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.

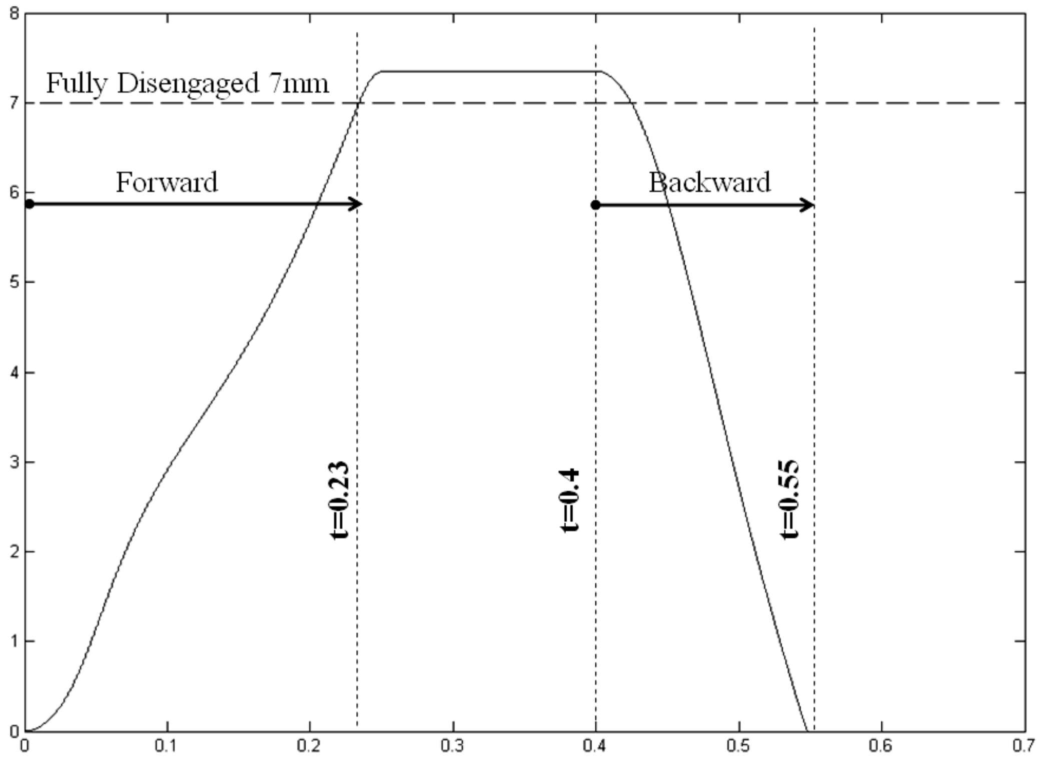

Figure 3.2-1 Clutch Travel (mm) vs. Time (sec.)

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 $W_{Wt} < W_{Wtf}$ as expressed in Eq.(2.2-9) and Eq.(2.2-10). In the simulated model, where μ=0.25, ϕ_n=10.5°, and λ=5°, such self-lock condition exists. The condition is the high frequency vibratile region shown in Figure 2.3-9 caused by numerical calculation.

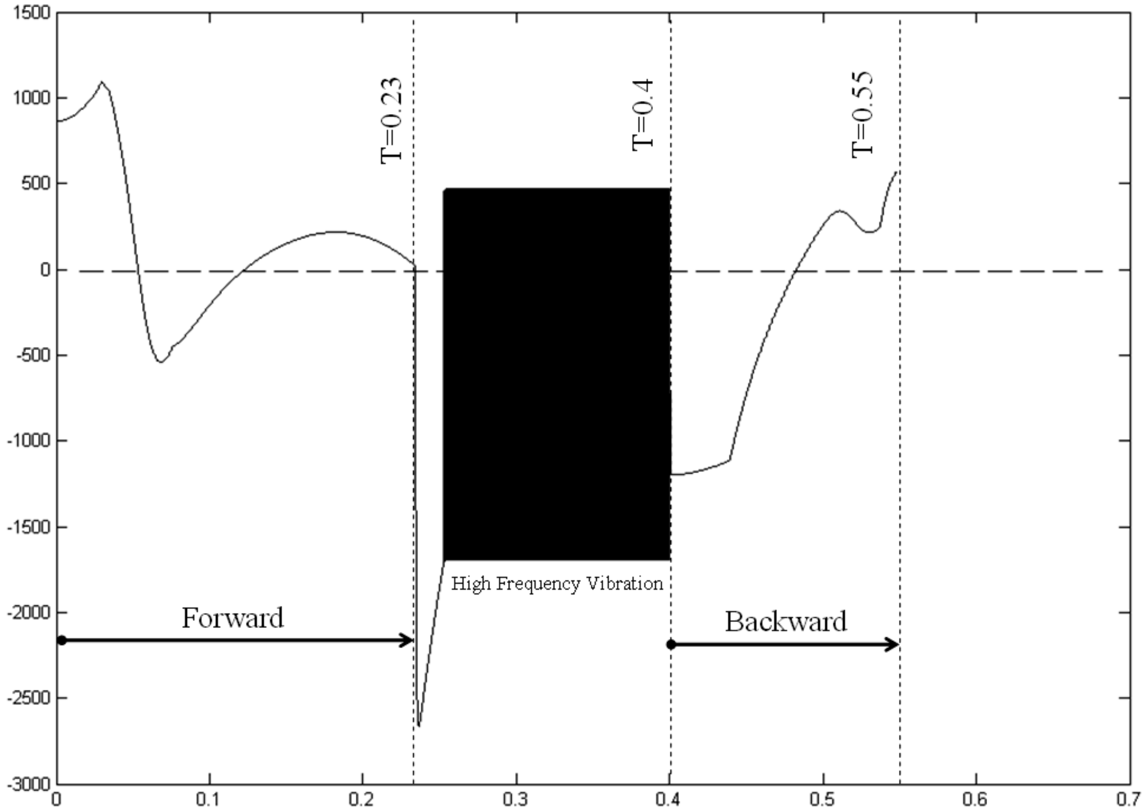

Figure 3.2-2 Clutch Travel Acceleration (mm/sec^2) vs. Time (sec)

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 3.2-3, with a comparison of original result expressed with dash line.

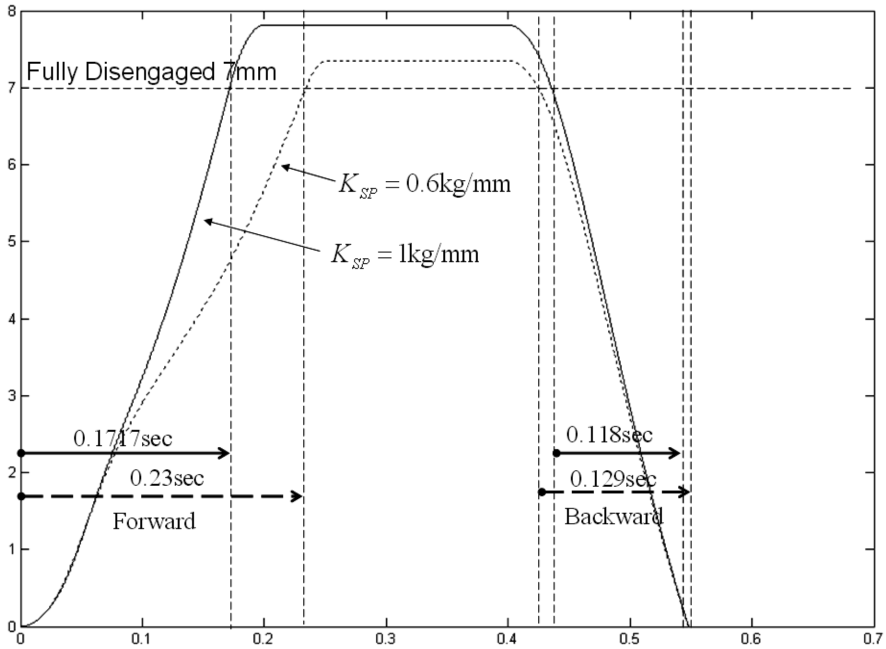

Figure 3.2-3 Comparison of Different Spring Coefficients

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.

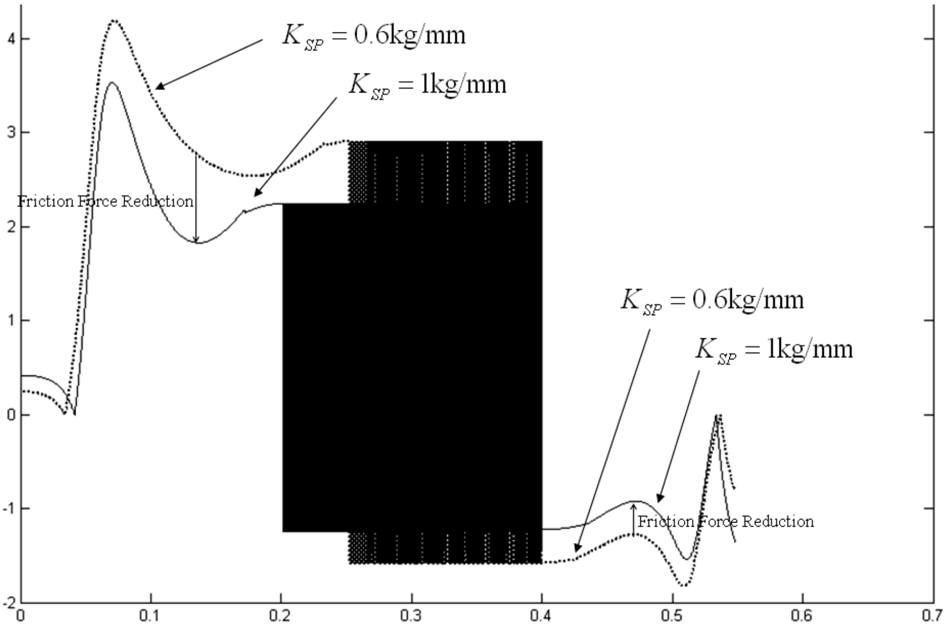

Figure 3.2-4 Friction Force between Worm Shaft and Worm Gear

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 lager 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 λ is increased to 20° which leads to a condition where $W_{Wt} > W_{Wtf}$. 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 λ leads to an increased loading W_{Wt} according to Eq.(2.2-9), thus the clutch actuator can't travel to 7mm without an increased assisting spring.

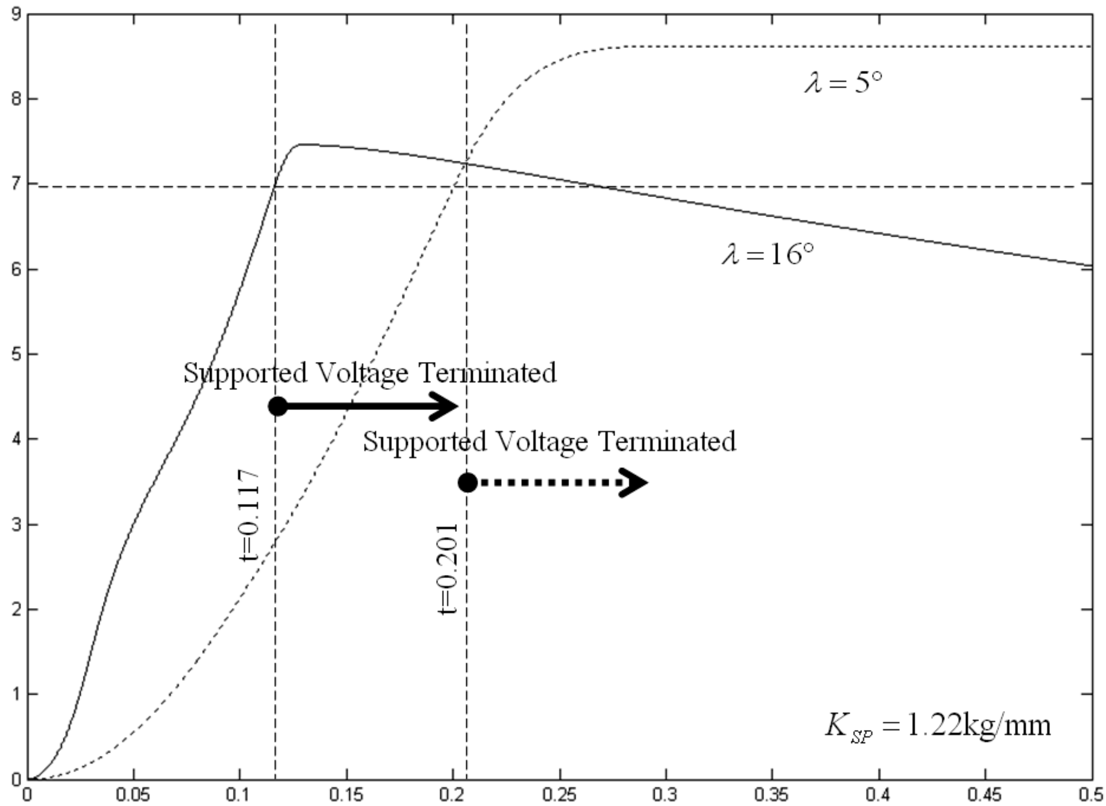

Figure 3.2-5 Clutch Travel (mm) vs. Time (sec.) with Increased Lead Angle

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 λ and the increased spring coefficient K_{SP}.

Since the simulation result of disengaging time is 0.23sec which not conforms to the requirement of 0.1sec, 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.

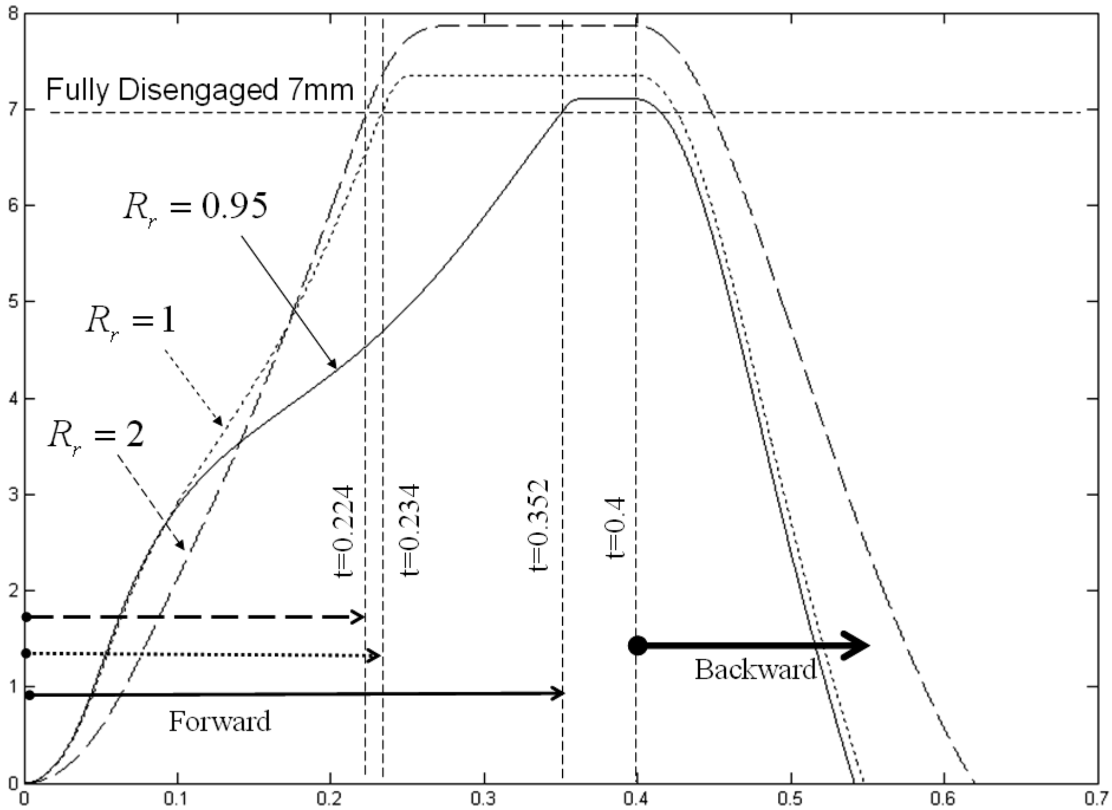

Figure 3.2-6 Comparison with Different Lever Ratio R_r

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.

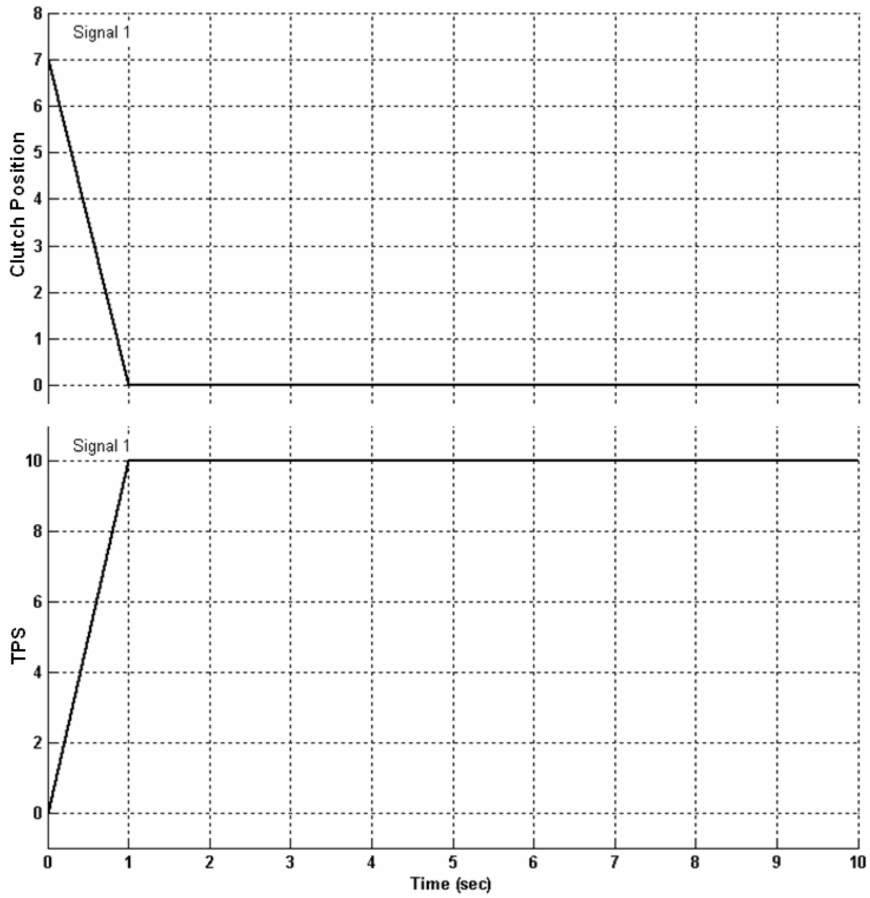

Figure 3.2-7 TPS and Clutch Position while Vehicle Start

The simulation result of vehicle speed is shown in Figure 3.2-8.

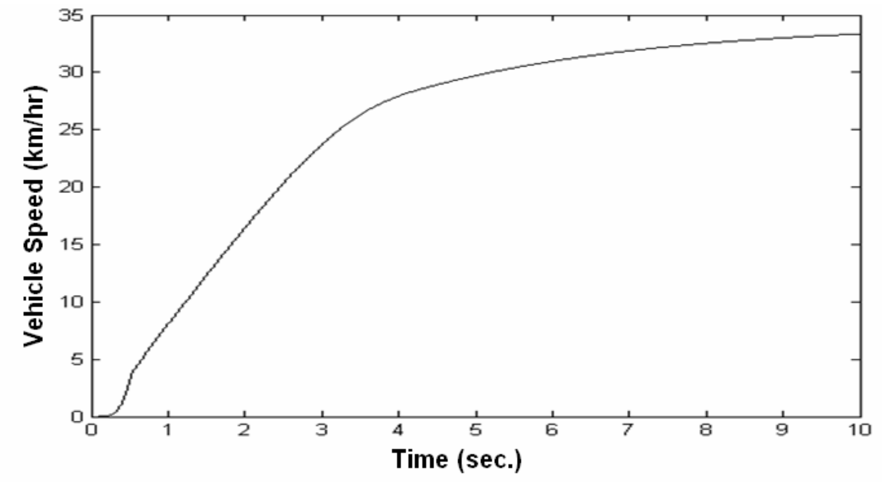

Figure 3.2-8 Vehicle Speed

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 toque 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.

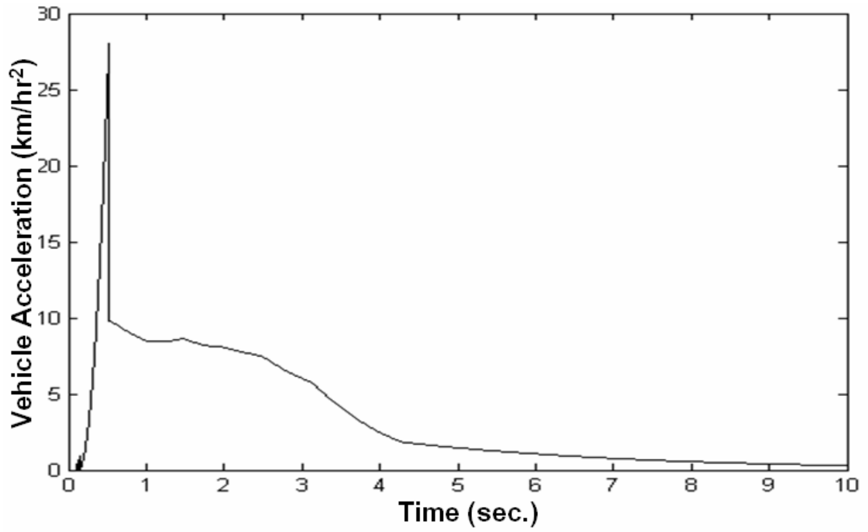

Figure 3.2-9 Vehicle Acceleration

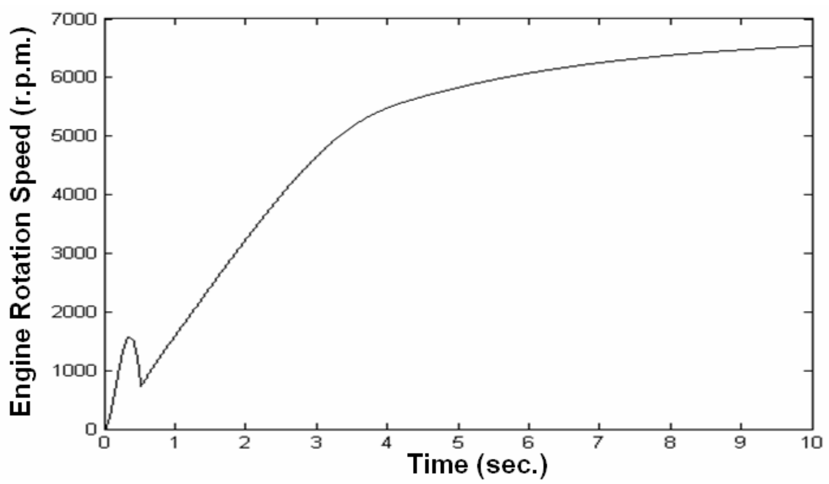

Figure 3.2-10 Engine Speed

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.

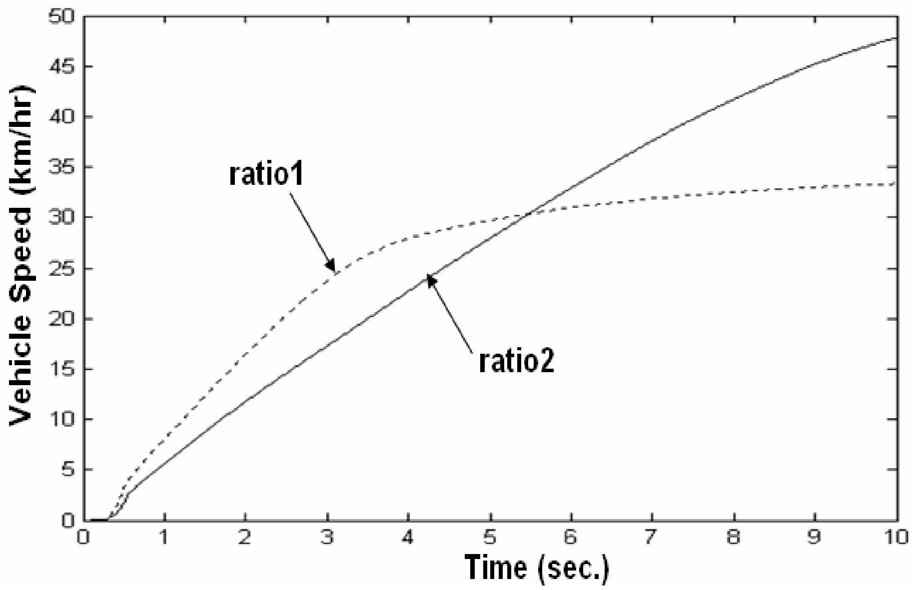

Figure 3.2-11 Vehicle Speed

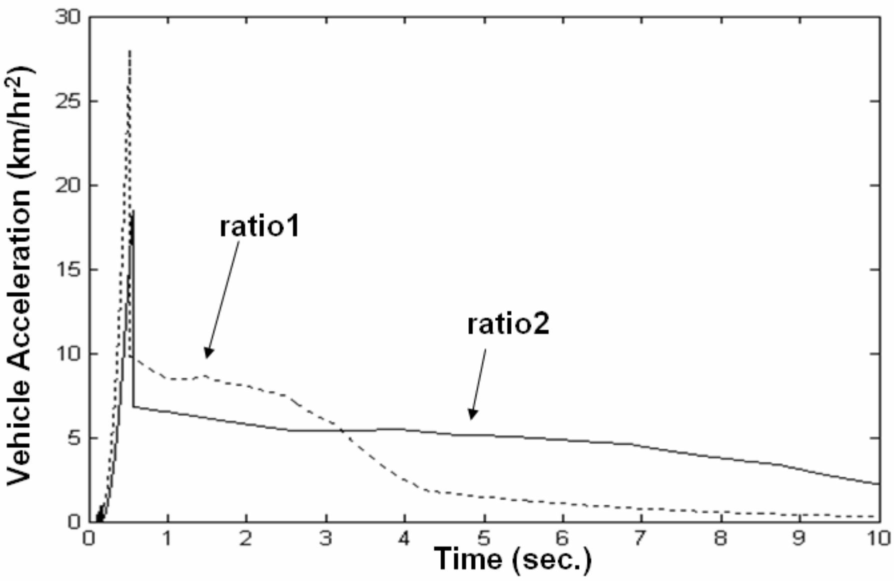

Figure 3.2-12 Vehicle Acceleration

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 lager 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.

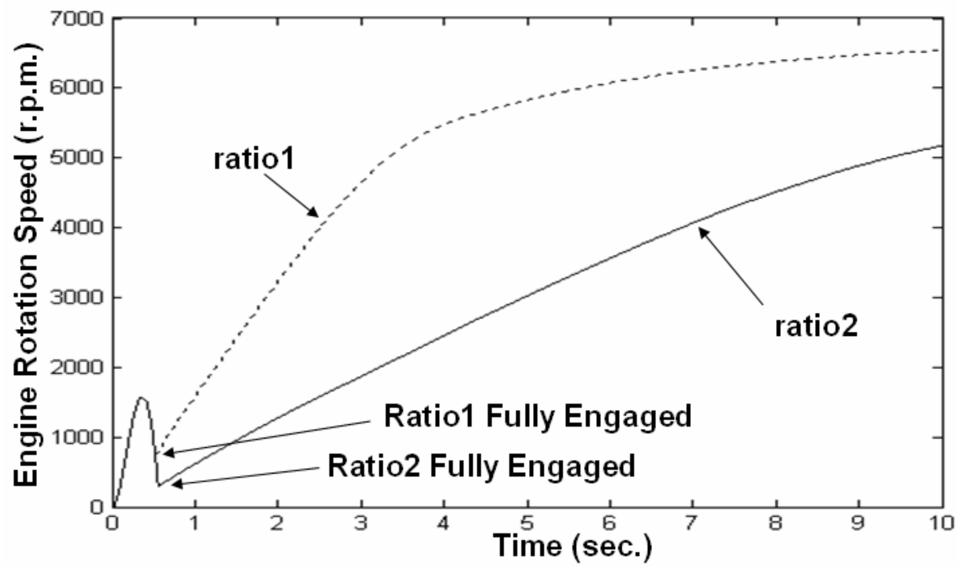

Figure 3.2-13 Engine Speed

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.

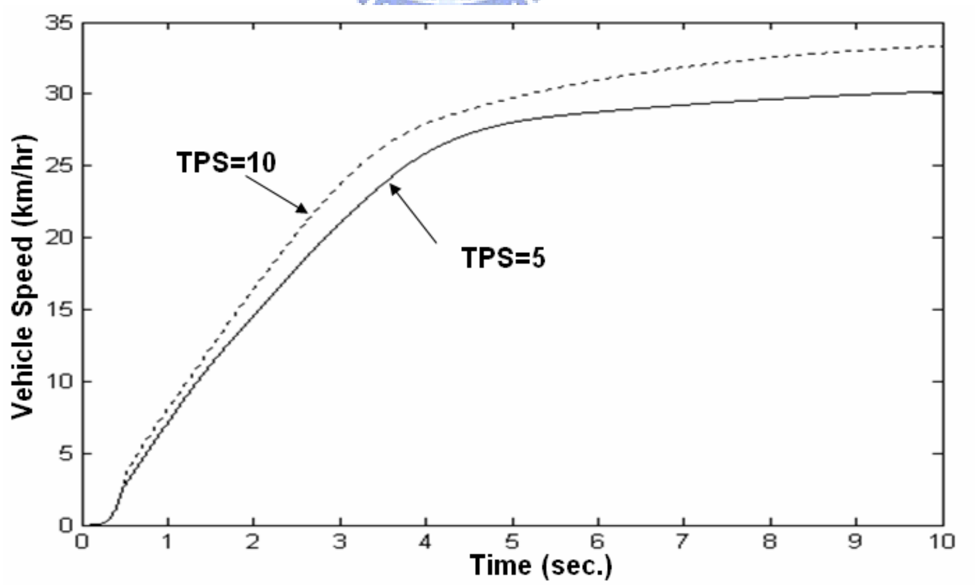

Figure 3.2-14 Vehicle Speed

A clearer phenomenon is displayed on engine speed diagram, as shown in Figure 3.2-15.

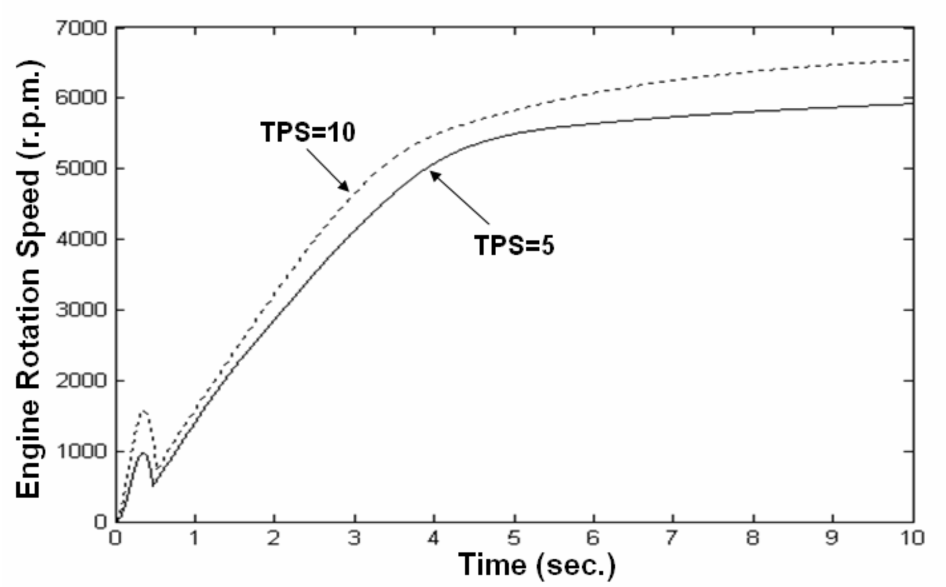

Figure 3.2-15 Engine Speed

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.

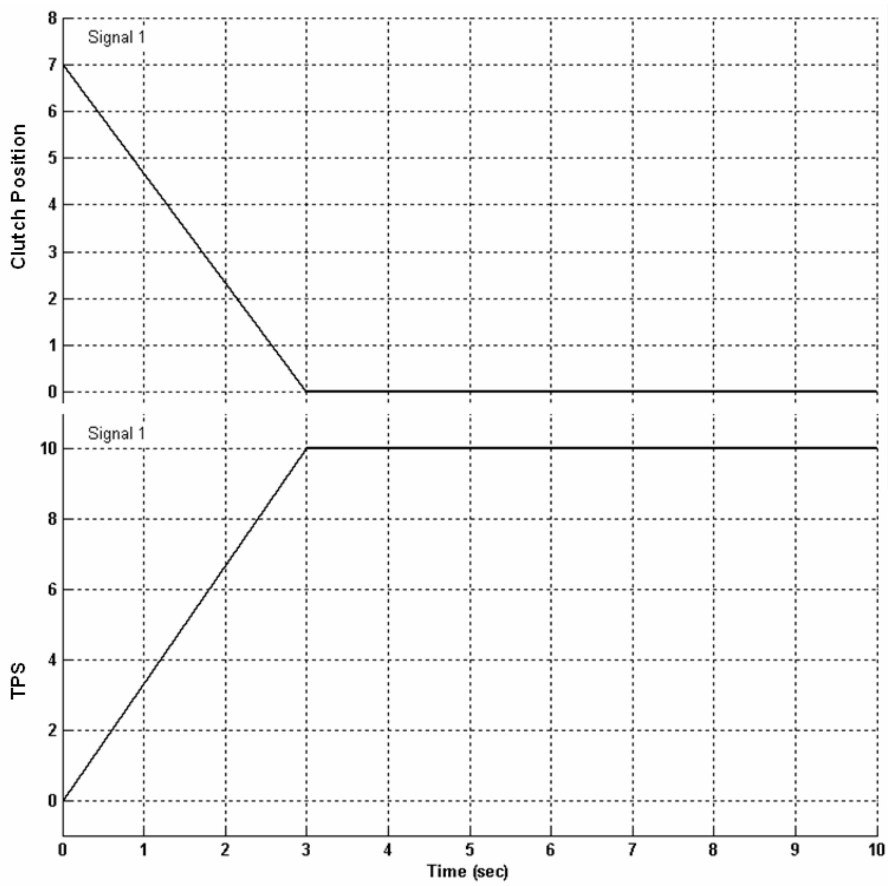

Figure 3.2-16 Clutch Control

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.

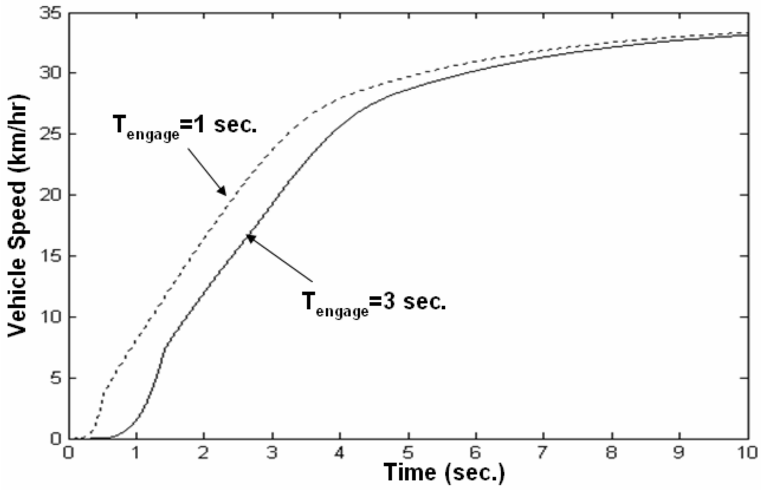

Figure 3.2-17 Vehicle Speed

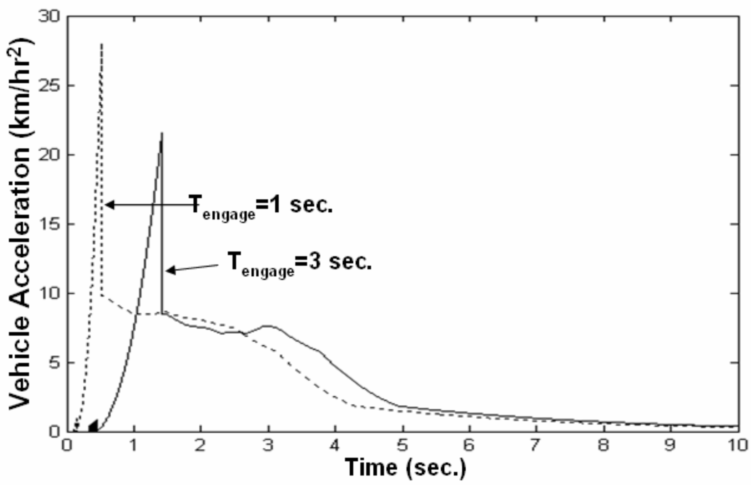

Figure 3.2-18 Vehicle Acceleration

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.

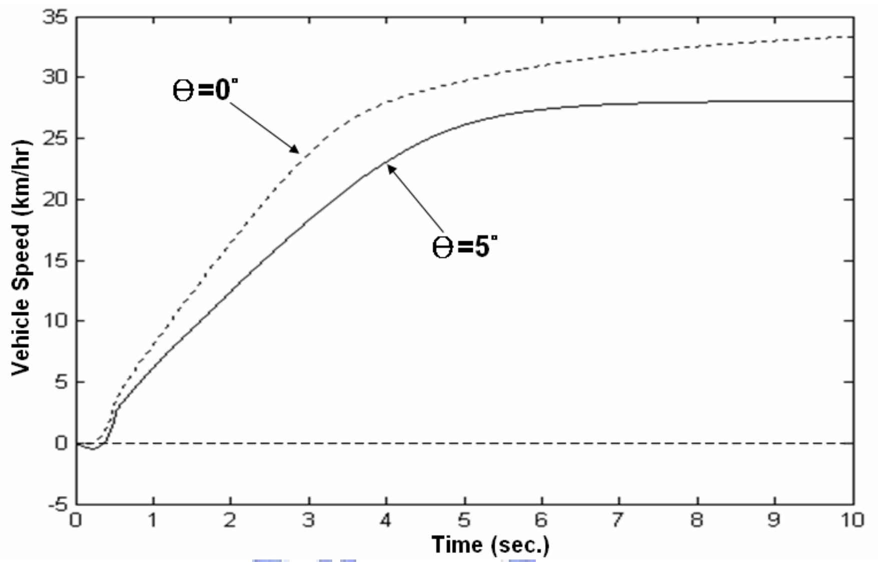

Figure 3.2-19 Vehicle Speed

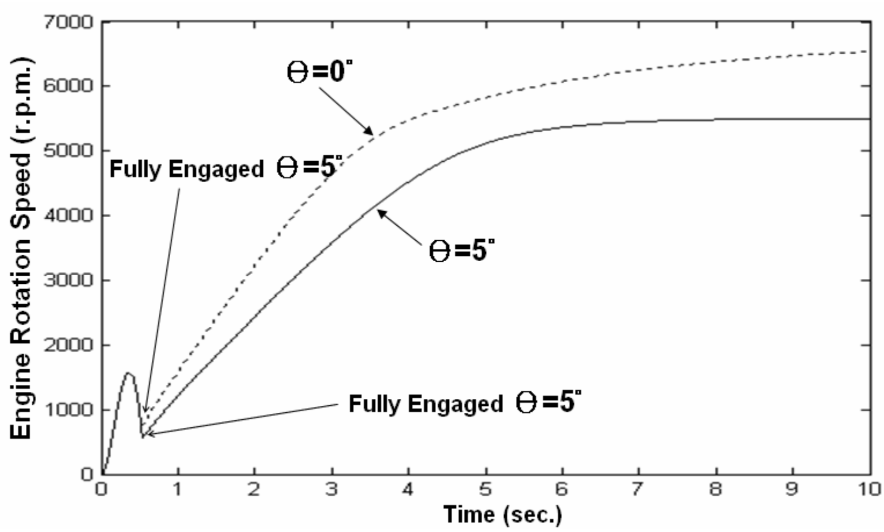

Figure 3.2-20 Engine Speed

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.

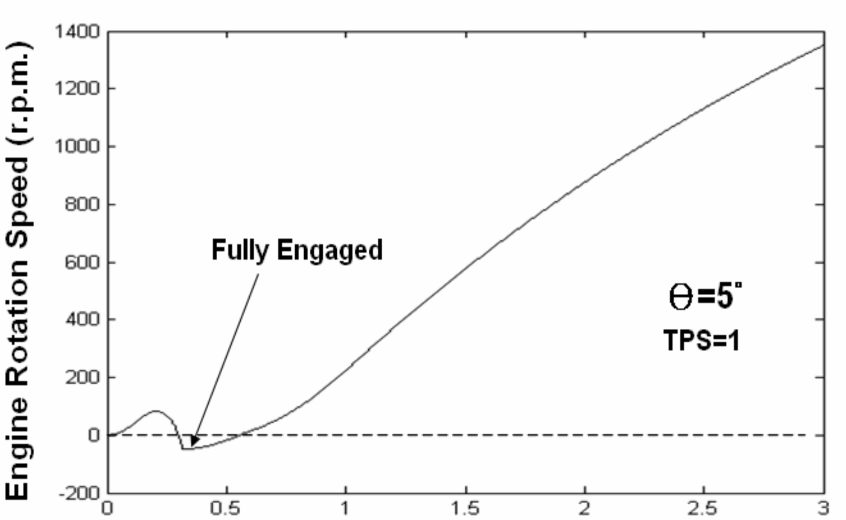

Figure 3.2-21 Engine Speed

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.

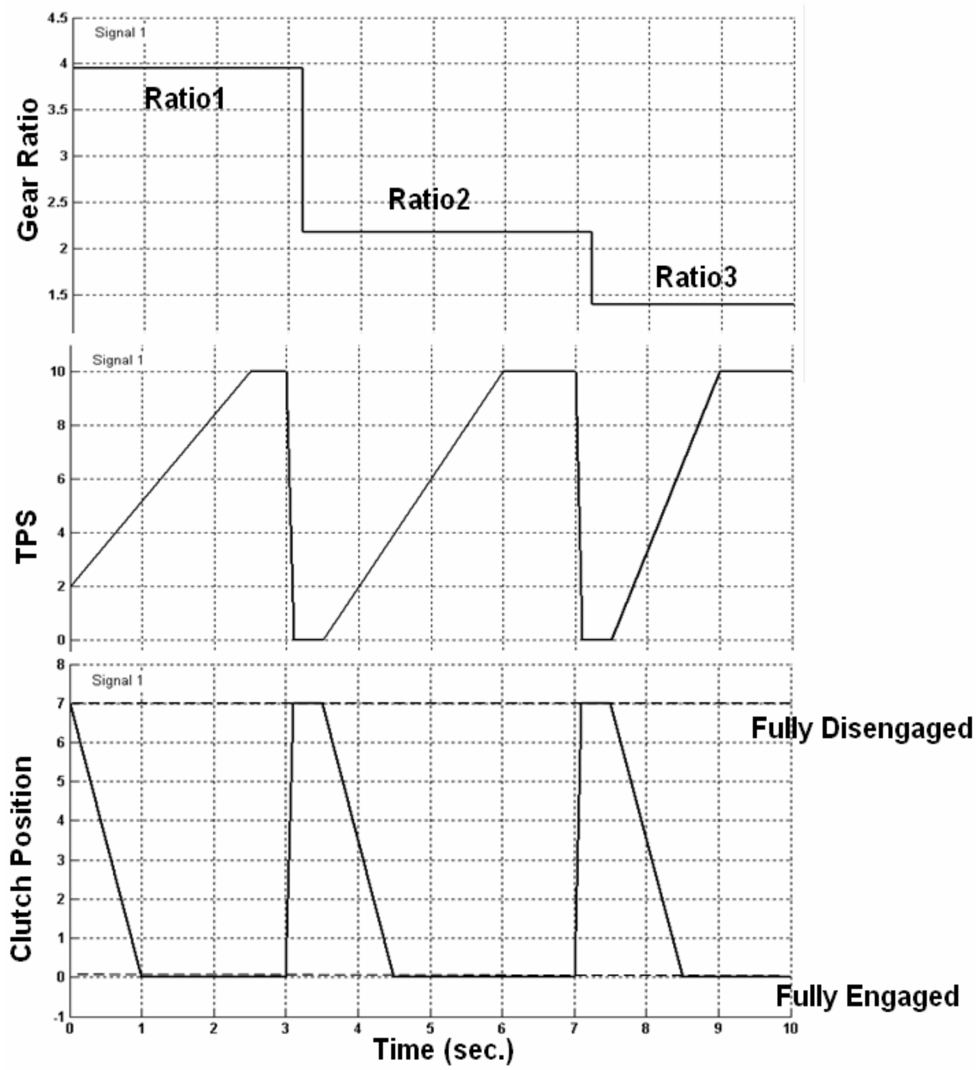

Figure 3.2-22 Control of Shifting Process

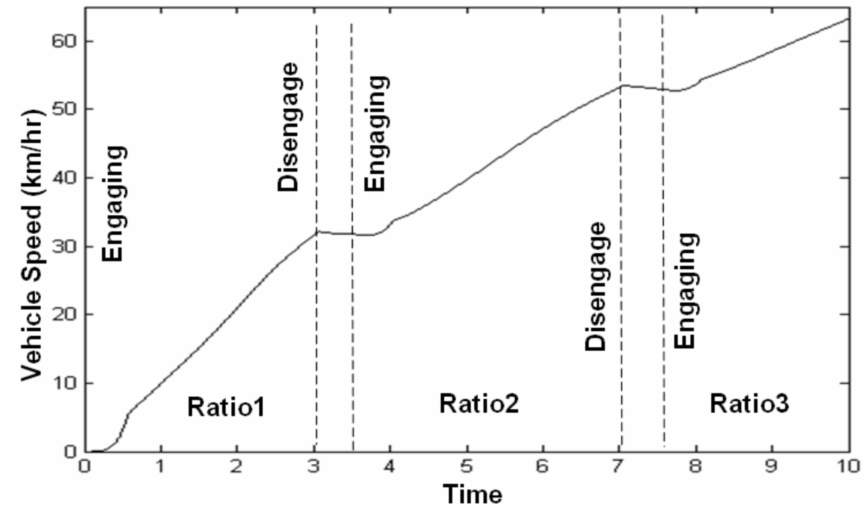

Figure 3.2-23 Vehicle Speed

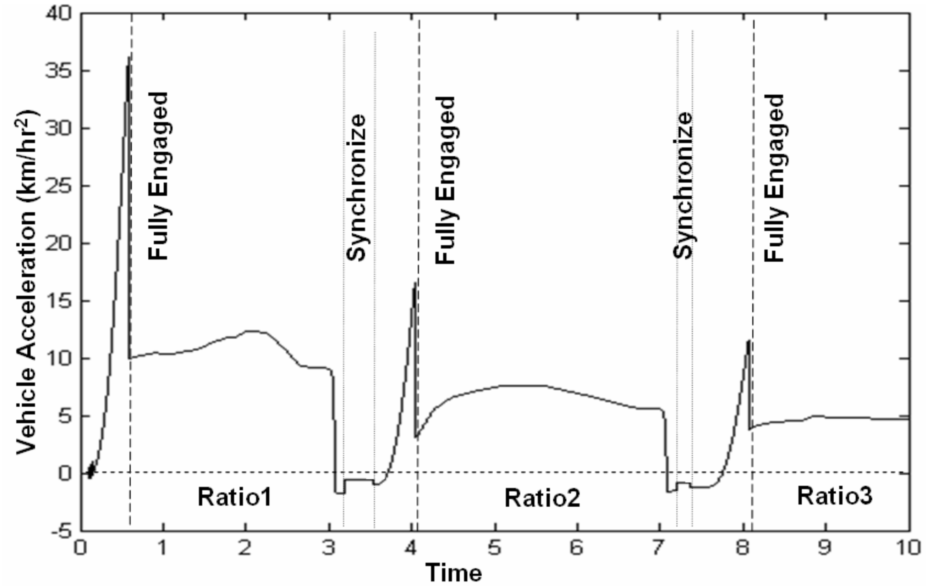

Figure 3.2-24 Vehicle Acceleration

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.

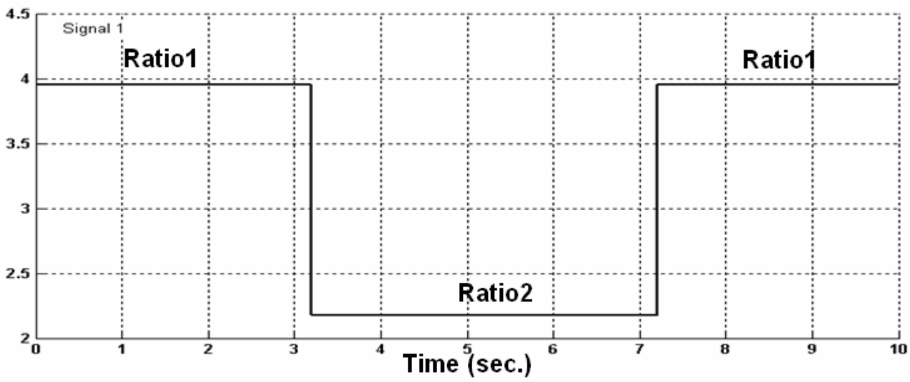

Figure 3.2-25 Gear Ratio

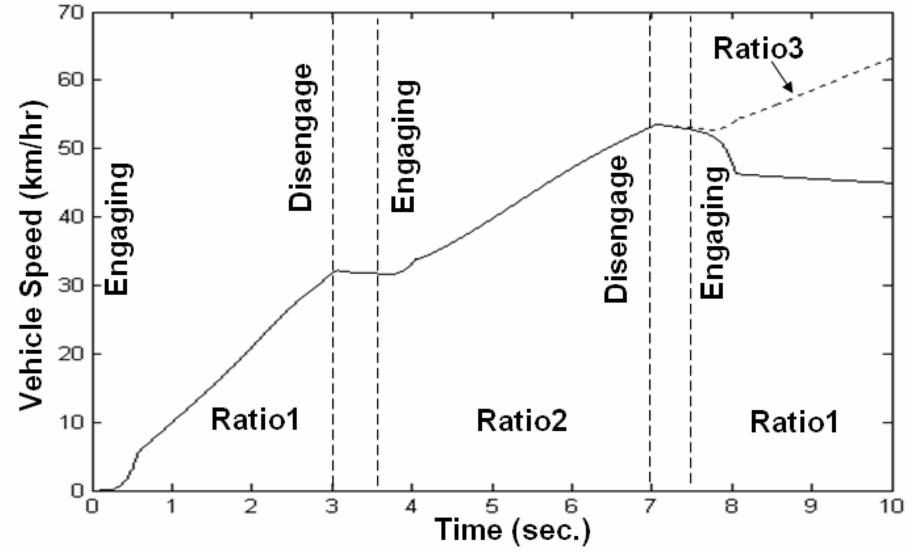

Figure 3.2-26 Vehicle Speed

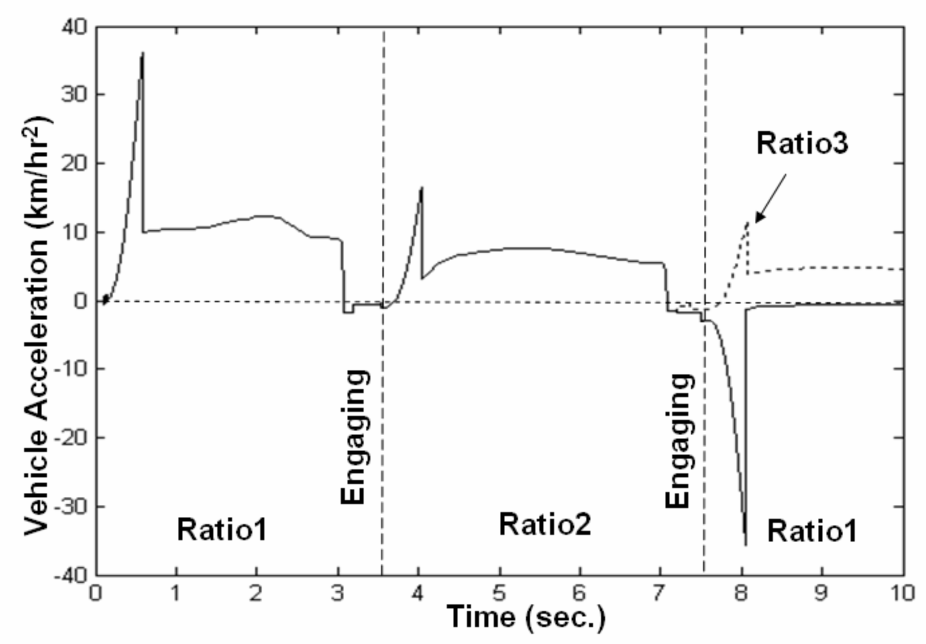

Figure 3.2-27 Vehicle Acceleration

From Figure 3.2-26, it is obvious that when down-shifting, cultch 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.

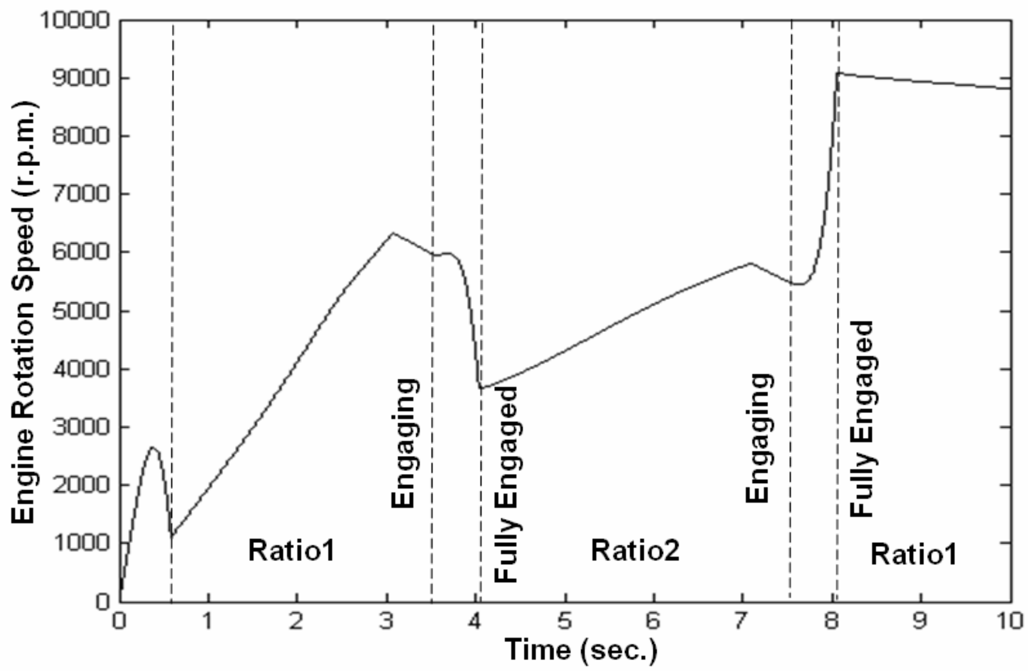

Figure 3.2-28 Engine Speed

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.

Design Variables	Lead Angle	Spring Coefficient	Pre-Deformation	Spring Mass	Lever Ratio
Symbol	λ	K_{SP}	s_{ins}	M_{spring} $(\frac{M_{spring}}{s^0_{ins}} s^n_{ins})$	R_r

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 unproper 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.

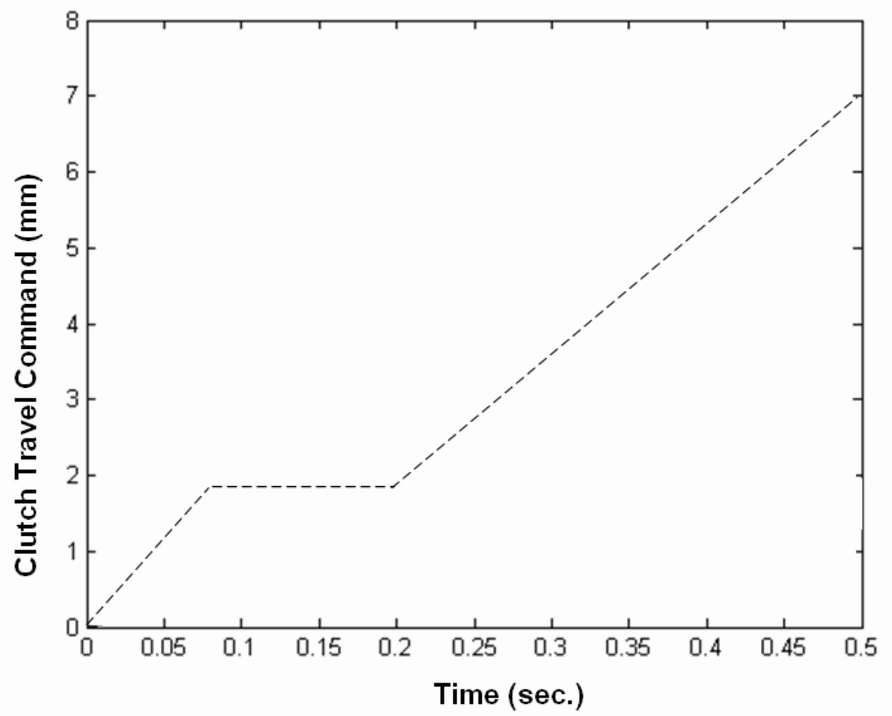

Figure 3.3-1 Clutch Actuator Travel Command

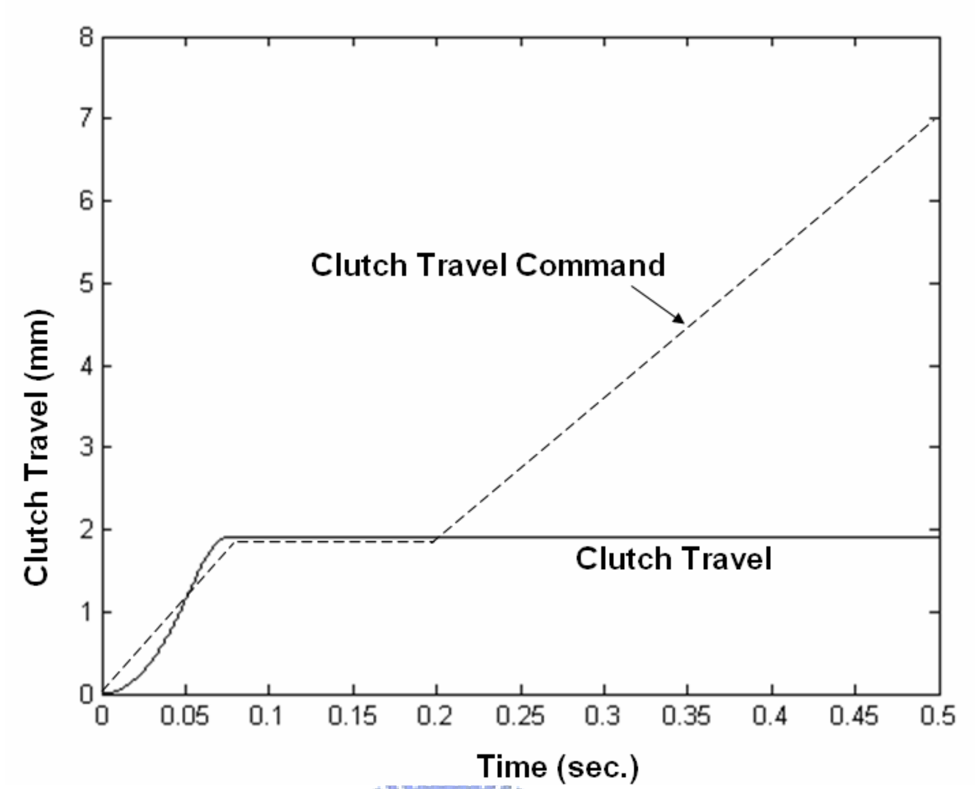

Figure 3.3-2 Clutch Actuator Travel

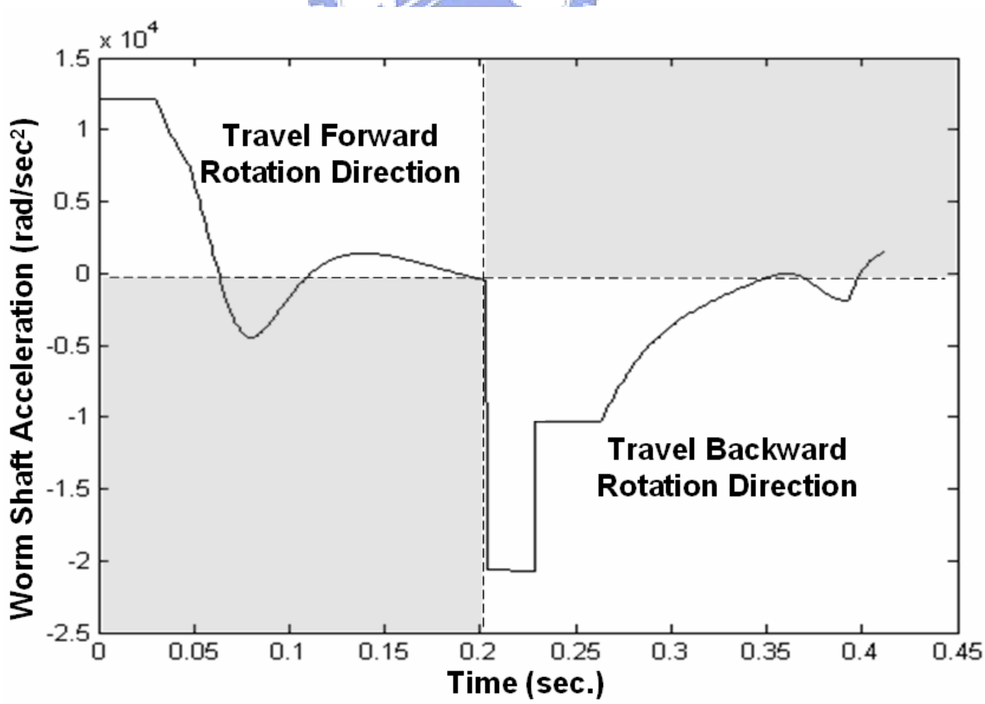

Figure 3.3-3 Worm Shaft Rotation Acceleration

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_{Wt} should be smaller than tangential force caused by friction W_{Wtf}. From Eq.(2.2-9) and Eq.(2.2-10), the following relation of lead angle λ should be constrained.

$$\tan\lambda < \frac{\mu}{\cos\phi_n};$$
$$\lambda < 14.81°$$

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

All the constraints are marshaled in Table 3.3-2.

$T_{engage} \leq 0.3 \quad \sec ond$
$\lambda < 14.81°$
$Tl \times (Tl - WW_t \, dw/2 - W_{Wtf} \, dw/2) > 0$
$s_{ins} \leq 100 \quad mm$
$0.3 \leq R_r \leq 3$

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.

Initial Condition			**Optimization Results**
Design Variables	*Lead Angle* λ	*5°*	*11.14°*
	Spring Coefficient K_{SP}	*0.6 kg/mm*	*0.854 kg/mm*
	Pre-Deformation s_{ins}	*40 mm*	*72.93 mm*
	Lever Ratio R_r	*2*	*1.207*
Cost Function $T_{disengage}$		*0.234 second*	*0.0896 second*

Table 3.3-3 Optimization Results

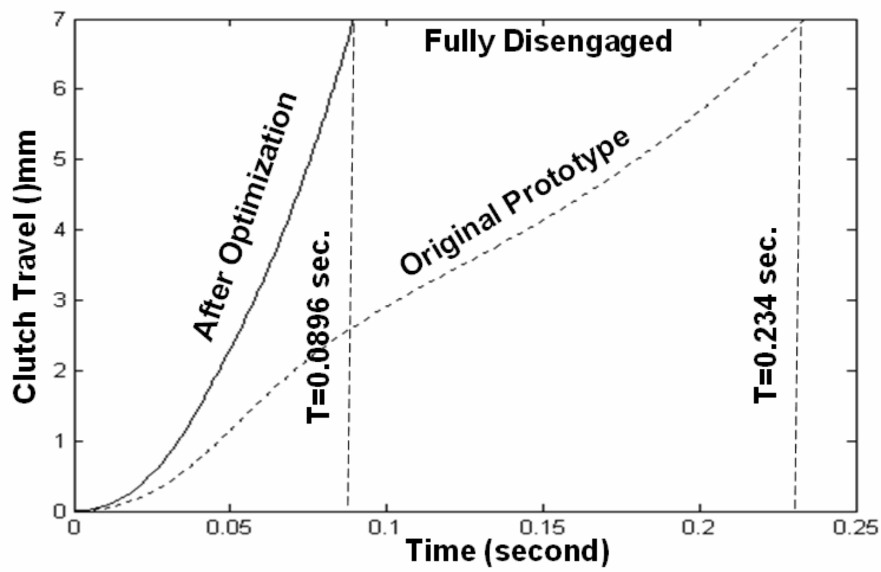

Figure 3.3-4 Optimization Result

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 $T_{disengage} = 0.0798$ 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.