plate, \( R_o \) is outer radius of pressure plate, and \( R_i \) is inner radius of pressure plate.

Relation between normal force \( N_{clu} \), release bearing motion \( D_{out} \) and release bearing loading \( W_c \) is emphasis of the black box.

The deformation of diaphragm spring can’t base on Hook’s law, since it has no linear relation between loading force and deformation. Moreover, using finite element method to find the relation between loading and deformation is not easy since most of finite element packages are linear which always simplify the transform term it use, for example: Catia. The best way to obtain data of black box is from experiments.

The real line in Figure 2.3-9 shows relation between release bearing travel and release bearing load from experiments [ITRI].
Using curve fitting method, a fitted curve is obtained as shown in Eq.(2.3-3), and drawn in dash line in Figure 2.3-9.

\[ W_c(D_{out}) = \frac{(74.578D_{out}^3 - 194.726D_{out}^2 + 1053.5D_{out})}{(D_{out}^2 - 3.02D_{out} + 3.608)} \]  

(2.3-3)

Cushion spring plays an important role in clutch engagement. As shown in Figure 2.3-10, the range of cushion spring deflection includes the entire zone of engage modulation. Thus, cushion spring deflection is equal to travel of pressure plate \( D_p \) before full disengagement, and normal force \( N_{clu} \) can be seen as deflection force of cushion spring. In conclusion, the normal force \( N_{clu} \) can be represented in terms of pressure plate travel \( D_p \) which has a relation with release bearing travel \( D_{out} \).
The real line in Figure 2.3-11 shows the relation between release bearing travel and pressure plate travel from experiments data. From these data, a five degrees polynomial equation, which represents the relation between $D_{out}$ and $D_p$, can be obtained using curve fitting method. The equation is shown in Eq. (2.3-4)

$$D_p (D_{out}) = 0.0002615D_{out}^5 - 0.005043D_{out}^4 + 0.02873D_{out}^3 - 0.02388D_{out}^2 + 0.004408D_{out}$$

(2.3-4)
And then from Figure 2.3-12 the relation between deflection and load of cushion spring can be obtained as Eq.(2.3-5)
From Eq.(2.3-4) and Eq.(2.3-5), a function of \( N_{\text{clu}}(D_p) \) can be obtained.

To provide more comfort to driver and passenger, stage springs, which work like a torsional damper, always used in clutch as shown in Figure 2.3-6. The stage springs play important rules on clutch, it reduce gear rattle and boom noises transmitted into the passenger compartment [24].

Considering with stage springs, a free body diagram from output shaft of engine to input shaft of gear box is shown in Figure 2.3-13.

![Free Body Diagram of Clutch](image)

**Figure 2.3-13 Free Body Diagram of Clutch**

From Figure 2.3-13, and considering Eq.(2.3-3), Eq.(2.3-4), and Eq.(2.3-5), system equation of clutch can be expressed as Eqs.(2.3-6).
\[
\begin{align*}
&\text{if } T_{\text{eng}} - (I_{\text{eng}} \dot{\theta}_{\text{eng}} + C_{\text{eng}} \ddot{\theta}_{\text{eng}}) > T_{\text{cf}} \\
&\quad \{ T_{IG} = T_{\text{cf}} - \left[ (I_{CP} + \frac{1}{2} I_{ss}) \ddot{\theta}_{\text{clu}} + (C_{CP} + C_{ss}) \dot{\theta}_{\text{clu}} \right] \} \\
&\text{if } T_{\text{eng}} - (I_{\text{eng}} \dot{\theta}_{\text{eng}} + C_{\text{eng}} \ddot{\theta}_{\text{eng}}) \leq T_{\text{cf}} \\
&\quad \left\{ \begin{array}{l}
\text{if } \dot{\theta}_{\text{eng}} \neq \dot{\theta}_{\text{clu}} \\
T_{IG} = T_{\text{cf}} - \left[ (I_{CP} + \frac{1}{2} I_{ss}) \ddot{\theta}_{\text{clu}} + (C_{CP} + C_{ss}) \dot{\theta}_{\text{clu}} \right] \\
\text{if } \dot{\theta}_{\text{eng}} = \dot{\theta}_{\text{clu}} \\
T_{IG} = T_{\text{eng}} - (I_{\text{eng}} \dot{\theta}_{\text{eng}} + C_{\text{eng}} \ddot{\theta}_{\text{eng}}) - \left[ (I_{CP} + \frac{1}{2} I_{ss}) \ddot{\theta}_{\text{clu}} + (C_{CP} + C_{ss}) \dot{\theta}_{\text{clu}} \right] \\
\end{array} \right. \quad (2.3-6) \\
&\{ \theta_{IG} = \theta_{\text{clu}} - \theta_{ss}; \\
&K_{ss} \theta_{ss} + C_{ss} \dot{\theta}_{ss} = T_{IG}; \}
\end{align*}
\]

Where \( I_{CP} \) is inertia moment of clutch disc, \( I_{ss} \) is inertia moment of stage springs, \( C_{CP} \) is damping coefficient of clutch disc, \( C_{ss} \) is damping coefficient of stage springs, \( T_{IG} \) is output torque of clutch, \( \theta_{IG} \) is rotation angle of output shaft of clutch, \( \theta_{ss} \) is deformation angle of stage springs, and \( K_{ss} \) is stiffness coefficient of stage springs.

According to [20], transmission efficiency of total powertrain can be simplified to the efficiency of engine output torque. Thus, the term \( T_{\text{eng}} \) in Eq.(2.3-6) is assigned to be \( \eta_{\text{eng}} T_{\text{engR}} \). Where \( \eta_{\text{eng}} \) is transmission efficiency of total powertrain, and \( T_{\text{engR}} \) is torque generated by engine as shown in engine map of Figure 2.3-4.

### 2.3.4 Gear Box

The analyses of gear box in this subsection include components of powertrain from clutch to driving wheels. As shown in the shadow parts of Figure 2.3-14.
Figure 2.3-14 Final Transmission Structures [18]

Figure 2.3-14 shows final components of powertrain within a front-engine front-drive (FF) car, which is the type used in AMT prototype car. Torque is transmitted from clutch mentioned before into gear box, after a transformation by gears, transmitting the desired torque to output shaft. Finally, through a differential mechanism which only works while turning, the power is transmitted to the driving wheels to move a car. A simplified final powertrain diagram is shown in Figure 2.3-15.

Figure 2.3-15 Simplified Final Powertrain Diagram
From Figure 2.3-15, the transmission components from clutch to drive wheels, which are objectives of this subsection, can be divided into two components: gear box and differential mechanism.

The inner structure of gear box is shown in Figure 2.3-16.

![Inner Structure of Gear Box](image)

**Figure 2.3-16 Inner Structure of Gear Box [18]**

The torque transmitted from input shaft is firstly transmitted to inverse shaft by a couple of gears, and then the inverse shaft couples with output shaft by one of the gears ratios. Synchronizers are set between output shaft and gears on output shaft, which are used to engage gear with desired gear ratio with output shaft. A process of synchronizer engaging a gear is shown in Figure 2.3-17.
As shown in Figure 2.3-17. To shift the transmission into gear, the synchronizer sleeve is moved toward that gear. The sleeve slides on the hub splines and carries the three key with it. The keys butt against the synchronizer ring and push it toward the gear. This brings the cone surface in the ring into contact with the cone surface on the gear. Friction between the synchronizer ring and the gear brings the two into synchronous rotation, which rotate at the same speed. As shown in the right figure of Figure 2.3-17, when the external teeth on the synchronizer ring and the gear rotate at the same speed, the sleeve slides over them. This locks the gear to the shaft and completes the shift. Power flows from the gear, through the synchronizer sleeve and hub, to the shaft [18]. The same reverse process from right to left of Figure 2.3-17 is done to disengage a gear while shifting starts.

By the use of synchronizer, different transmission gear ratios in gear box can be shifted by disengage the original gear ratio and engage to the desire gear ratio, as shown in Figure 2.3-18. Note that synchronizer can be worked only after clutch disengages the transmission from engine which leads to a free rotation of input shaft.
The first step to analyze dynamic characters of the gear box is to consider with a general condition while no shifting process is proceeded. A free body diagram of such condition is shown in Figure 2.3-19.
Figure 2.3-19 Free Body Diagram of Gear Box

Where the equilibrium equations can be expressed as Eqs.(2.3-7)

\[
\begin{align*}
T_{IG} &= T_{IR} + I_{IG} \ddot{\theta}_{IG} + C_{IG} \dot{\theta}_{IG} \\
T_{IR} &= T_{OR} + I_{RG} \ddot{\theta}_{RG} + C_{RG} \dot{\theta}_{RG} \\
T_{RG} &= T_{OG} + I_{OG} \ddot{\theta}_{OG} + C_{OG} \dot{\theta}_{OG}
\end{align*}
\]

(2.3-7)

\(T_{IG}\) is the torque transmitted from clutch, \(T_{IR}\) is the torque transmitted from input shaft to inverse shaft where the gear ratio between these two shaft is usually one, \(T_{OR}\) is the reacting torque from output shaft to the inverse shaft, \(T_{RG}\) is the torque from inverse shaft that acts on output shaft, \(T_{OG}\) is the torque transmitted to differential mechanism by output shaft; \(I_{IG}, I_{RG}, I_{OG}\) are inertia moments of input shaft, inverse shaft, and output shaft; \(C_{IG}, C_{RG}, C_{OG}\) are damping coefficients on input shaft, inverse shaft, and output shaft; \(\theta_{IG}, \theta_{RG}, \theta_{OG}\) are rotation angle of input shaft, inverse shaft, and output shaft.

Let transmission gear ratio in gear box be \(R_G\), and gear ratio between input shaft and
inverse shaft be one, the relationships between $\theta_{IG}$, $\theta_{RG}$, $\theta_{OG}$ and $T_{IR}$, $T_{OR}$, $T_{RG}$ can be expressed as:

\[
\begin{align*}
\theta_{IG} &= \theta_{RG}; \dot{\theta}_{IG} = \dot{\theta}_{RG}; \ddot{\theta}_{IG} = \ddot{\theta}_{RG} \\
\theta_{OG} &= \frac{\theta_{RG}}{R_G}; \dot{\theta}_{OG} = \frac{\dot{\theta}_{RG}}{R_G}; \ddot{\theta}_{OG} = \frac{\ddot{\theta}_{RG}}{R_G} \\
T_{IR} &= T_{IR} \\
T_{RG} &= R_G T_{OR}
\end{align*}
\] (2.3-8)

From Eq.(2.3-8), Eq.(2.3-7) can be expressed as a simplified system equation:

\[
T_{IG} = (I_{ig} + I_{rg} + \frac{I_{OG}}{R_G^2})\ddot{\theta}_{IG} + (C_{ig} + C_{rg} + \frac{C_{OG}}{R_G^2})\dot{\theta}_{IG} + \frac{T_{OG}}{R_G} \quad (2.3-9)
\]

The second step is to consider the situation when shifting process is proceeded. Synchronizer exerts a torque $T_{syn}$ on output shaft. The magnitude of $T_{syn}$ is assumed to be constant here, since the torque depends on the shifting speed and rotating speed that is not easy to be determined theoretically, and the exact $T_{syn}$, which is not the emphases of this study, also plays no important rule on the shifting process. The direction of $T_{syn}$ depends on shifting directions of up-shift or down-shift. When up-shift is executed, rotation speed of gear to be synchronized is slower than output shaft. Thus, synchronizer exerts a positive $T_{syn}$ against the rotation direction to speed up the gear. In the same way, a negative $T_{syn}$ is exerted while down-shift is executed. A free body diagram of up-shift process is shown in Figure 2.3-20.
The system dynamic equations according to Figure 2.3-20 can be expressed as:

\[
T_{IG} = (I_{IG} + I_{RG})\ddot{\theta}_{IG} + (C_{IG} + C_{RG})\dot{\theta}_{IG} + \frac{T_{syn}}{R \ G} \\
T_{syn} = T_{OG} + I_{OG}\ddot{\theta}_{OG} + C_{OG}\dot{\theta}_{OG}
\]  \hspace{1cm} (2.3-10)

Considering with differential mechanism, it is only used to provide different speed upon the two drive wheels while vehicle is turning, such affection doesn’t matter on transmission in general condition of straight driving which is the case in this study. However, differential mechanism still provides a final gear ratio that should be considered. Thus, it is simplified to a couple of gears with gear ratio \( R_{DM} \).

The free body diagram of such simplified differential mechanism is shown in Figure 2.3-21, where the input shaft is output shaft of gear box, and the output shaft connects to drive wheels.
System dynamic equation of differential mechanism according to Figure 2.3-21 can be expressed as:

\[ T_{GD} = I_{DM} \dot{\theta}_{DM} + C_{DM} \dot{\theta}_{DM} + T_{DM} \]  \hspace{1cm} (2.3-11)

Where \( T_{GD} \) is the torque transmitted from output shaft of gear box, \( I_{DM} \) is the identity inertia moment of shafts from differential mechanism to output shafts on drive wheels, \( C_{DM} \) is the damping coefficient of identity shaft, \( \theta_{DM} \) is the rotation angle of output shaft which equals to rotation angle of drive wheels, and \( T_{DM} \) is the torque exerted on drive wheels.

Let the final gear ratio in differential mechanism be \( R_{DM} \), the following relations are obtained:

\[ \theta_{DM} = \theta_{OG} / R_{DM}; \quad \dot{\theta}_{DM} = \dot{\theta}_{OG} / R_{DM}; \quad \ddot{\theta}_{DM} = \ddot{\theta}_{OG} / R_{DM} \]  \hspace{1cm} (2.3-12)

\[ T_{GD} = R_{DM} T_{OG} \]

From Eqs.(2.3-9), Eqs.(2.3-11), and Eqs.(2.3-12), the system dynamic equation of final
powertrain from output shaft on clutch to output shaft on drive wheels can be expressed as Eq.(2.3-13), which is in condition when no shifting process is proceeded.

\[
T_{IG} = (I_{IG} + I_{RG} + \frac{I_{OG}}{R_g} + \frac{I_{DM}}{R_g^2 R_{DM}} \dot{\theta}_{IG} + (C_{IG} + C_{RG} + \frac{C_{OG}}{R_g^2} + \frac{C_{DM}}{R_g^2 R_{DM}} \dot{\theta}_{OG} + \frac{T_{DM}}{R_g R_{DM}} \tag{2.3-13}
\]

The system dynamic equations while shifting process is proceeded, from Eqs.(2.3-10), Eqs.(2.3-11), and Eqs.(2.3-12), can be expressed as Eq.(2.3-14), which is an up-shift process.

\[
T_{IG} = (I_{IG} + I_{RG}) \dot{\theta}_{IG} + (C_{IG} + C_{RG}) \dot{\theta}_{IG} + \frac{T_{syn}}{R_g} \tag{2.3-14}
\]

\[
T_{syn} - \frac{T_{DM}}{R_{DM}} = (I_{OG} + \frac{I_{DM}}{R_{DM}} \dot{\theta}_{OG} + (C_{OG} + C_{DM}) \frac{T_{DM}}{R_{DM}} \dot{\theta}_{OG} \tag{2.3-14}
\]

From \( \theta_{DM} \), car speed \( v_{car} \) can be obtained as \( v_{car} = \theta_{DM} r_{wh} \), where \( r_{wh} \) is radius of wheels.

In shifting process, system dynamic equation transfers from Eq.(2.3-13) to Eqs.(2.3-14) where synchronizer is actuated to disengage the original gear and to engage the new gear of the next desired transmission ratio. After shifting process is completed, where rotation speed is the same between the new gear and the output shaft of the gear box as in the condition of: \( \dot{\theta}_{RG} = \frac{\dot{\theta}_{OG}}{R_{G(NEW)}} \), system equation transfers from Eqs.(2.3-14) to Eq.(2.3-13).

### 2.3.5 Car Loading

There are two general forms to express the car loading on transmission system: full car equation and components equation. Full car equation concerns the relation between car loading and square of car speed, using experiments data to mesh the equation in the form of

\[
F_s = f_1 v_{car}^2 + f_0, \quad \text{where } F_s \text{ is the force of car loading, } f_1 \text{ and } f_0 \text{ are coefficients to be determined. Components equation concerns affections from rolling resistance, aerodynamic,} \]


gravity, and other outside loadings on a car, since it is an expression with part modules which is easier to be modified. Components equation is used in this study and is introduced in the following essay.

Forces that act on a car are shown in Figure 2.3-22.

![Figure 2.3-22 Forces Acting on a Vehicle](image)

From Figure 2.3-22, loading condition of a car can be expressed as Eq.(2.3-15) [20], which is in the form of components equation.

\[ F_x = M_{\text{car}}a_x + R_x + D_A + R_{\text{hs}} + W_{\text{car}} \sin \theta_{\text{load}} \quad (2.3-15) \]

Where \( M_{\text{car}} \) is mass of the car, \( a_x \) is longitudinal acceleration of the car, \( R_x \) is rolling resistance forces on the four wheels, \( D_A \) is aerodynamic drag force, \( R_{\text{hs}} \) is hitch forces, \( W_{\text{car}} \) is the force according to the gravity that acts on a car, which is known as \( gM_{\text{car}} \) where \( g \) is the gravity, \( \theta_{\text{load}} \) is gradient angle of the road, and \( F_x \) is the force from drive wheels transmitted from engine.

In general, rolling resistance forces \( R_x \) on wheels can be expressed as:

\[ R_x = f_WW_{\text{car}} \quad (2.3-16) \]
Where $f_r$ is called rolling resistance coefficient.

The aerodynamic drag force $D_A$ can be expressed as:

$$D_A = \frac{1}{2} \rho v_{car}^2 C_D A = \frac{1}{2} \rho (\dot{\omega}_{DMr_{wh}})^2 C_D A$$  \hspace{1cm} (2.3-17)

Where $\rho$ is air density, $r_{wh}$ is radius of wheels, and $v_{car}$ is speed of the car which equals to $\dot{\omega}_{DMr_{wh}}$ if the wind speed is zero, otherwise, it should be the relative speed between car and wind. $C_D$ is aerodynamic drag coefficient, which is determined empirically from car, and $A$ is frontal area of the car.

According to Eq.(2.3-13) where $T_{DM}$ is the torque that acts on wheels, Eq.(2.3-15) can be expressed as:

$$T_{DM} = I_{wh} \ddot{\omega}_{DM} + C_{wh} \dot{\omega}_{DM} + (M_{car} \ddot{\omega}_{DMr_{wh}} + R_x + D_A + R_{hx} + W_{car} \sin \theta_{load}) r_{wh}$$  \hspace{1cm} (2.3-18)

Where $I_{wh}$ is the identity inertia moment of the four wheels and $C_{wh}$ is the identity damping coefficient on wheels.

2.4 Dynamic Model Creation

Dynamic models according to system dynamic equations discussed in previous subsections are created into computer program in this section.

There are many commercial packages available to create dynamic models and do dynamic analysis, for example: Adams®, Working Model®, Visual Nastrain®, etc. However, the difficulty in engineering is not just the analyses of dynamic or static problems, but the problems to combine dynamic characters to control design, since the final purpose of engineering is to create a machine that is workable as expectation. Packets like Adams®, Working Model®, Visual Nastrain®, etc. are convenient for dynamic analysis. However, it is
difficult to combine such dynamic characters to control design.

Engineering problems, whatever dynamic problem or control problem, are all sorts of math problems. The study here combines both dynamic problems, control problems, and optimization problems. The best way to solve all these problems integrally is to solve by mathematical methods which is the basis of all these engineering problems. Matlab®, a mathematically based engineering package produced by Math Work is used, because one of its packages: Simulink®, is able to create system equations with modules, which is one of the objectives in this study as mentioned in section 1.2.

2.4.1 Matlab Simulink

Simulink® is one of the packages in Matlab®. In Simulink®, system equations can be created into a block according to the mechanical structure it belongs. And then with blocks created, many blocks can be combined into a grater block which can represent a complete system be simulated. Relations between block and block are connected by transfer lines defined in Simulink®. Each transfer line has one end of input and one or more ends of output that transfers parameters or variables it connects. And the combination of blocks can be in the form of close loop, where iterations are done automatically in the computing core.

In the following subsections, dynamic models of clutch actuator and AMT powertrain are created into Matlab® Simulink according to the system equations developed before

2.4.2 Module Creation

In this subsection, program models are created according to each component defined previously.

The components created here are combined further in subsection 2.4.4.
Electrical Motor

The system equations of the electrical motor are shown in Eq.(2.2-3) and Eq.(2.2-5). From Eq.(2.2-5), using Laplace Transform, current through armature can be obtained. According to the current, torque generated by electrical motor can be obtained from Eq.(2.2-3). The blocks according to these operations and equations in Simulink® are shown in Figure 2.4-1.

![Figure 2.4-1 Electrical Motor in Simulink® Model]

Combining these operating blocks, system dynamic equations of electrical motor can be created into an electrical motor module, as shown in Figure 2.4-2.

![Figure 2.4-2 DC Motor Module]

The parameters in this module are stated in Table 2.4-1.

<table>
<thead>
<tr>
<th>Input</th>
<th>$v_o$, $\dot{\theta}_m$</th>
</tr>
</thead>
<tbody>
<tr>
<td>Output</td>
<td>$T$</td>
</tr>
<tr>
<td>System Parameters</td>
<td>$K_i$, $K_e$, $La$, $R_a$</td>
</tr>
</tbody>
</table>

Table 2.4-1 Parameter Table of Electrical Motor
Worm Shaft

System equations of the worm shaft are shown in Eq.(2.2-12) and depends on Eq.(2.2-10) and Eq.(2.2-11). Creating the equations into Simulink®, the blocks are shown in Figure 2.4-3.

![Figure 2.4-3 Worm Shaft in Simulink® Model](image)

The parameters in this module are stated in Table 2.4-2.

<table>
<thead>
<tr>
<th>Input</th>
<th>Output</th>
<th>System Parameters</th>
</tr>
</thead>
<tbody>
<tr>
<td>$T$, $W_{Gr}$</td>
<td>$\theta_{WS}$</td>
<td>$I_{WS}$, $C_{WS}$, $\phi_n$, $\lambda$, $\mu$, $d_w$</td>
</tr>
</tbody>
</table>

Table 2.4-2 Parameter Table of Worm Shaft

Note that equation blocks in figures above not completely state the equations developed before, but simplified diagrams which include many operators within the blocks used to solve the equations. In this study, for example, most of second order differential equations are solved by Laplace Transform method [21], transferring original equations to Laplace transfer functions for easier computing. For instance, electrical motor module stated above solving Eq.(2.2-3) and Eq.(2.2-5) using blocks in Figure 2.4-1 and finally merges these operating blocks to a combined module. On the other hand, system equations of each component, including many mathematical equations and operators as blocks, are combined into a bigger
block as a module, as shown in Figure 2.4-2. Such process is reiterative in each component. Thus, the same process is not stated in the following discussions.

**Worm Gear**

System dynamic equation of worm gear is shown in Eq.(2.2-16), which is a general condition where the working angle is between upper and lower limits. A diagram of system equation created in Simulink® is shown in Figure 2.4-4.

![Figure 2.4-4 Worm Gear in Simulink® Model](image)

Besides the general condition, the dynamic model should also consider the collision states as shown in Figure 2.2-12 and Figure 2.2-13. The system equations of such states are stated in Eq.(2.2-17). To check the condition, the model should estimate the rotation angle overtakes the upper/lower limits or not. If collision occurs, the model shown in Figure 2.4-4 should add a collision term as expressed in Eq.(2.2-17). Model of such condition is shown in Figure 2.4-5.
Figure 2.4-5 Collision Model of Worm Gear

The parameters in this model are shown in Table 2.4-3.

<table>
<thead>
<tr>
<th>Input</th>
<th>( \theta_{WS}, W_{LGx}, W_{LGy} )</th>
</tr>
</thead>
<tbody>
<tr>
<td>Output</td>
<td>( W_{Gi} )</td>
</tr>
<tr>
<td>System Parameters</td>
<td>( I_G, C_G, R_G, r_G, \theta_{GL}, \phi_n, \lambda, \mu_{Ga}, \mu_{Gj}, R_{GP}, r_{GG}, m_G, R_{Ga}, K_c, C_c )</td>
</tr>
</tbody>
</table>

Table 2.4-3 Parameter Table of Worm Gear

Linkage System

The system equations of linkage system are shown in Eqs.(2.2-23).

Besides Eqs.(2.2-23), Eq.(2.2-21), Eq.(2.2-22), and Eq.(2.2-24) provide some relations of inputs and outputs. From these equations, the dynamic model can be expressed in diagram as shown in Figure 2.4-6.
The parameters in this model are shown in Table 2.4-4.

<table>
<thead>
<tr>
<th>Input</th>
<th>$\theta_{WS}$, $W_e$, $R_y$</th>
</tr>
</thead>
<tbody>
<tr>
<td>Output</td>
<td>$W_{LGx}$, $W_{LGy}$</td>
</tr>
<tr>
<td>System Parameters</td>
<td>$m_{BL}$, $C_{BL}$, $L_{Lb}$, $L_{La}$, $L_{LC}$, $m_k$, $M_{SP}$, $C_{SP}$, $K_{SP}$, $s_{ins}$, $\mu_{Gj}$, $\mu_{LC}$, $R_{GJ}$, $R_{LC}$</td>
</tr>
</tbody>
</table>

Table 2.4-4 Parameter Table of Linkage System

Engine

According to Table 2.3-1 and using interpolation-extrapolation method to compute the output torques where no experiment is available, the engine torque map according to engine speed and TPS in shown in Figure 2.3-4.

In Matlab® Simulink, such data can be computed using 2-D lookup table, which has two
inputs: TPS and engine speed, and one output of engine torque. As shown in Figure 2.4-7.

![Figure 2.4-7 Engine Map in Simulink® Model](image)

**Figure 2.4-7 Engine Map in Simulink® Model**

**Clutch**

System equation of the clutch is stated in Eq.(2.3-6). Besides, Eq.(2.3-3), Eq.(2.3-4), and Eq.(2.3-5) show the dynamic characters of the clutch.

The model of clutch system is complicated since it contains logical judgment, dynamic equations, and experiments data. Thus, the model creation chart here is not expressed with detail, where the complete system equations and logic judgment are expressed in 2.3.3 and the creation process is the same with above. Figure 2.4-8 just shows a diagram of simplified flowchart.

![Figure 2.4-8 Clutch in Simulink® Model](image)
The parameters in this model are stated in Table 2.4-5.

<table>
<thead>
<tr>
<th>Input</th>
<th>$T_{\text{eng}}$, $D_{\text{out}}$, $T_{\text{IG}}$</th>
</tr>
</thead>
<tbody>
<tr>
<td>Output</td>
<td>$\theta_{\text{eng}}$, $W_{c}$, $\theta_{\text{IG}}$</td>
</tr>
<tr>
<td>System Parameters</td>
<td>$I_{\text{eng}}$, $C_{\text{eng}}$, $\mu_{\text{clutch}}$, $R_0$, $R_i$, $I_{CP}$, $I_{ss}$, $C_{CP}$, $C_{ss}$, $K_{ss}$, $\eta_{\text{eng}}$</td>
</tr>
</tbody>
</table>

**Table 2.4-5 Parameter Table of Clutch**

**Gear Box**

The system equation of gear box in general condition is shown in Eq.(2.3-13). Combing with Eq.(2.3-12), an output of car speed can be obtained. The Simulink® model diagram is shown in Figure 2.4-9.

![Figure 2.4-9 Gear Box in Simulink® Model](image)

While shifting process is proceed, system equations should be transformed to Eqs.(2.3-14). The model built in Simulink® is shown in Figure 2.4-10.
Figure 2.4-10 Gear Box (Shifting) in Simulink® Model

The parameters in this model are stated in Table 2.4-6.

<table>
<thead>
<tr>
<th>Input</th>
<th>$T_{IG}$, $T_{syn}$, $R_G$, $T_{DM}$</th>
</tr>
</thead>
<tbody>
<tr>
<td>Output</td>
<td>$\theta_{DM}$, $\theta_{IG}$</td>
</tr>
<tr>
<td>System Parameters</td>
<td>$I_{IG}$, $I_{RG}$, $I_{OG}$, $I_{DM}$, $C_{IG}$, $C_{RG}$, $C_{OG}$, $C_{DM}$, $R_{DM}$</td>
</tr>
</tbody>
</table>

Table 2.4-6 Parameter Table of Gear Box

Car Loading

The car loading equation is shown in Eq.(2.3-18). Figure 2.4-11 shows the diagram of this model.

Figure 2.4-11 Loading Function in Simulink® Model
The parameters in this model are stated in Table 2.4-7.

<table>
<thead>
<tr>
<th>Input</th>
<th>(\theta_{DM}, R_{ks}, \theta_{load})</th>
</tr>
</thead>
<tbody>
<tr>
<td>Output</td>
<td>(T_{DM})</td>
</tr>
<tr>
<td>System Parameters</td>
<td>(M_{car}, g, f_r, \rho, I_{wh}, C_D, A, I_{wh}, C_{wh})</td>
</tr>
</tbody>
</table>

**Table 2.4-7 Parameter Table of Car Loading**

### 2.4.3 Parameter Setting

According to the models created in previous subsection, system parameters revealed in Table 2.4-1 to Table 2.4-7 are set here.

There are three methods to set the system parameters in this study: experiments data, table data, and CAD model computation. Most dimensions of components have been defined in the prototype; such parameters can be set directly from the design chart. Other properties of the components, like masses and inertia moments, are estimated by creating CAD model in Catia, and then computing inertia properties according to the materials applied. Parameters of some structures that are too complex to be built up into CAD model like engine, gear box, differential mechanism, etc., are determined by experiments data. Other parameters like friction coefficients, damping coefficients, air density, etc., are obtained by data tables.

Besides, note that static friction coefficients are set to be 1.2 times the dynamic friction coefficient in all models according to general conception [16].

Parameters of part dimension that have been defined in prototype design are shown in Table 2.4-8.

<table>
<thead>
<tr>
<th>(\phi_{a})</th>
<th>(\lambda)</th>
<th>(d_w)</th>
<th>(R_G)</th>
<th>(\theta_{GL})</th>
<th>(R_{GP})</th>
<th>(R_{GJ})</th>
<th>(K_{SP})</th>
<th>(s_{ins})</th>
<th>(R_{LC})</th>
</tr>
</thead>
<tbody>
<tr>
<td>10.5°</td>
<td>5°</td>
<td>8mm</td>
<td>34mm</td>
<td>30°</td>
<td>3.5mm</td>
<td>3mm</td>
<td>0.6kg/mm</td>
<td>40mm</td>
<td>4mm</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>(M_{car})</th>
<th>(A)</th>
<th>(R_{DM})</th>
<th>(r_{wh})</th>
<th>(R_G)</th>
<th>(Ro)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1400kg</td>
<td>2.74 m(^2)</td>
<td>5.375</td>
<td>0.287m</td>
<td>3.945, 2.177, 1.394, 1.000, 0.853</td>
<td>95mm</td>
</tr>
<tr>
<td>$R_i$</td>
<td>$R_r$</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>-------</td>
<td>-------</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>66mm</td>
<td>1</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

Table 2.4-8 Parameters from Prototype

By CAD model created in Catia, parameters of some inertia moment and mass are obtained as shown in Table 2.4-9.

<table>
<thead>
<tr>
<th>$I_{WS}$</th>
<th>$I_G$</th>
<th>$m_G$</th>
<th>$r_{GG}$</th>
<th>$m_{BL}$</th>
<th>$L_{Lu}$</th>
<th>$L_{Lb}$</th>
</tr>
</thead>
<tbody>
<tr>
<td>9.318e-7 $kg \cdot m^2$</td>
<td>9.861e-5 $kg \cdot m^2$</td>
<td>0.16kg</td>
<td>12.8mm</td>
<td>1.24kg</td>
<td>60.748mm</td>
<td>70.162mm</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>$L_{LC}$</th>
<th>$m_L$</th>
<th>$M_{SP}$</th>
<th>$I_L$</th>
</tr>
</thead>
<tbody>
<tr>
<td>44.762mm</td>
<td>0.052kg</td>
<td>0.048</td>
<td>9.828e-005 $kg \cdot m^2$</td>
</tr>
</tbody>
</table>

Table 2.4-9 Parameters from Catia Models

$M_{SP}$ is an identity mass of assist spring and fix plate, note that the mass of spring is not just the mass measured from CAD model.

In general, model of spring is simplified as Figure 2.4-12.

![Figure 2.4-12 General Spring Model](image)

However the mass of spring $M_{spring}$ in the general model can’t just be the mass of spring from balance.

For an exact view of spring, a spring can be decomposed into many small masses, as shown in Figure 2.4-13.
From Figure 2.4-13, the system equation of spring, considering the term of mass \( F_M \), can be expressed as:

\[
F_M = \int_0^s \left( \frac{x}{s} \cdot \frac{m}{s} \right) dx = \frac{m}{2} \ddot{x}_{spring} = M_{spring} \ddot{x}_{spring}
\]  

(2.4-1)

Where each mass element \( dM \) is expressed as \( \frac{m}{s} dx \), and acceleration \( \ddot{x}_n \) is expressed as \( \frac{x}{s} \ddot{x}_{spring} \), where \( s \) is length of spring, \( \ddot{x}_{spring} \) is longitudinal acceleration of spring at right end.

Thus, the mass of spring \( M_{spring} \) is appointed to \( \frac{m}{2} \), where \( m \) is the mass of spring from balance.

Table 2.4-10 shows parameters obtained from experiments data.

<table>
<thead>
<tr>
<th>( I_{IG} )</th>
<th>( I_{RG} )</th>
<th>( I_{OG} )</th>
<th>( I_{DM} )</th>
<th>( C_{IG} )</th>
<th>( C_{RG} )</th>
</tr>
</thead>
<tbody>
<tr>
<td>( 0.074 \text{ kg} \cdot \text{m}^2 )</td>
<td>( 1.4 \times 10^{-3} \text{ kg} \cdot \text{m}^2 )</td>
<td>( 0.74 \times 10^{-3} \text{ kg} \cdot \text{m}^2 )</td>
<td>( 2 \times 10^{-3} \text{ kg} \cdot \text{m}^2 )</td>
<td>( 0.1 \text{ N} \cdot \text{t/m} )</td>
<td>( 0.01 \text{ N} \cdot \text{t/m} )</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>( C_{OG} )</th>
<th>( C_{DM} )</th>
<th>( \mu_{clutch} )</th>
<th>( \eta_{eng} )</th>
<th>( C_{WS} )</th>
<th>( C_G )</th>
</tr>
</thead>
<tbody>
<tr>
<td>( 0.01 \text{ N} \cdot \text{t/m} )</td>
<td>( 0.05 \text{ N} \cdot \text{t/m} )</td>
<td>0.4</td>
<td>0.9</td>
<td>( 4.4 \times 10^{-6} \text{ N} \cdot \text{t/m} )</td>
<td>( 1 \times 10^{-6} \text{ N} \cdot \text{t/m} )</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>( C_{BL} )</th>
<th>( C_{SP} )</th>
<th>( f_r )</th>
<th>( C_D )</th>
<th>( C_{wh} )</th>
<th>( I_{eng} )</th>
</tr>
</thead>
<tbody>
<tr>
<td>( 1 \times 10^{-9} \text{ N} \cdot \text{t/m} )</td>
<td>( 1 \times 10^{-7} \text{ N} \cdot \text{t/m} )</td>
<td>0.01386</td>
<td>0.53</td>
<td>( 0.1 \text{ N} \cdot \text{t/m} )</td>
<td>( 0.074 \text{ kg} \cdot \text{m}^2 )</td>
</tr>
<tr>
<td>$C_{en}g$</td>
<td>$\mu_{clutch}$</td>
<td>$I_{CP}$</td>
<td>$I_{ss}$</td>
<td>$C_{CP}$</td>
<td>$K_{ss}$</td>
</tr>
<tr>
<td>---------</td>
<td>------------</td>
<td>--------</td>
<td>--------</td>
<td>--------</td>
<td>--------</td>
</tr>
<tr>
<td>0.15 $N \cdot t/m$</td>
<td>0.4</td>
<td>0.001 $kg \cdot m^2$</td>
<td>5e-3 $kg \cdot m^2$</td>
<td>1e-4 $N/m$</td>
<td>735.47 $N \cdot m/arc$</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>$C_{ss}$</th>
<th>$T_{syn}$</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.1 $kg/t$</td>
<td>0.1 $N \cdot m$</td>
</tr>
</tbody>
</table>

Table 2.4-10 Parameters from Experiments data

Besides, torque constant $K_t$, EMF constant $K_e$, and armature resistance $R_s$ of electrical motor can be obtained from experiments data too. Table 2.4-11 shows experiment data of electrical motor of the clutch actuator.

<table>
<thead>
<tr>
<th>Torque</th>
<th>3</th>
<th>3.15</th>
<th>4.19</th>
<th>5.1</th>
<th>5.61</th>
<th>6.18</th>
<th>7.42</th>
</tr>
</thead>
<tbody>
<tr>
<td>$\omega$(R.P.M.)</td>
<td>6788</td>
<td>6779</td>
<td>6738</td>
<td>6630</td>
<td>6469</td>
<td>6309</td>
<td>6174</td>
</tr>
<tr>
<td>Current</td>
<td>5.006</td>
<td>5.103</td>
<td>5.249</td>
<td>5.684</td>
<td>6.624</td>
<td>7.185</td>
<td>7.735</td>
</tr>
<tr>
<td>Voltage</td>
<td>12.11</td>
<td>12.1</td>
<td>12.07</td>
<td>12</td>
<td>11.94</td>
<td>11.88</td>
<td>11.85</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Torque</th>
<th>8.22</th>
<th>9.42</th>
<th>10.12</th>
<th>10.97</th>
<th>11.92</th>
<th>12.8</th>
<th>13.8</th>
</tr>
</thead>
<tbody>
<tr>
<td>$\omega$(R.P.M.)</td>
<td>6059</td>
<td>5948</td>
<td>5831</td>
<td>5707</td>
<td>5586</td>
<td>5468</td>
<td>5350</td>
</tr>
<tr>
<td>Voltage</td>
<td>11.82</td>
<td>11.79</td>
<td>11.75</td>
<td>11.69</td>
<td>11.66</td>
<td>11.61</td>
<td>11.57</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Torque</th>
<th>14.45</th>
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<td>17.819</td>
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</tr>
<tr>
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<td>11.15</td>
<td>11.11</td>
<td>11.1</td>
<td>11.08</td>
<td>11.04</td>
<td>10.99</td>
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</table>

<p>| Torque | 27.2 | 27.57 | 28.47 | 29.27 | 30.6 | 30.7 | 31.87 |</p>
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<th>3201</th>
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**Table 2.4-11 Experiments Data of Electrical Motor**

From Eq.(2.2-3), which is the system equation of the electrical motor in terms of current and torque, using curve fitting method, the fitted curve is shown in Figure 1.1-1.

![Figure 2.4-14 Torque-Current Fitting Curve](#)

Where the fitted function is:

\[ f(x) = 1.551x - 4.272 \] (2.4-2)
Comparing to Eq.(2.2-3) \( T = K_i \), where \( i \) corresponds to independent variable \( x \), \( T \) corresponds to function \( f(x) \), \( K_i \) is obtained as 1.551. The last term in Eq.(2.4-2) is a negative constant, which physically means that the torque is produced only after current \( i \) arrives at a constant of \( \frac{1.551}{4.272} \). This may be caused by capacitance which can lead to experiment error of current measure. However, the error doesn’t influence the linear relation between current and torque, and the magnitude is small which can be ignored in the system equation.

Eq.(2.2-5) shows the relation between inductance, resistance, source voltage, and EMF constant. Since inductance is always small enough to be ignored [11], Eq.(2.2-5) can be expressed in matrix form as shown in Eq.(2.4-3)

\[
\begin{bmatrix}
  i_a \\
  \dot{i}_a \\
  \vdots \\
  i_{an} \\
  \dot{i}_{an}
\end{bmatrix}
\begin{bmatrix}
  R_s \\
  K_e \\
  \vdots \\
  R_{sn} \\
  K_{en}
\end{bmatrix}
= \begin{bmatrix}
  v_{a1} \\
  v_{a2} \\
  \vdots \\
  v_{a(n-1)} \\
  v_{an}
\end{bmatrix}
\]  

(2.4-3)

Putting experiments data Table 2.4-11 into Eq.(2.4-3), the matrix form is shown below. Where \( n \) is the number of experiments according to different currents.

\[
\begin{bmatrix}
  i_{a1} & \dot{i}_{m1} \\
  i_{a2} & \dot{i}_{m2} \\
  \vdots & \vdots \\
  i_{a(n-1)} & \dot{i}_{m(n-1)} \\
  i_{an} & \dot{i}_{mn}
\end{bmatrix}
\begin{bmatrix}
  R_s \\
  K_e \\
  \vdots \\
  R_{sn} \\
  K_{en}
\end{bmatrix}
= \begin{bmatrix}
  v_{a1} \\
  v_{a2} \\
  \vdots \\
  v_{a(n-1)} \\
  v_{an}
\end{bmatrix}
\]  

(2.4-4)

The objective here is to solve \( R_s \) and \( K_e \) that best mesh to the experiments data: \( i_{an} \), \( \dot{i}_{mn} \), and \( v_{an} \), which is a two unknowns with \( n \) equations algebraic problem.

To get the most approximate solution of \( R_s \) and \( K_e \) in Eq.(2.4-4), using least square error method, Eq.(2.4-4) can be solved using Eq.(2.4-5) [12].
Solving Eq.(2.4-5), $R_s = 0.2638$ and $K_r = 0.0016$ is obtained. A comparison of experiments data and least square error result is shown in Figure 2.4-15, where the dash line stands for experiments data.

![Figure 2.4-15 Voltage-Current-Speed](image)

Table 2.4-12 shows other parameters obtained from tables [22] [23].
<table>
<thead>
<tr>
<th>$\mu$</th>
<th>$\mu_{GJ}$</th>
<th>$\mu_{LC}$</th>
<th>$K_c$</th>
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<td>0.33</td>
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<td>3500kg/mm</td>
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Table 2.4-12 Parameters from Tables

2.4.4 Module Combination

Combining both input and output of modules created before, a system model of AMT transmission system is obtained, which mainly includes modules of clutch controller, DC motor, clutch actuator, clutch, engine, and powertrain, where the “Controller” module is defined in CHAPTER 3 and the “Powertrain” module includes gear box, differential gear, and vehicle loading. The combined system model is shown in Figure 2.4-16, where.

![Figure 2.4-16 Combined AMT System Model](image)

The main concept of module combination can be decomposed into two procedures. The first is motion transform, which introduces relative motions of each component according to the mechanism connecting. The second procedure computes loading of each mechanical part from system equations, with accordance of forces exerted by other components caused by
relative motions obtained from the first procedure. And the computed loadings of procedure two are transformed to motion by system equations than serve as the input of procedure one. A flowchart of such concept is shown in Figure 2.4-17.

![Flowchart of Modules Combination](image-url)

**Figure 2.4-17 Flowchart of Modules Combination**
CHAPTER 3
DYNAMIC SIMULATION AND OPTIMIZATION

3.1 Introduction

In this chapter, modules of dynamic models created in CHAPTER 2 are proceeded to be simulated and confirmed. Since no experiment data is available in this time, the simulation results are verified with tendencies. After the models are confirmed in 3.2, optimization of some parts of the clutch actuator is introduced in 3.3 to modify the prototype to be more close to the design requirements.

Design requirements of the clutch actuator are shifting speed and stability. Stability is mainly concerned by controller design in CHAPTER 5. Speed of disengage and engage of the clutch actuator is the project in this chapter. After clutch actuator and clutch are assembled, the clutch actuator is expected to drive the clutch to travel 7mm within 0.1 sec, where 7mm is the distance for a clutch traveling from engage position to fully disengaged position. Shorter time available for disengaging is expected. Besides disengaging time, ability of clutch to engage within 0.2 sec is expected too.

3.2 Dynamic Simulation and Tendencies Discussion

3.2.1 Clutch Actuator

Simulation result of the clutch actuator model is shown in Figure 3.2-1. Where clutch actuator receives a maximum supportable voltage of 12V by controller at t=0sec. After the output travel arrives at fully disengaged distance 7mm, the supported voltage is terminated. And after t=0.4sec, a negative maximum supportable voltage -12V is provided to push back the actuator to engage clutch. Note that a maximum current of ±25A are limited to provide to the clutch actuator for electrically concern.
To concern with tendency of the result shown in Figure 3.2-1, some characters of the curve are discussed.

First, the maximum travel surpasses the distance of 7mm where supporting voltage is terminated. Such condition is reasonable owing to no position tracking control is used. The control of the actuator is just a switch from on to off after 7mm. Since no supporting power is available after 7mm, clutch actuator surpasses 7mm by inertia, and stops at about 7.3mm by friction forces and clutch resistance. Such condition can be verified in Figure 3.2-2, which shows acceleration of the clutch travel. Acceleration of clutch travel descends to negative value suddenly after the supported voltage is terminated at about 0.234sec. Besides, the shaded period from 0.234sec to 0.4 sec is caused by numerical computation, which is a state of force equilibrium.
Second, traveling time from 0mm to 7mm is obviously longer than 7mm to 0mm. From data of clutch and clutch actuator, resistance force from clutch is up to 1200N at maximum as shown in Figure 2.3-9, which is a force restricting clutch actuator to travel forward. An assisting force from assist spring (Eq.(2.2-22)) assists clutch actuator to travel forward, however, only a maximum force of 235N is available. Obviously, resistance force is larger than assist force. Thus, a longer traveling time from 0mm to 7mm is reasonable. To make a detail description of the process, Figure 3.2-2 shows that acceleration of forward travel increases at initial, which is because of the increase in transmission angle between linkage and worm gear. But acceleration decreases after about 0.03sec because the resistance force from clutch is increasing. However, since diaphragm spring provides smaller force after the maximum point, as shown in Figure 3.2-2, and the motor driving torque increases because of the slowing down rotation velocity, acceleration increases gradually after clutch travel passes the maximum resistance force point at about 2mm travel at about 0.05sec. In backward process, since resistance force from clutch always provides an assisting force for clutch actuator to travel back, it leads to a larger acceleration than forward process as shown in Figure 3.2-2. However, this force also causes friction forces within the actuator to increase, and then leads to deceleration especially at travel distance about 2mm which corresponds to 0.5sec when the loading forces on clutch actuator is maximum. After 0.5sec, when loading force from clutch decreases, travel acceleration increases gradually. But at position close to 0mm, force from resist spring is increased, which again leads to an increase in deceleration, as the final rising curve in Figure 3.2-2.

Third, the time period where no power is provided (0.2sec~0.4sec), except motion caused by inertia, clutch actuator is suspended without any motion when resisting force from clutch still exerts. This is the character caused by worm shaft - worm gear system called self-lock. If lead angle of the worm gear is small enough, friction force is able to restrict
motion caused by force from worm gear, which is the condition of \( W_{fr} < W_{wf} \) as expressed in Eq.(2.2-9) and Eq.(2.2-10). In the simulated model, where \( \mu = 0.25 \), \( \phi_f = 10.5^\circ \), and \( \lambda = 5^\circ \), such self-lock condition exists. The condition is the high frequency vibratile region shown in Figure 2.3-9 caused by numerical calculation.

To verify the tendency discussion mentioned before, some adjustments of parameters are proceeded to compare the results.

First, to verify the second character of the above tendency discussion, spring coefficient \( K_{sp} \) is increased from 0.6kg/mm to 1kg/mm. According to tendency expectation, forward time should decrease and backward time should increase because of the increase force from resist spring that exerts on the direction of forward travel. The simulated result is shown in Figure 3.2-2.
Figure 3.2-3, with a comparison of original result expressed with dash line.

Figure 3.2-3 Comparison of Different Spring Coefficients

Figure 3.2-3 shows that forward time decreases from 0.23sec to 0.1717sec. However, backward time decrease from 0.129sec to 0.118sec too, which not conform to the tendency expectation. Checking forces that exert on parts, a comparison of friction force between worm gear and worm shaft of the two cases are shown in Figure 3.2-4.
Figure 3.2-4 shows that when spring coefficient is increased, friction force between worm shaft and worm gear become smaller in most of the region. The reason is that assisting spring counteracts more resistance force from clutch when spring coefficient is increased, thus the loading upon clutch actuator is decreased in most regions and friction force is decreased too. Such condition not only appears between worm gear and worm shaft, but also in most parts of the clutch actuator.

Since the increase of spring coefficient increases assist force to travel clutch forward, and reduces friction forces within clutch actuator in most conditions, the decrease of forward time and backward time is reasonable. However, Figure 3.2-4 also shows that increase in spring coefficient also increases friction forces at position near 0mm where clutch loading is small, as shown in the begin and ultimate of the curve in Figure 3.2-4. Such condition can be obvious when $K_{sp}$ is increased too large, for example: 10kg/mm. Clutch actuator can’t actuate in such condition, because force from assisting spring leads to a big loading at initial position and causes friction forces that larger than the electrical motor drivable.
Figure 3.2-3 also shows a larger overshoot after 7mm. Since the hardened assisting spring provides a higher traveling velocity at position of 7mm, the clutch actuator has larger inertia at this point, thus leads to a larger overshoot comparing to previous case. The tendency verifies the inference of first tendency discussion mentioned before.

To verify the third character of the above tendency discussion, lead angle $\lambda$ is increased to $20^\circ$ which leads to a condition where $W_{w_t} > W_{w_f}$. According to tendency expectation, clutch actuator will be pushed backward by resistance force from the clutch. The result is shown in Figure 3.2-5, where supporting voltage is terminated after clutch travel has reached 7mm. And note that spring coefficient of the assisting spring is up to 1.22kg/mm, because the increased lead angle $\lambda$ leads to an increased loading $W_{w_t}$ according to Eq.(2.2-9), thus the clutch actuator can’t travel to 7mm without an increased assisting spring.

Figure 3.2-5 Clutch Travel (mm) vs. Time (sec.) with Increased Lead Angle
Obviously in Figure 3.2-5, clutch actuator is pushed back after supporting voltage is terminated and forward inertia is vanished by attrition. But see also that the forward time is decreased by the increased lead angle $\lambda$ and the increased spring coefficient $K_{SP}$.

Since the simulation result of disengaging time is 0.23 sec which not conforms to the requirement of 0.1 sec, the lever ratio $R_r$ between clutch actuator and clutch is verified to see the tendency of such variation.

Figure 3.2-6 shows curves with different lever ratios: 0.95, 1, and 1.5.

![Figure 3.2-6 Comparison with Different Lever Ratio $R_r$](image)

Figure 3.2-6 shows that increase in lever ratio shall lead to some decrease in disengaging time. However, some decrease in lever ratio shall lead to a large increment in disengaging time. But the decrease in lever ratio is also able to decrease the engaging time. Such
phenomenon shows that lever ratio plays an important rule in the travel speed of clutch actuator.

3.2.2 Transmission System

To observe the simulation results of transmission system, the state of vehicle start is firstly implemented.

For the condition of vehicle start simulation, Clutch is disengaged in initial and engaged gradually until fully engaged after 1 second. For engine throttle position (TPS) control, TPS is zero at time 0, and increases in ramp to maximum TPS 10 at time of 1 second. And the gear ratio is the first ratio of 3.948. As shown in Figure 3.2-7.

![Figure 3.2-7 TPS and Clutch Position while Vehicle Start](image)
The simulation result of vehicle speed is shown in Figure 3.2-8.

![Figure 3.2-8 Vehicle Speed](image)

From Figure 3.2-8, some tendencies can be verified. First, vehicle speed increases rapidly from 0 second to 0.5 second, which is the period of clutch engagement. More obvious circumstances which explain such condition can be seen in Figure 3.2-9 and Figure 3.2-10. Figure 3.2-9 shows that the vehicle has an abnormal rapidly increased acceleration before 0.5 second, and decreases to general acceleration curve after 0.5 second. Figure 3.2-10 shows that the engine speed increases rapidly with the increase of TPS from 0 second. However, the engine speed decreases rapidly from time 0.3 to 0.5. The appearances provide information that engine speed increases rapidly with the rapidly increased TPS. However, with the engaging clutch, a restricting torque from clutch suddenly decreases engine speed and the reaction torque increases vehicle speed. After about 0.5 second, which is a condition that the torque transmittable from clutch is larger than output torque of engine and clutch is fully engaged where no relative motion between friction plate and pressure plate, the structures from engine to drive wheels are completely connected. The vehicle obtains normal output torque from engine after clutch is completely engaged.
Second, the increase of vehicle speed decreases gradually after 0.5 second, a more obvious condition is shown in Figure 3.2-9. Such condition is due to the decrease of engine torque and increase of vehicle resistance such as damping and wind resistance. From Figure 2.3-4, engine torque with TPS=10 decreases gradually after engine speed of 4000 r.p.m.. From Eq.(2.3-17), the increase in vehicle speed shall increase the resistant force on vehicle. Besides, damping on all parts shall increase resistant forces with increases of speeds. Thus, it is reasonable that vehicle acceleration decreases with increase of vehicle speed, and a rapider
decrease of vehicle acceleration after 4000 r.p.m. at about 2.5 second is another testimony of such condition.

Verifications above are also confirmed by experiences. A greater acceleration is always felt by a driver when clutch engages to start to drive a car, especially when engage speed is high. Besides, even throttle control is always at the bottom, car speed will not increases with no limit, acceleration always decreases after some speed.

To verify the model, some parameters are changed to check the tendencies. First, gear ratio at start is changed from first gear ratio to second gear ratio. According to the forecast of experiences, the acceleration should smaller than start with first gear ratio and the final speed should larger than first gear ratio because the second gear ratio should has higher saturation speed.

The simulation results of vehicle start with second gear ratio are shown in Figure 3.2-11 and Figure 3.2-12, where the dash lines are comparisons of results of first gear ratio.

![Vehicle Speed](image-url)

Figure 3.2-11 Vehicle Speed
Obviously in Figure 3.2-11, start with second gear ratio leads to a slower speed at initial, but surpass the first gear ratio after some period. Such result fits with the experience forecast before.

On the other hand, Figure 3.2-12 shows a condition that the second gear ratio needs a longer time to full engage the clutch. Full engage is completed within about 0.5 second by first gear ratio. However, it needs about 0.6 second to finish by second gear ratio. Such situation can be felt in experiences. Theoretically, second gear ratio provides a larger inertia moment to engine. It needs more impetus to drive the vehicle with second ratio. Since the power from engine is the same, a longer time is required to synchronize speed of engine and vehicle, and a larger clutch torque is required. Figure 3.2-13 shows such condition more clearly. Since TPS control is the same, the curves before full engage is matched. However, after ratio one is fully engaged, vehicle speed of ratio two has not reached the speed synchronized with engine speed. Thus, a longer time for clutch to engage more compact, to slow down engine speed, and to speed up vehicle speed is required.
Second, control of TPS is verified. TPS is varied from 10 to 5. According to experiences, vehicle speed should be smaller since engine provides lesser power. The simulation result is shown in Figure 3.2-14, which clearly shows that speed of vehicle with TPS=5 is always lower than vehicle with TPS=10.
A clearer phenomenon is displayed on engine speed diagram, as shown in Figure 3.2-15.

![Engine Speed Diagram]

**Figure 3.2-15 Engine Speed**

Comparing Figure 3.2-14 and Figure 3.2-15, it also shows that saturate speed is smaller when TPS is smaller.

Third, the clutch engage time is verified to simulate a condition where clutch control is gentler. The engage time is verified from one second to three seconds. Besides, in order to avoid too large overshoot of engine speed caused by the lower engaging speed, TPS control is also modified to a smoother ramp as shown in Figure 3.2-16.
According to tendency forecast and experiences, a more gradual engagement should lead to smoother vehicle acceleration. The simulation results are shown in Figure 3.2-17 and Figure 3.2-18. From Figure 3.2-17, with a comparison of previous curve shown in dash line, vehicle speed curve is smoother, which has more gradual speed increase. Such condition is clearer in Figure 3.2-18. Acceleration rises more peacefully with longer engaging time, and a smaller maximum acceleration is also received, which leads to a smoother acceleration change and thus a more comfortable driving condition.

Still another character in Figure 3.2-17 is worth to be concerned. Ends of two curves shown in Figure 3.2-17 are joined together. Such condition shows a fact that in a desired gear ratio and TPS value, saturate speed is the same whatever TPS control or clutch control is.
Theoretically, saturate speed only depends on gear ratio and final TPS value. Such conditions can be verified in simulation results shown in Figure 3.2-11, Figure 3.2-14, and Figure 3.2-17.

![Figure 3.2-17 Vehicle Speed](image)

**Figure 3.2-17 Vehicle Speed**

![Figure 3.2-18 Vehicle Acceleration](image)

**Figure 3.2-18 Vehicle Acceleration**

Forth, an outside affection is verified. Slope of the road where vehicle is started is varied from zero degree to five degree. Such verification of road affection is important. Start at a ramp is difficult for manual transmission. It’s also a challenge for automated manual transmission system whose clutch control is administrated by clutch actuator and electrical
control unit (ECU).

From experiences, if no proper control of clutch and throttle is exercised while starts at a ramp, misfire may occur or vehicle may fall back at initial because no enough power is exercised on vehicle to resist the gravity. Simulation results where clutch control and TPS control is as general are shown in Figure 3.2-19 and Figure 3.2-20.
Since the control of clutch and TPS is the same as general as in Figure 3.2-7, Figure 3.2-19 shows that vehicle speed is negative in some period at initial, and after torque transmitted from clutch is high enough to resist gravity, vehicle speed begin to rise to positive value, which conforms with experiences. Figure 3.2-20 shows a condition which may cause misfire. From Figure 3.2-20, a comparison of zero degree and five degree slope represents that engine speed in five degree slope is dropped to a lower speed while fully engaged. If such condition is magnified, for example: no enough engine speed at initial or clutch engages too soon, such lower speed may lower than the speed engine maintainable to keep driving and than leads to misfire. The situation is shown in Figure 3.2-21, where TPS is reduced to 1 which leads to a slower engine speed. Obviously, a negative engine speed is caused while fully engaged as a mean of misfire.

![Figure 3.2-21 Engine Speed](image)

**3.2.3 Shifting Process**

After simulation results of transmission system of vehicle start has been verified in 3.2.2, simulation of shifting process is proceeded in this subsection.
Since TPS control is not the emphasis of this study, the control of TPS only use general concepts in the simulations.

The simulation result of shifting process is shown in Figure 3.2-23 and Figure 3.2-24, where the clutch position, TPS, and gear ratio control is shown in Figure 3.2-22.

![Figure 3.2-22 Control of Shifting Process](image-url)
From Figure 3.2-23, it is obvious that vehicle speed decreases after clutch is disengaged, and vehicle regains velocity when clutch is engaging. Such state is identical from experiences.

For a more detail view in Figure 3.2-24, like vehicle start as discussed in 3.2.2, vehicle suffers a larger acceleration when clutch engage because of the synchronization of engine and transmission shaft. And the unusual acceleration will be reduced to general acceleration after
fully engaged.

Like Figure 3.2-23, Figure 3.2-24 also shows the affection of disengages. Vehicle acceleration reduces to negative values when clutch is fully disengaged. The deceleration is cause by vehicle loading as mentioned in Eq.(2.3-15), which is also felt in our driving experiences on manual transmission vehicle.

Besides, comparing vehicle acceleration of gear ratio one, gear ratio two, and gear ratio three, it is obvious that first gear ratio provides larger acceleration than second gear ratio, and so does second gear ratio larger than third gear ratio. It can be easily verified by our experiences that we always down-shift to overtake a car.

On the other hand, Figure 3.2-24 also shows the affection of synchronizer while shifting gear ratio. Shifting of synchronizer always causes a small acceleration when up-shifting and deceleration when down-shifting. Such acceleration/deceleration is small and swift which always cause little sense for people on vehicle. However, it can always be measure by experiments [ITRI].

To verify the transmission system model, a shifting process which includes both up-shifting and down-shifting is simulated as shown in Figure 3.2-26 and Figure 3.2-27, where the gear ratio command is shown in Figure 3.2-25.
From Figure 3.2-26, it is obvious that when down-shifting, clutch engage causes deceleration instead of acceleration. An obvious view in vehicle acceleration curve is shown in Figure 3.2-27. This character is caused by the relative lower engine speed comparing to vehicle speed when gear ratio is shifted to a lower gear ratio. Such character is usually felt
when down-shifting. Sometimes such skill is used by professional drivers as “engine brake”.

On the other hand, in general conception, vehicle should be able to perform a higher acceleration after gear ratio is shifted to a lower ratio. However, according to Figure 3.2-27, simulation result shows that the acceleration is lower than prior gear ratio, even lower to deceleration. Such phenomenon is caused by the over-speed of engine. As shown in Figure 3.2-28, after down-shifting, engine speed is raised to about 9000 r.p.m. by vehicle inertia, which is out of general-used range of engine speed, and thus the engine is able to provide only very small torque to vehicle as shown in Figure 2.3-4. If the torque is too small to overtake vehicle loading, deceleration as shown in the final part of Figure 3.2-27 is caused.

![Figure 3.2-28 Engine Speed](image)

**Figure 3.2-28** Engine Speed

### 3.3 Structure Optimization

In this section, some parts of the clutch actuator are optimized to perform a better disengaging time. In the following subsections, cost function, design variables, and
constraints are defined according to various times of simulation experiences and engineering estimation. And finally in 3.3.4, optimization is executed to modify the original prototype.

3.3.1 Cost Function

The time needed for clutch actuator to fully disengage a clutch is defined to be the cost function of the optimization.

There are two key points to evaluate an automated manual transmission system: comfort and shifting time. The comfort mainly depends on the control of clutch, however, the mechanical structure of clutch actuator should still be able to provide enough movability to exercise according to the control command, and such demand is required in the constraints of the optimization. The main challenge for mechanical structure of clutch actuator is disengaging time, which mainly dominates the shifting time. Shifting time includes disengaging time, synchronizer shifting time, and engaging time. Shifting time of gear ratio by synchronizer is dominated by shifting actuator of gear box, which is not the emphasis of this study. Engaging time always depends on shifting process; it differs with different gear ratios, engine speed, vehicle speed, etc., and speed of such process is not required generally, because it should proceed smoothly to provide a gradual variation of acceleration/deceleration for vehicle as discussed in 3.2.3. Thus, minimization of disengaging time is the chief factor to minimize disengaging time, which is chosen to be the cost function here.

3.3.2 Design Variables

In order to be more practical for the optimization, parts which are easier to be modified and contributive for minimize disengage time are chosen to be design variables.

All parameters in the clutch actuator are listed in “System Parameters” from Table 2.4-1 to Table 2.4-4. To avoid variation of the main structure, design variables are chosen to be
parameters workable without changing of main dimensions, like lead angle on worm, spring coefficient of assisting spring (pre-stressed spring), clutch lever ratio, etc..

The first design variable chosen is lead angle of the worm. Lead angle dominates gear ratio between worm shaft and worm gear, as mentioned in 2.2.4. Since electrical motor provides different torques with different rotation speeds, adjustment of lead angle must modify the speed range of motor rotation, thus provides a more efficient output torque and relative motion. Such affection has been shown in Figure 3.2-5.

Second, spring coefficient of the assisting spring is chosen. As shown in Figure 3.2-3, the increase of spring coefficient can reduce both engaging time and disengaging time. On the other hand, pre-deformation of the spring is worth to be chosen as design variable too. Different pre-deformation provides different distribution of assisting force. For a fixed pre-stressing force, a larger pre-deformation provides a more uniform assist force, and a smaller pre-deformation provides smaller assisting force after full-traveled. Such affection not only provides different assisting forces, but also influences affections like friction forces within the actuator and the motor torque distribution. Besides, both changes of spring coefficient and spring pre-deformation effects spring mass. Spring coefficient is mainly affected by applied material and coil dimension, such affections on mass can be adjusted by different designs, it has no direct and absolute relation with spring mass. Thus, the affection is ignored in the optimization. However, since the pre-deformed length of the spring is fixed, change of pre-deformation means change of initial length, which directly changes the spring mass. For general conception, spring mass $M_{spring}$ should be proportional to length of the spring $s_{ins}$.

Third, lever ratio between clutch actuator and clutch is chosen. As shown in Figure 3.2-6, lever ratio influences disengaging time obviously. Different lever ratios cause different
rotation speeds of electrical motor, different loading forces on the clutch actuator, and different clutch actuator travel distances, which are all influential to disengage time.

The design variables are marshaled in Table 3.3-1.

<table>
<thead>
<tr>
<th>Design Variables</th>
<th>Lead Angle</th>
<th>Spring Coefficient</th>
<th>Pre-Deformation</th>
<th>Spring Mass</th>
<th>Lever Ratio</th>
</tr>
</thead>
<tbody>
<tr>
<td>Symbol</td>
<td>$\lambda$</td>
<td>$K_{sp}$</td>
<td>$s_{ins}$</td>
<td>$M_{spring}$</td>
<td>$R_y$</td>
</tr>
</tbody>
</table>

| | | $M_{spring}$ | | | $(\frac{M_{spring}}{s_{ins}}-s_{ins}^{n})$ | |

Table 3.3-1 Design Variables

3.3.3 Constraints

To avoid unimplementable optimization result, some constraints are defined in this subsection.

First, since cost function of the optimization is to minimum disengaging time, disengaging time should be minimized absolutely. However, such result may also extend or shorten the engaging time. As mentioned in 3.3.2, even the engaging time is not demanded to be shorten, clutch actuator should still has a movability to engage a clutch within a desired time. The desired engaging time is 0.3 second at minimum in general. Thus, the first constraint is set to be the engaging time shorter than 0.3 second.

Second, clutch actuator should be set to be workable at any state and any time within the working space. The clutch actuator not only executes the command as shown in Figure 3.2-22, which travels directly to disengage and engage, but also many other types of commands. In some special cases, control command for the clutch actuator may similar to Figure 3.3-1, where disengaging command is interrupted at some place between fully disengaged and fully engaged, and restarts to disengage after a period of time. However, such command is not
workable for some improper design, for example: the original prototype. The simulation result of such command applied on the prototype is shown in Figure 3.3-2. The same result is experienced on the prototype. The reason for such result that clutch actuator can not be re-actuated after the suspense can be explained in Figure 3.3-3, which is the simulation result of acceleration curve of worm shaft administrated by a command the same with Figure 3.2-1. Figure 3.3-3 shows that the direction of worm shaft acceleration is opposite to travel direction in some periods, which means that the drive torque from electrical motor is smaller than outside loading and inside resistances of the clutch actuator in these periods. The clutch actuator travels through these periods only by inertia. Thus, if no enough inertia is available in these periods, clutch actuator is not workable, as the case result shown in Figure 3.2-2. Such state should be avoided. Thus, a constraint that direction of worm shaft acceleration be the same with the direction of clutch travel command when a maximum torque from electrical is applied is required.

Figure 3.3-1 Clutch Actuator Travel Command
Figure 3.3-2 Clutch Actuator Travel

Figure 3.3-3 Worm Shaft Rotation Acceleration
Third, the property of self-lock between worm shaft and worm gear as mentioned in 3.2.1 should be kept. Self-lock property provides a character that the clutch actuator can be driven only by worm shaft (motor), force from clutch that acts on worm gear can’t drive the actuator. Such property provides an important advantage for electrical motor that electrical motor works only at the time of shifting. If no such property is applied, electrical motor should be actuated all the time to resist forces from clutch and resist spring (pre-stressed spring), thus largely decreases life-span of the electrical motor. Besides, self-lock property also provides perfect stability at steady state.

To keep self-lock property on worm, tangential force from worm gear $W_{ft}$ should be smaller than tangential force caused by friction $W_{ftf}$. From Eq.(2.2-9) and Eq.(2.2-10), the following relation of lead angle $\lambda$ should be constrained.

$$\tan \lambda < \frac{\mu}{\cos \phi_n};$$

$$\lambda < 14.81^\circ$$

Forth, under the constraints of space of the structure, pre-deformation $s_{ins}$ should be constrained less than 100mm, and lever ratio $R_c$ should be set between 0.3 and 3.

All the constraints are marshaled in Table 3.3-2.

<table>
<thead>
<tr>
<th>$T_{engage}$</th>
<th>0.3 second</th>
</tr>
</thead>
<tbody>
<tr>
<td>$\lambda$</td>
<td>&lt;14.81°</td>
</tr>
<tr>
<td>$TI \times (TI - WW_f \frac{dw}{2} - W_{ftf} \frac{dw}{2}) &gt; 0$</td>
<td></td>
</tr>
<tr>
<td>$s_{ins}$</td>
<td>≤100 mm</td>
</tr>
<tr>
<td>$0.3 \leq R_c \leq 3$</td>
<td></td>
</tr>
</tbody>
</table>

**Table 3.3-2 Constraints**
3.3.4 Optimization Implement

According to the definitions in previous subsections, and initial condition from the prototype (besides the lever ratio, which is modified from 1 to 2 to conform the constraint as discussed in Figure 3.3-3), optimization of mechanical structure of the clutch actuator is executed in this subsection using Matlab®.

Optimization Toolbox within Matlab® is used to deal with the problem. Since the optimization problem is defined as a nonlinear constrained multivariable problem, “fmincon”, which is used to find a minimum of a constrained multivariable function, is chosen to compute the problem.

“fmincon” deals with the constrained problem using Sequential Quadratic Programming (SQP) method (also known as Constrained Variable Metric (CVM) or Recursive Quadratic Programming (RQP) [25]). SQP method uses Kuhn-Tucker (KT) equation as basis. SQP method attempts to compute the Lagrange multiplier directly. Constrained quasi-Newton method guarantees superlinear convergence by accumulating second order information regarding the KT equations using quasi-Newton updating procedure [28].

There are three main stages to implement SQP method. The first is updating of the Hessian matrix. At each major iteration, a positive definite quasi-Newton approximation of the Hessian of the Lagrangian function is calculated using BFGS (Broyden-Fletcher-Goldfarb-Shanno) method. The second is to compute Quadratic Programming QP solution. At each major iteration, a QP problem, which is a subproblem generated from Hessian of the Lagrangian function calculated before, is solved, and the solution is used to form a search direction for a line search procedure. The third is line search and merit function calculation. Using the search direction produced in QP problem, a step
length which is sufficient to decrease a merit function is determined, where the merit function is in the form defined by Han [26] and Powell [27] [28].

Using “fmincon” as the implement program, and call the dynamic model created in Simulink® as cost function, the optimization computes with the following results.

<table>
<thead>
<tr>
<th>Design Variables</th>
<th>Initial Condition</th>
<th>Optimization Results</th>
</tr>
</thead>
<tbody>
<tr>
<td>Lead Angle λ</td>
<td>5 °</td>
<td>11.14 °</td>
</tr>
<tr>
<td>Spring Coefficient $K_{sp}$</td>
<td>0.6 kg/mm</td>
<td>0.854 kg/mm</td>
</tr>
<tr>
<td>Pre-Deformation $s_{ins}$</td>
<td>40 mm</td>
<td>72.93 mm</td>
</tr>
<tr>
<td>Lever Ratio $R_e$</td>
<td>2</td>
<td>1.207</td>
</tr>
</tbody>
</table>

**Cost Function $T_{disengage}$**

<table>
<thead>
<tr>
<th></th>
<th>Initial Condition</th>
<th>Optimization Results</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>0.234 second</td>
<td>0.0896 second</td>
</tr>
</tbody>
</table>

Table 3.3-3 Optimization Results

![Optimization Result](image)

**Figure 3.3-4 Optimization Result**

From Table 3.3-3 and Figure 3.3-4, it is obviously shown that the clutch disengaging time is reduced from 0.234 second, which not conforms to design requirement of 0.1 second, to 0.0896 second, which is in the range of design requirement. And from the results of optimization, all the constraints, including the violated constraint in the prototype that direction of the worm shaft acceleration was not always kept the same with control command.
direction even if a maximum torque from motor is applied, is conformed.

For an extra consideration, since the optimization result of the lead angle is near the constraint, another optimization is implemented with a larger friction coefficient on worm \( \mu = 0.3 \), which provides an extensive constraint for lead angle. Such computation provides a better result of cost function \( T_{\text{disengage}} = 0.0798 \) second, which physically means torque provided by the motor is sufficient to give better performance for the clutch actuator. However, such alteration of part material is not the scope of the study. The result just gives a suggestion of tendency for designers.
CHAPTER 4
DEVELOPMENT OF CONTROL FUNCTION

4.1 Introduction

In this chapter, control function of the clutch actuator, according to the dynamic model created before, is built up.

In general, steps to develop a control function are firstly to understand the process and translate dynamic performance requirements into time, frequency, or pole-zero specification, and then make a linear approximate model. Based on the linear approximate model, sketch a frequency response (Bode plot) and a root locus to form an initial estimate control function of the complexity of the design model. Finally, if the trial-and-error compensators do not give entirely satisfactory performance, consider a design based on optimal control. The symmetric root locus shows possible root locations from which to select locations for the control poles that meet the response specification [10].

In this study, instead of a linear approximate model, a complete model which is more close to the real system is exercised, with an expectancy of better outcome than general process.

To deal with such problem, the dynamic model is seen as a black box. The first estimate of the control parameters can base on turning methods such as Ziegler-Nichols (Z-N) method and Internal Model Control (IMC) method, which use some strategies to forecast the system dynamic characters, and then using optimization methods to optimize the control parameters to be able to control the process more close to the setpoint. Theoretically, such manner can provide a better control performance than general process, because of the complete controlled model, a model more close to reality, is applied.
According to these processes, control function with firstly estimated control parameters is built up in this chapter. And in the next chapter, optimization will be implemented to modify the control parameters using Matlab®. Such combination of dynamic modeling, dynamic simulation, part optimization, control function design, control function optimization, and system simulation within a unit of interface are the speciality of this study.

4.2 Control Strategies

According to the dynamic model created in CHAPTER 2, the clutch actuator seems can be controlled immediately with an open loop, since it looks as though that the dynamic model has included all the outside affections on the clutch actuator, the clutch loadings. However, in reality, vibration, heat affection, rotation speed, etc. may affect the clutch resistance force that applies on clutch actuator as an outside loading. The dynamic model just gives a general case of operations. Thus, a feedback control is required absolutely for a reliable control.

To define the control problem, ascribing the affections from engine, transmission shaft, vibration, heat affection, rotation speed, etc., which work upon the clutch, as outside loadings of the clutch/clutch-actuator system, the clutch/clutch-actuator system can be seen as an independent dynamic system to be controlled.

To formulate the problem, first, the control signal of the electrical motor is chosen to be input of the dynamic system, or output of the control function. The DC motor used on the clutch actuator is generally controlled by Pulse-Width Modulation (PWM). Such frequency control with a fixed input voltage can be seen as a transformation from voltage control. To be convenient, the DC motor control signal is chosen to be voltage control. Second, for output of the dynamic system, or feedback signal of the control function, the best choice is the torque transmit ability of the clutch \( T_{cf} \), which is a final output of the clutch/clutch-actuator system.
However, such parameter is not easy to be measured in practice, so does the normal force $N_{clu}$ which has a direct relation with $T_{cf}$. Thus, a most reliable choice for the dynamic system output, or feedback of the control function, is travel position of the clutch, or travel position of the clutch actuator $D_{out}$, which is also practical in general AMT clutch control.

However, control of such dynamic system with a single loop is not enough. In general, there should be one more loop for the control of the driving motor. According to Eq.(2.2-3), motor rotation angle $\theta_m$, which has an immediate relation with system output $D_{out}$, is controlled directly by armature current $i_a$. Since the control input is voltage of the motor $v_a$, the transformation from $v_a$ to $i_a$ can be seen as another control loop with a plant gain as Eq.(2.2-5), and suffers outside loadings from motor speed $\dot{\theta}_m$, as stated in Eq.(2.2-5), and other environment effects.

The block diagram of the control loop is shown in Figure 4.2-1.

![Figure 4.2-1 Control Loops](image)

For the strategy of the control plants, Proportional-Integral-Derivative (PID) control method is used. There are many methods to control a feedback loop, such as PI control, PD
control, PID control, and many other sophisticated control methods. In general, PI controller and PID controller are most practical in commercial, and thus with lower costs. PI control is adequate for all processes where the dynamics are essentially of the first order. If the step response of the dynamic system looks like that of a first-order system or, more precisely, if the Nyquist curve lies in the first and the forth quadrants only, then PI control is sufficient. PID control is sufficient for process where dominant dynamics are the second order. And it is beneficial when tight control of a higher-order system is required. It can also speeds up the transient response of the system. A sophisticated control, a higher order control, can provide high quality to the controlled system, however, such strategy always be of a significant economic value. Since clutch control of AMT system requires a fine transient response, a low cost for commercial, and dynamic system of the clutch/clutch-actuator is a high order system more than two, as Eq.(2.3-3) and Eq.(2.3-4), PID control is chosen to be the control method. Thus, in the control function, the two control plants, as shown in Figure 4.2-1, are both dominated by PID control strategy.

The PID controller was first described by Callender et al. (1936). This technology was based on extensive experimental works and simple linearized approximations to the system dynamic. It led to standard experiments suitable to application in the field and eventually to satisfactory “turning” of the coefficients of the PID controller. It also results in the development of a comprehensive set of technologies for the design of servomechanisms, which is a control method that focuses on the control of output position as in this study [10]. The standard form, or ISA form, of PID algorithm is shown in Eq.(4.2-1), where \( y_{sp}(t) \) is the set point, \( y(t) \) is the real output, which is also feedback signal of the control function, \( e(t) \) is the control error, \( u(t) \) is output of the control function or input of the dynamic system, and \( K \), \( T_i \), and \( T_d \) are the three main parameters of PID control.
The three main parameters in PID control, $K$, $T_i$, and $T_d$, are ratios that adjust the control error, time integral of the error, and the time rate of change of the error. The increase of the control error, which means increase of $K$, is effective in reducing the error of the system. However, an over-large $K$ may typically leads to instability. The increase of integral of the error, which means decrease of $T_i$, is effective in reducing or eliminating constant steady-state errors, but this benefit typically comes with the cost of worse transient response. The increase of differential of the error, which means increase of $T_d$, can improve the stability of the system, or physically means increase of the damping effect. Such property of differential is able to “predict” the system output. However, as the physical meaning, a larger $T_d$ will leads to a slower system response, thus a longer rise time. The block diagram in Matlab® of such standard form is shown in Figure 4.2-2.

Since the differential term in PID control has the ability of “predict”, a better performance by a modified PID algorithm is obtained if the “predict” term interacts other terms. Block diagram of such algorithm is shown in Figure 4.2-3, and the modified transfer function is shown in Eq.(4.2-2). Such algorithm is also more commercial in practice.
The derivative term in PID control may result in difficulties that should be prevented. If the control error \( e(t) \) is sinusoidal, for example: \( a \sin \omega t \), and high frequency, the differential term may lead to an arbitrarily large amplitude:

\[
    u_d(t) = KT_d \frac{de(t)}{dt} = aKT_d \omega \cos \omega t
\]

Such high frequency gain of the derivative term is therefore limited to avoid noise amplification. This is done by implementing the derivative term as Eq.(4.2-3). And the transfer function form is shown in Eq.(4.2-4).

\[
    u_d(t) = \frac{T_d}{N} \frac{dD}{dt} + KT_d \frac{de(t)}{dt}
\]

\[
    D(s) = \frac{sKT_d}{1 + s \frac{T_d}{N}} e(t)
\]

Where \( N \) is a constant with typical value from 8 to 20.

Defining of setpoint weightings are also improvable of reducing steady-state errors and large transients. Setpoint weightings are used to provide different control errors \( e(t) \) to each proportional, integral, and derivative term. For example, in general form of PID control, the control function should be modified to Eq.(4.2-5).
\[ u(t) = K(e_p(t) + \frac{1}{T_i} \int_0^t e(\tau) d\tau + T_d \frac{de_d(t)}{dt}) \]  \hspace{1cm} (4.2-5)

where:

\[ e_p(t) = b y_{sp}(t) - y(t) \]
\[ e_d(t) = c y_{sp}(t) - y(t) \]

\( b \) and \( c \) are constants of setpoint weighting. But note that these two constants are typically 0 or 1 in commercial controllers.

From the general form, and modify with interacting modification, derivative limitation, and setpoint weighting, an improved PID control algorithm, which is used in the clutch actuator control function, is expressed in Eq.(4.2-6).

\[ U(s) = K((b + \frac{1}{sT_i}) + \frac{1+sT_d}{1+sT_i} \frac{1}{N}) - \frac{1}{sT_i} \frac{1+sT_d}{1+sT_i} \frac{1}{N} Y(t) \]  \hspace{1cm} (4.2-6)

Besides, both control plants, as shown in Figure 4.2-1, have saturate outputs. In the control function of the clutch actuator, the first control plant, PID1 in Figure 4.2-1, is limited at 25/-25, which is the maximum current restricted for the motor, and the second control plant, PID2 in Figure 4.2-1, is limited at 12/-12, which is the maximum voltage provideable to the motor.

Integrator windup is also established in the control function, which gives upper and lower limits for integrators. The limits are the same with the saturate outputs of each control plants.

The turning of these control parameters should depend on the system dynamic characters. Methods used in this study are introduced in the next section. According to these methods, initial estimates of the control parameters are obtained and are used as initial values of the
optimization implemented in CHAPTER 5.

4.3 Turning of PID Parameters

In this section, parameters of the control functions, the two control plants with PID algorithm as shown in Eq.(4.2-6), are determined. Such parameters are used to be the initial values of the optimization implemented in CHAPTER 5.

To design the parameters of the PID control function, it is necessary to understand what the primary goal the control is. The two common types are setpoint tracking and disturbances adjusting. In the control of the clutch actuator, in general operation, most disturbances are not abruptly, which always affect gradually with gradual variations, for example: temperature increase. Setpoint of the clutch control always works with sudden changes, which requires clutch to be suddenly disengaged and engaged. And the most difficulty for the control of clutch actuator is to cope with the resistant force from clutch, which is in a high order form. Obviously, the clutch control should focus on setpoint tracking more than disturbances adjusting. Such tendency is taken as a base to choice turning methods for the PID control functions.

In the control function, Zinger-Nichols (Z-N) method is used to design parameters within the first control plant, and Internal Model Control (IMC) method is used to design parameters within the second control plant. There are many turning methods for PID control, for example: Z-N, IMC, ISE, ITAE, Cohen-Coon…etc. Both Z-N method and IMC method are good for setpoint tracking [30].

IMC method is very good for setpoint tracking, and disturbances can be completely eliminated theoretically. Steady state error can be almost zero if the controlled plant is realized thoroughly. But the premise is the system can not be too complicate, otherwise the
controller will be in very high order terms. The control loop of PID2 from motor voltage to motor current is a first order system, as shown in Eq.(2.2-5). IMC method suits for such system very well.

Z-N method is widely used and is easy to find appropriate parameters from a complex model. But it is sometimes insufficient for perfect solution. However, for initial values of optimization, it is enough and efficient. Therefore it is chosen to be the turning method of first control plant that controls the clutch position.

Besides the turning methods design the primary parameters $K$, $T_i$, and $T_d$, other parameters, derivative limitation $N$ and setpoint weighting $b$ and $c$, should also be defined. The typical values of $N$ are 8 to 20, an average value of 14 is chosen in initial. And both $b$ and $c$ are 0 or 1 in commercial controller, both $b$ and $c$ are set to 1 in initial.

The two turning methods used to design parameters of the two control plants are introduced and implemented in the following subsections.

### 4.3.1 Internal Model Control

The internal model principle is a general method for design of control systems that can be applied to PID control. A block diagram of such a system is shown in Figure 4.3-1.

![Figure 4.3-1 Block Diagram of Internal Model Control](image)
In the diagram, it is assumed that all disturbances acting on the controlled system are reduced to an equivalent disturbance $d$ at the controlled system output. In the figure, $G_m$ denotes a model of the controlled system, $G_m'$ is an approximate inverse of $G_m$, and $G_f$ is a low-pass filter.

If the model match the controlled system, i.e., $G_m = G_p$, the signal $e$ is equal to the disturbance $d$ for all control signals $u$. If $G_f = 1$ and $G_m'$ is an exact inverse of the process, then the disturbance $d$ will be canceled perfectly. The filter $G_f$ is introduced to obtain a system that is less sensitive to modeling error. A common choice is $G_f = 1/(1+sT_f)$, where $T_f$ is a design variable.

The controller obtained by the internal model principle can be represented as an ordinary series controller with the transfer function shown in Eq.(4.3-1)

$$G_c = \frac{G_f G_m'}{1 - G_f G_m' G_m} \quad (4.3-1)$$

From this expression, it follows that controller of this type cancels poles and zeros of the controlled system.

By making special assumptions, it is possible to obtain PI or PID controller from the principle. The approximated equations are [29]:

$$PI: \quad G_c(s) = \frac{1+sT}{K_p s(L+T_f)}$$

$$PID: \quad G_c(s) = \frac{(1+sT)(1+sL/2)}{K_p s(L+T_f)}$$
where the special assumptions are [29]:

\[
G_p(s) = \frac{K_p}{1 + sT} e^{-sl}
\]

\[
G_m'(s) = \frac{1 + sT}{K_p}
\]

\[
G_f(s) = \frac{1}{1 + sT_f}
\]

Such Internal Model Control (IMC) method is used in the second control plant as shown in Figure 4.2-1, because the controlled system \( G_p \) is in a simple and reliable term as stated in Eq.(2.2-5).

From Eq.(2.2-5), the transfer function of \( G_p \) can be expressed as Eq.(4.3-2), where the term \( K_i \dot{\theta}_m \) in Eq.(2.2-5) is seen as a disturbance.

\[
G_p = \frac{1}{L_a s + R_a}
\]

(4.3-2)

Since the system equation of the electrical model, Eq.(2.2-5), is reliable based on many experiences, and the parameters according to experiences are verified as in Table 2.4-11 and Figure 2.4-15, it is reasonable to choose \( T_f \) to be 0 for the confidence from experiments and theories. However, even experiences can have errors and the dynamic characters may change with working condition. Still a small constant 1e-3 is assigned to \( T_f \).

From Eq.(4.3-2), transfer functions of \( G_m \) and \( G_m' \) are shown in Eq.(4.3-3).

\[
G_m = \frac{1}{L_a s + R_a}
\]

\[
G_m' = L_a s + R_a
\]

(4.3-3)

Where \( L_a = 0.001355 \) and \( R_a = 0.2638 \) according to experiences as discussed in 2.4.3.
Block diagram of the control loop is shown in Figure 4.3-2.

Figure 4.3-2 Block Diagram of Second Control Plant

Figure 4.3-3 shows simulation result of the control plant, where a sinusoid setpoint and an impulse setpoint are implemented. It is obvious that errors are small in both transient state and steady state. It also shows a good disturbance adjusting ability. In the scale views, where clutch actuator traveled to the upper limit and suffered a collision, the controller can eliminate the interference quickly and steadily.
Figure 4.3-3 Simulation Results of Second Control Plant

Since the performance of the control plant designed by internal model method is good, such strategy and parameters are adopted directly into the final design, and is not required to submit into optimization implementation in CHAPTER 5.
4.3.2 Ziegler-Nichols

Two classical methods for determining the parameters of PID controller were presented by J.G. Ziegler and N.B. Nichols in 1942. These methods are still widely used today. They often form the basis for tuning procedures used by controller manufacturers and process industry. The methods are based on determination of some features of the controlled dynamics.

The first method is step response method. Z-N recognized that the step response of a large number of process control system exhibits a process reaction curve like that shown in Figure 4.3-4.

![System Reaction Curve]

This curve can be generated from either experimental data or dynamic simulation of the plant. The S-shape of the curve is characteristic of many high-order system, and such plant transfer function may be approximated by Eq.(4.3-4).
\[
\frac{Y(s)}{U(s)} = \frac{Ke^{-ts}}{\tau s + 1} \quad (\text{4.3-4})
\]
which is simply a first-order system plus a time delay of \(t_d\) seconds. The constants in Eq.(4.3-4) can be determined from the unit step response of the controlled system. If a tangent is drawn at the inflection point of the reaction curve, then the slope of the line is \(R = K / \tau\) and the intersection of the tangent line with the time axis identifies the time delay \(L = t_d\).

Such method is based on a decay ratio of approximately 0.25. This means that a dominant transient decays to a quarter of its value after one period of oscillation. A quarter decay corresponds to \(\xi = 0.21\) and is a good compromise between quick response and adequate stability margins. The regulator parameters suggested by Z-N for the common controller terms are shown in Table 4.3-1 [10].

<table>
<thead>
<tr>
<th>PID Controller</th>
<th>1.2 (\times) (RL)</th>
<th>2(L)</th>
<th>0.5(L)</th>
</tr>
</thead>
<tbody>
<tr>
<td>(K)</td>
<td>1.2 (\times) (RL)</td>
<td>2(L)</td>
<td>0.5(L)</td>
</tr>
<tr>
<td>(T_i)</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>(T_d)</td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

Table 4.3-1 Step Response Method [10]

The second method is frequency response method. This method is based on a simple characterization of the controlled dynamics. The design is based on knowledge of the point on the Nyquist curve of the process transfer function where the Nyquist curve intersects the negative real axis. This point is characterized by the parameters \(K_u\) and \(T_u\), which are called the ultimate gain and the ultimate period. These parameters can be determined in the following way. Connect a controller to the controlled system, set the parameter so that action is proportional, i.e., \(T_i = \infty\) and \(T_d = 0\). Increase the gain slowly until the process starts to continuously oscillate. The gain when this occurs is \(K_u\) and the period of the oscillation is \(T_u\). The formulas for the parameter of the controller in terms of the ultimate gain and the
ultimate period of the controller are shown in Table 4.3-2 [29].

<table>
<thead>
<tr>
<th>PID Controller</th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>$K$</td>
<td>$0.6K_u$</td>
</tr>
<tr>
<td>$T_i$</td>
<td>$0.5T_u$</td>
</tr>
<tr>
<td>$T_d$</td>
<td>$0.125T_u$</td>
</tr>
</tbody>
</table>

**Table 4.3-2 Frequency Response Method [29]**

Z-N method is used to design parameters of Eq.(4.2-6), the first control plant of the control function where the setpoint input $Y_{sp}(t)$ is signal of the desired travel of the clutch, the feedback signal $Y(t)$ is clutch position measured by position sensor, and the output $U(s)$ is the setpoint of DC motor current. Block diagram of the control loop according to this control plant is shown in Figure 4.3-5.

For step response method, a unit step input to the controlled system is not workable.

Physically, the unit input of one ampere is not enough to lend an impetus to the actuator because of the friction restriction. The same problem is emerged in the simulated model. To
be convenient, frequency method is selected to be used.

By the criteria of frequency response method, proportional gain $K$ is adjusted with other parameters set to $T_i = \infty$ and $T_d = 0$. The adjustment results are shown in Figure 4.3-6, which shows that the controlled system continuously oscillates with a period of 0.0078 second with $K = 7.9$.

![Figure 4.3-6 Frequency Response](image)

**Figure 4.3-6 Frequency Response**

From Figure 4.3-6, $K_u = 7.9$ and $T_u = 0.0078$ is obtained. From Table 4.3-2, parameters within the first control plant based on Eq.(4.2-6) are shown in Table 4.3-3.
<p>| | |</p>
<table>
<thead>
<tr>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>$K$</td>
<td>4.74</td>
</tr>
<tr>
<td>$T_i$</td>
<td>0.0039</td>
</tr>
<tr>
<td>$T_d$</td>
<td>0.000975</td>
</tr>
<tr>
<td>$N$</td>
<td>14</td>
</tr>
<tr>
<td>$b$</td>
<td>1</td>
</tr>
<tr>
<td>$c$</td>
<td>1</td>
</tr>
</tbody>
</table>

**Table 4.3-3 Parameters of First Control Plant**

The control function with parameters designed before is simulated as shown in Figure 4.3-7. It is obvious that overshoot is conspicuous. Such instability may be caused from a too large proportional gain $K$ or a too small derivative gain $T_d$. Such defects will be amended in the next chapter using optimization method.

**Figure 4.3-7 Simulation Result with Control Function**
CHAPTER 5
CONTROL OPTIMIZATION AND SYSTEM SIMULATION

General control turning methods, such as Z-N method, are used for general purpose and perform with general performance. Some range of instability or error is permitted for such turning methods, since they are “general” methods. For specific cases or specific requirements, optimization must be exercised to reach the control demands.

Since the clutch actuator requires rapid motion within short distance, and only small setpoint tracking error is allowed. The control parameters obtained from Z-N turning method is not sufficient for the system requirement, as shown in Figure 4.3-7. Implementation of optimization aimed at the control stability is definitely required, which is executed in this chapter.

Beside optimization of the control parameters, results of the simulation with regard to the optimized systems, including both clutch controller, clutch actuator, and the whole powertrain, are presented in the last section.

5.1 Optimization Algorithm

For the last 50 years, several methods for determining the PID controller parameters have been developed. Some of them deal with some kinds of optimal approach. Actually, the development of PID optimum turning rules has been one of the major areas of research about the PID controller. From the original works of Ziegler and Nichols (Z-N) and Cohen and Coon, a great number of methods have been proposed, some of them giving approach to obtain the PID optimal gains. Macgregor, Wright & Hong (1975) for example, suggested a method to obtain the optimal gains from plots of contour variances under PID feedback control. The resulting controller can be seen as a linear-quadratic-Gaussian (LQG) type. The
PID design based on the Optimal Linear Quadratic theory was also discussed by Argelaget (1995). On the other hand, Vu (1992) gives an iterative algorithm that arises from a Riccati-like equation to obtain the PID parameters that minimize the output variance of the closed-loop system [31]. Also some other general optimum methods are practically used. The method by Haalman is designed for system having dead time; the methods Modulus Optimum (BO) and Symmetrical Optimum (SO) apply to systems without dead time. However, such methods always take controlled system as a simplified process such as second-order system or third-order system.

The target of this study in this chapter is to optimize the PID control parameters with a complete model which much close to the reality. Such propose is expected to have a better result then deal with a simplified model, because the complete model provides more details of system dynamic characters to the optimization. For example, some discrete-like characters, i.e. static-dynamic friction, is rarely be shown in a simplified model.

As the optimization of mechanical parts in CHAPTER 3, the control optimization is implemented in the interface the same with the dynamic model to give more efficient computation.

For the above proposes, the optimum control turning is set to be an optimization problem where the cost function is to minimum the control error between set point and feedback signal, where the dynamic model has been programmed in CHAPTER 2. In the optimization problem, the controlled system is the complete model of clutch/clutch actuator system. And the implementation uses optimization toolbox within Matlab®. The detail definition of the optimization problem is in the next section.

Local minima are the most challenge for optimization. In general, the control cost function may have many local minima. Since no algorithm can guarantee to obtain a global
minimum, efforts should still be exercised to obtain a better solution. Genetic Algorithm (GA) is thought to be a more possible method to find the global minimum in this day. However, since the design variables, the control parameters, have wide ranges for variations in the control function, GA may be very inefficient in computation to be used. To deal with such problem, in this study, better initial variables are tried to obtain to provide an initial point which more closes to the minimum point suitable for the control requirement, and then using SQP method with different minimum step size of direction searching to find the minimum point which feasible for the system requirement and to avoid local minima which may give an unsuitable solution to the optimization. Such initial variables, which may close to the global minimum, are determined by traditional PID turning method, as discussed in section 4.3.

5.2 PID Control Function Optimization

Optimization problem according to the control function is defined in this section.

5.2.1 Cost Function

Since the implementation of optimization is to obtain a set of parameters which control the mechanism to travel with the setpoint as close as possible, the cost function is defined to be the error between setpoint and system output, which is known as the control error.

However, such control error is not only a constant, but also a time depending sequence. Thus, Integrated Absolute Error (IAE), which is always used to judge a control system, is used as the cost function. Where IAE is a constant integrating absolute control error within the time period as defined in Eq.(5.2-1).

\[
IAE = \int_{0}^{t} |y_{sp}(t) - y(t)| dt = \int_{0}^{t} |e(t)| dt
\]  

(5.2-1)
For the setpoint $y_{sp}(t)$, a most common used control signal with an elongated standing time at disengaged position is exercised as shown in Figure 4.3-7. Which is a signal disengaging the clutch in 0.1 second, and then remain disengaged for about 0.8 second for synchronizer to shift gear ratios, finally engages gradually within 0.8 second. The elongated standing time at disengaged position is used to increase the weighting of steady-state error while the clutch is actuated to the fully disengaged position. And the same weighting of time period of 0.8 second is imposed on the final section where the clutch is engaged again to the final steady-state.

![Diagram of setpoint signal](image)

Table 5.2-1 General Setpoint Signal

<table>
<thead>
<tr>
<th>Time (second)</th>
<th>Clutch Travel</th>
</tr>
</thead>
<tbody>
<tr>
<td>0</td>
<td>0.1</td>
</tr>
<tr>
<td>0.8</td>
<td>0.8 (weighed)</td>
</tr>
<tr>
<td>1.6</td>
<td>Engaged</td>
</tr>
<tr>
<td>2.4</td>
<td>Disengaged</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Time (second)</th>
<th>Clutch Travel</th>
</tr>
</thead>
<tbody>
<tr>
<td>2.4</td>
<td>0.8 (weighed)</td>
</tr>
</tbody>
</table>

5.2.2 Design Variables

The design variables are control parameters of the control function defined in section 4.2.

There are two set of PID controls within the control function. The second PID control
plant, which control current of the DC motor and designed by IMC method in subsection 4.3.1, is well performed as shown in Figure 4.3-3. Thus the parameters within this control plant are not modified again in this chapter. The optimization of parameters in this chapter is focus on the first PID control plant, which controls position of the clutch and the first design by Z-N turning method is not well performed as shown in Figure 4.3-7. The control function of this control plant is expressed in Eq.(4.2-6), and the control parameters, the design variables, are shown in Table 5.2-2.

<table>
<thead>
<tr>
<th>Design Variables</th>
<th>$K$</th>
<th>$T_i$</th>
<th>$T_d$</th>
<th>$N$</th>
<th>$b$</th>
<th>$c$</th>
</tr>
</thead>
<tbody>
<tr>
<td>Initial Values</td>
<td>4.74</td>
<td>0.0039</td>
<td>0.000975</td>
<td>14</td>
<td>1</td>
<td>1</td>
</tr>
</tbody>
</table>

Table 5.2-2 Design Variables of Control Optimization

5.2.3 Constraints

There is no absolute constraint for the PID control. However, the parameters still can’t be less then zero. Thus, the main control parameters $K$, $T_i$, and $T_d$ are constrained to be zero to infinity, which is practicable in commercial product.

For parameters $N$, $b$, and $c$ are mainly depends on the controller chosen. Such parameters are set within a controller. Different manufacturers with different specs provide different values of $N$, $b$, and $c$. The optimization on these parameters is to give a guideline for controller choice. In common commercial PID controller, $N$ is in the range from 8 to 20, and $b$ and $c$ are 1 or 0.

The constraints for the design variables are summed up in Table 5.2-3.

<table>
<thead>
<tr>
<th>Design Variables</th>
<th>$K$</th>
<th>$T_i$</th>
<th>$T_d$</th>
<th>$N$</th>
<th>$b$</th>
<th>$c$</th>
</tr>
</thead>
<tbody>
<tr>
<td>Constraints</td>
<td>$0&lt;K&lt;\infty$</td>
<td>$0&lt;T_i&lt;\infty$</td>
<td>$0&lt;T_d&lt;\infty$</td>
<td>$8 \leq N \leq 20$</td>
<td>1 or 0</td>
<td>1 or 0</td>
</tr>
</tbody>
</table>

Table 5.2-3 Constrains of Control Optimization
5.2.4 Optimization Implement

The optimization is implemented in Matlab® using optimization program “fmincon”, which uses SQP method to find minimum point with constraints defined as introduced in subsection 3.3.4.

To avoid local minima, optimizations are executed several times with an increasing minimum change in design variables for finite difference derivatives “DiffMinChange”. The optimization with increasing of minimum change in difference derivative is proceeded until the system output is well fit with the setpoint. The optimizations results with different change steps are shown in Table 5.2-4. And a comparison of the results is shown in Figure 5.2-1 and Figure 5.2-2. Note that each design variables are normalized to be units in the optimizations to avoid modifications with different scales.

<table>
<thead>
<tr>
<th>DiffMin.</th>
<th>K</th>
<th>$T_i$</th>
<th>$T_d$</th>
<th>N</th>
<th>b</th>
<th>c</th>
<th>Cost(1e-3)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1e-4</td>
<td>4.388</td>
<td>0.00815</td>
<td>0.00105</td>
<td>20</td>
<td>1</td>
<td>1</td>
<td>235.64</td>
</tr>
<tr>
<td>1e-3</td>
<td>5.613</td>
<td>0.0185</td>
<td>0.115</td>
<td>10</td>
<td>1</td>
<td>1</td>
<td>137.21</td>
</tr>
<tr>
<td>1e-2</td>
<td>4.613</td>
<td>0.0185</td>
<td>0.117</td>
<td>18</td>
<td>1</td>
<td>1</td>
<td>136.817</td>
</tr>
<tr>
<td>1e-1</td>
<td>4.369</td>
<td>0.0447</td>
<td>0.086</td>
<td>20</td>
<td>1</td>
<td>1</td>
<td>115.51</td>
</tr>
</tbody>
</table>

Table 5.2-4 Optimization Results with Different Derivative Step Size
Figure 5.2-1 Optimization Results with Different DiffMinChange

Figure 5.2-2 Scale View of Figure 5.2-1
From Figure 5.2-2, it is obvious that optimization results with minimum derivative step size of 0.01 and 0.1 provide system outputs that very fit to the setpoint and the transient states are steady with little overshoots. Whatever they are global minimum or not, the solutions are feasible for the control requirement.

The optimization result is shown in Table 5.2-5 and Figure 5.2-3, as a result from minimum derivative step size of 0.1. A comparison of initial design is also shown with a dash line in Figure 5.2-3.

<table>
<thead>
<tr>
<th>Design Variables</th>
<th>Cost</th>
</tr>
</thead>
<tbody>
<tr>
<td>$K$</td>
<td>4.74</td>
</tr>
<tr>
<td>$T_i$</td>
<td>0.0039</td>
</tr>
<tr>
<td>$T_d$</td>
<td>0.000975</td>
</tr>
<tr>
<td>$N$</td>
<td>14</td>
</tr>
<tr>
<td>$b$</td>
<td>1</td>
</tr>
<tr>
<td>$c$</td>
<td>1</td>
</tr>
<tr>
<td>Initial Design</td>
<td>242.7*1e-3</td>
</tr>
<tr>
<td>Optimization Result</td>
<td>115.5*1e-3</td>
</tr>
</tbody>
</table>

Table 5.2-5 Optimization Result

![Figure 5.2-3 Optimization Result](image.png)
Such result corresponds to the forecast in subsection 4.3.2, that the proportional term $K$ is reduced and the derivative term $T_d$ is increased.

Figure 5.2-4 shows a condition where setpoint is modified to an irregular signal. The result shows that the mechanism is still well controlled.

![Figure 5.2-4 Modification with Irregular Setpoint](image)

### 5.3 Optimized System Simulation

The simulation combining the optimized clutch actuator, the optimized control function, and the powertrain are presented in this section.

The following simulation results show a sequence of action. The vehicle starts with 1st gear ratio, and then accelerates with the engaging clutch and the increasing TPS control; the TPS is increasing with time during this time period; at about the third second, where the
engine speed is about 5500 R.P.M., the ECU command decides to shift to 2nd gear ratio, a control command to clutch actuator is sent to disengage the clutch; after fully disengaged, the synchronizer begins to synchronize the next gear; after the synchronizer have finished the shifting process, the ECU gives a command to the clutch actuator to engage the clutch to the original position gradually. In order to unroll the simulation ability, another command of down-shifting is exerted at seventh second. The same disengage/engage command is issued, but instead of up-shifting to 3rd gear ratio, the command is assigned to down-shifting to 1st gear ratio to unfold an action of engine-brake.

Figure 5.3-1 and Figure 5.3-2 show the setpoint signal and the TPS control signal. And Figure 5.3-3 shows the shifting signal.

![Figure 5.3-1 Setpoint Signal of Clutch Position](image-url)
Figures below show the simulation results. Figure 5.3-4 shows the locus of clutch travel, Figure 5.3-5 shows torque generated by the engine, Figure 5.3-6 shows the torque transmit-ability of the clutch, Figure 5.3-7 shows the vehicle speed, Figure 5.3-8 shows the
vehicle acceleration, Figure 5.3-9 shows the engine speed, and Figure 5.3-10 shows the engine acceleration. Note that clutch is fully disengaged with travel distance of 5.9mm in reality, and TPS starts to increase after the clutch is fully disengaged.

![Figure 5.3-4 Clutch Travel](image)

![Figure 5.3-5 Engine Torque](image)
Figure 5.3-5 Engine Torque

Figure 5.3-6 Torque Transmit-Ability of the Clutch

Figure 5.3-7 Vehicle Speed
Figure 5.3-8 Vehicle Acceleration

Figure 5.3-9 Engine Speed
Figure 5.3-10 Engine Acceleration
CHAPTER 6
CONCLUSIONS AND FUTURE WORKS

6.1 Conclusions

The index for Automated Manual Transmission (AMT) is shifting speed and shifting stability, which is dominated by clutch control. The prototype of clutch actuator for AMT by ITRI had been developed for one year. Disengaging speed and control stability are always the difficulties to be overcome because there is no guideline to help modification.

In this study, all these problems are solved. Through the creation of dynamic model and tendency analysis, parameters that dominate the disengaging speed have been found out. Through optimization method, disengaging time of the clutch actuator is reduced from 0.234 second to 0.089 second in the evolution design, which is better than many commercial AMT systems. Through the design of control function and using optimal control method with the combination of control model and dynamic model, the stability of AMT clutch control is improved. By the combined model, a complete clutch control system from controller to clutch travel can be simulated to provide information for control parameters turning. And through optimization method, the control parameters are turned to provide a very stable clutch control. As shown in Table 5.2-5 and Figure 5.2-3, IMC, the index of stability in this study, has been reduced to half of the original design.

Besides the contribution of modifying the clutch control speedier and more stable, the study also builds up a program for simulation and optimization of vehicle transmission system. The program combines dynamic simulation, control simulation, and optimization into a unit. Dynamic characters from engine, through clutch and gear box, to vehicle loading, and the relative control of TPS, clutch, and gear shifting, can be simulated in the program. The shifting process simulation is the emphasis. Besides, through the combined optimization
module, both mechanical parts and control parameters can be optimized according to the design requirements.

Moreover, since the program is created in modules, not only AMT can be simulated. It can be modified to simulate many other transmission systems. Manual Transmission (MT), Double Clutch Transmission (DCT), and hybrid transmission can all be simulated just by substituting or modifying one or two modules.

6.2 Future Works

In the future works of this study, some directions can be focused on as listed below.

First, for practical utility of this study, modification of parameters according to experiments is necessary. Deserving to be mentioned, all the results in this study are not “absolute” results. Most of the data in this study are “theoretical”, and some hypotheses are supposed. Modifications with experiments are thought to be necessary for providing a set of practical results and confirming the hypotheses.

Second, the development of AMT transmission system is prevalent in recent years. There are many novel technologies in such field, like double clutch transmission (DCT) [8], which has been used on Audi TT and BMW M3, and torque tracking control method [9], which has been used on Opel Corsa and Benz Smart. Since the program created in this study is in modules, it is easy to modify it to such new types. Such works must be helpful for the development of AMT system.

Third, not only AMT, hybrid vehicle is famous in recent years. Many studies of hybrid vehicle focus on auto clutch which shifts power between fuel and electrics. Such structure is very close to the auto clutch on AMT of this study. The program produced by this study provides a good base to simulate and optimize such structure. Future study on such topic can
deal with the system just by adding another electrical power source module and redesigned the control strategies.

By the way, the development of AMT is not for a long time, there are many chances for new innovations. And hybrid vehicle is another hot topic that can be followed according to this study. The basic program for simulation and optimization has been created in this study, which must be a practical tool in future studies. According to this foundation, lots works of vehicle transmission problems are waiting for expansions in the future works.
REFERENCES


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