ISTANBUL TECHNICAL UNIVERSITY ★ GRADUATE SCHOOL OF SCIENCE ENGINEERING AND TECHNOLOGY

CONTROLLING OF AUTOMATIC TRANSMISSION GEAR SHIFT BY USING MODEL PREDICTIVE CONTROL

M.Sc. THESIS

Neşe TOPUZ

Department of Control and Automation Engineering

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<u>ISTANBUL TEKNİK ÜNİVERSİTESİ ★ FEN BİLİMLERİ ENSTİTÜSÜ</u>

MODEL ÖNGÖRÜLÜ KONTROL İLE OTOMATİK ŞANZIMANIN VİTES DEĞİŞİMİ KONTROLÜ

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Neşe TOPUZ, a M.Sc student of İTU Graduate School of Science Engineering and Technology student ID 504141124, successfully defended the thesis/dissertation entitled "CONTROLLING OF AUTOMATIC TRANSMISSION GEAR SHIFT BY USING MODEL PREDICTIVE CONTROL", which she prepared after fulfilling the requirements specified in the associated legislations, before the jury whose signatures are below.

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To my family and friends,

FOREWORD

I would like to thank my academic advisor Asst. Prof. Dr. Ali Fuat ERGENÇ, for his interest in my research and transferring his academic knowledge during my dissertation.

I would like also like to thank my supervisor at AVL, Yiğit Çokay, for help and guidance to solve the problems I encountered throughout this study.

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Finally, I would like to offer my thanks to my family and my brother for their moral support during my studies.

September 2017

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ABBREVIATIONS

MIMO	: Multi Input Multi Output
POUS	: Power on Upshift
PODS	: Power on Downshift
PFUS	: Power off Upshift
PFDS	: Power off Downshift
STS	: Speed Torque Sequence
TSS	: Torque Speed Sequence
THO	: Torque Handover
AT	: Automatic Transmission
CVT	: Continuously VariableTransmission
AMT	: Automated Manual Transmission
DCT	: Dual Clutch Transmission
TCU	: Transmission Control Unit
PREP	: Preparation Phase
TORQUE	: Torque Handover Phase
REINST	: Reinstatement Phase
SPEED	: Speed Synchronization Phase
PID	: Proportional Integral Derivative
MPC	: Model Predictive Control
A/F ratio	: Air-Fuel Ratio
HEVs	: Hybrid Electric Vehicles

SYMBOLS

C	: Planet gear carrier
R	: Ring gear
S	: Sun gear
T	: Torque [Nm]
C1	: First Clutch
C2	: Second Clutch
C3	: Third Clutch
C4	: Fourth Clutch
BR1	: First brake
BR2	: Second brake
w	: Rotational speed [rpm]
z_{r1}	: The number of teeth of ring gear of first planetary gear set
z_{s1}	: The number of teeth of sun gear of first planetary gear set
z_{r2}	: The number of teeth of ring gear of second planetary gear set
z_{s2}	: The number of teeth of sun gear of second planetary gear set
z_{r3}	: The number of teeth of ring gear of third planetary gear set
z_{s3}	: The number of teeth of sun gear of third planetary gear set
i_{o1}	: Ratio of first planetary gear set ring gear teeth number to sun gear teeth number
i_{o2}	: Ratio of second planetary gear set ring gear teeth number to sun gear teeth number
i_{o3}	: Ratio of third planetary gear set ring gear teeth number to sun gear teeth number
w1	: Input shaft rotation speed
w2	: Second shaft rotation speed
w3	: Third shaft rotation speed
w4	: Fourth shaft rotation speed
w5	: Fifth shaft rotation speed
w6	: Sixth shaft rotation speed
I_{R3}	: Moment of inertia of RING3 gear
I_{C2}	: Moment of inertia of CARRIER2 gear
I_{R2}	: Moment of inertia of RING2 gear
I_{C3}	: Moment of inertia of CARRIER3 gear
I_{S2}	: Moment of inertia of SUN2 gear
I_{S1}	: Moment of inertia of SUN1 gear
h_1	: Gear ratio for ravigneaux planetary gear (ratio of SUN1 and SUN2 radius)
h_2	: Gear ratio for planetary gear 2(ratio of RING2 and SUN2 radius)
h_3	: Gear ratio for planetary gear 3(ratio of RING3 and SUN3 radius)
w_{R3}	: Speed of RING3 gear
w_{C2}	: Speed of CARRIER2 gear
w_{R2}	: Speed of RING2 gear

$\mathbf{w_{C3}}$	: Speed of CARRIER3 gear
$\mathbf{w_{S1}}$	: Speed of SUN1 gear
$\mathbf{w_{S2}}$	: Speed of SUN 2 gear
$\mathbf{w_{Cl_1}}, \boldsymbol{\tau}_{\mathbf{Cl_1}}$	: Clutch 1 relative speed and friction torque
$\mathbf{w_{Cl_2}}, \boldsymbol{\tau}_{\mathbf{Cl_2}}$	: Clutch 2 relative speed and friction torque
$\mathbf{w_{Cl_3}}, \boldsymbol{\tau}_{\mathbf{Cl_3}}$	: Clutch 3 relative speed and friction torque
$\mathbf{w_{Cl_4}}, \boldsymbol{\tau}_{\mathbf{Cl_4}}$	: Clutch 4 relative speed and friction torque
$\mathbf{w_{Cl_5}}, \boldsymbol{\tau}_{\mathbf{Cl_5}}$	: Clutch 5 relative speed and friction torque
$\mathbf{w_{Cl_6}}, \boldsymbol{\tau}_{\mathbf{Cl_6}}$	: Clutch 6 relative speed and friction torque
$\mathbf{w_0}, \boldsymbol{\tau}_{\mathbf{0}}$	: speed and torque of gear set output shaft
$\mathbf{w_{in}}, \boldsymbol{\tau}_{\mathbf{in}}$	: Turbine speed and torque

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CONTROLLING OF AUTOMATIC TRANSMISSION GEAR SHIFT BY USING MODEL PREDICTIVE CONTROL

SUMMARY

The usage area of model predictive control technology with the developments in industrial process control is expanding day by day. Among the many effective factors for its popularity are allowing the control of the multi input multi output (MIMO) systems or systems that contain non-linear behaviors and dynamic systems to interact with each other. It has a significant advantage to provide the desired solution quickly under various constraints.

In the first section of the thesis, the position of the automatic gearbox in the industry, comparison with the other gearbox types, determination of advantages and disadvantages are included. Then, the usage area of model predictive control in the automotive industry and the control structure that can be used for transmission gear shifting with the aims and contents of the work to be done in this direction are mentioned.

In the second part, the architecture and software model of the transmission that is controlled with a model predictive controller is mentioned. The information about the torque handover control subsystem and its contents are included in the software is given. In addition, transmission software is supported with a simulation in the AVL Cruise™ environment where the the transmission architecture which has a simple planetary gear set and the combined ravigneaux planetary gear set is connected and the powertrain is also included.

The system model for model predictive control is included in the third section. The modeling of all the clutches, brakes and gear sets in the architecture are built by using bond graph methodology is in this section. In the modeling of the clutches has also benefited from different clutch friction models.

In the fourth chapter, general information about model predictive control, basic approaches and applications in the automotive sector are mentioned. After this information on the control design, the controller design and its contents for the torque handover phase are included in this section. The controller is located in speed synchronization phase of the component, which is named torque handover.

MODEL ÖNGÖRÜLÜ KONTROL İLE OTOMATİK ŞANZIMANIN VİTES DEĞİŞİMİ KONTROLÜ

ÖZET

Endüstriyel proses kontrolündeki gelişmelerle birlikte her geçen gün model öngörülü kontrol teknolojisinin kullanım alanı genişlemektedir. Özellikle kimya ve petrokimya endüstrilerinde gelişmiş ve yaygın olarak kullanılması ile birlikte farklı proseslerin yer aldığı farklı endüstri sahalarında da kullanılmaya ve yaygınlaşmaya başlamıştır. Yaygınlaşmasında etkili olan faktörler arasında çoklu giriş çoklu çıkış (MIMO) veya lineer olmayan davranışları barındıran sistemlerin kontrolüne izin vermesi, dinamik sistemlerin birbirleriyle etkileşimli olarak çalışabilmesine olanak sağlaması yer almaktadır. Çeşitli kısıtlamalar altında istenen optimum çözümü hızlı bir şekilde vermesi sağladığı önemli avantajlardandır.

Model öngörülü kontrol teknolojisinin otomotiv sektöründe de yaygınlaşması son yıllarda ivme kazanmıştır. Motorlu taşıtların kullanımının artmasıyla birlikte CO, HC ve NOX içeren egzoz gazı emisyonlarından kaynaklanan hava kirliliği gün geçtikçe artmaktadır. Dünya genelinde emisyon standartlarına uygun araçların üretilmek istenmesi, taşıttan çıkan zararlı gazların miktarının ayarlanması ve en aza indirilmesi gerekliliğini ortaya çıkarmıştır. Ayıca motorlu taşıtlarda birim hacimden elde edilmek istenen gücün arttılmaya çalışılması aracın ve motorun modellenmesi gerekliliği gibi durumları beraberinde getirmiştir. Tüm bu ihtiyaçlar doğrultusunda model öngörülü kontrol teknolojisi farklı tipteki problemlerle, farklı çözüm uygulamalarıyla kendine yer edinmiştir ve uygulanma sıklığı giderek artmıştır. Diğer sektörlerde olduğu gibi otomotiv sektöründe de elektronik kontrol uygulamalarının artması sistemlerdeki çeşitli sorunlar için optimum çözümü elde etme avantajını beraberinde getirmiştir. Model öngörülü kontrol uygulaması da bu avantajları sağlamak açısından gün geçtikçe gelişmektedir. Doğan ihtiyaçlar karşısında model öngörülü kontrol uygulaması ihtiyaca göre geliştirilmeye çalışılmaktadır. Bu doğrultuda özellikle nonlineer sistemleri içerisinde barındıran veya farklı dinamik sistemlerin beraber çalıştığı uygulamalarda kontrol uygulamasının geliştirilmesi için uzun bir süredir araştırma ve geliştirme üzerine çalışılmaktadır.

Tezin ilk bölümünde otomatik şanzımanın endüstri içerisindeki yeri, diğer şanzıman çeşitleri ile kıyaslanması, avantaj ve dezavantajlarının belirlenmesi yer almaktadır. Ardından model öngörülü kontrolün otomotiv sektöründeki kullanım alanları, şanzıman vites değişimi için kullanılabilecek kontrol yapısı ve bu doğrultuda yapılacak olan çalışmanın amaçları ve içeriğinden bahsedilmektedir.

İkinci bölümde model öngörülü kontrol tasarımı yapılacak olan şanzımanın mimarisi ve yazılım modelinden bahsedilmektedir. Yazılım içerisinde yer alan tork dönüştürücü kontrolü alt sistemi ve içeriği hakkında bilgi verilmektedir. Ayrıca basit güneş dişlisi grubunun ve birleşik ravigneaux güneş dişlisi grubunun bulunduğu şanzıman mimarisinin bağlı olduğu ve güç aktarım organlarının da yer aldığı AVL Cruise™ ortamında yapılan bir simülasyonla da şanzıman yazılımı desteklenmektedir. Yapılan bu simülasyon gerçek hayattaki aracın sahip olduğu birçok özelliği içerisinde

barındırabilmektedir. Örneğin model içerisinde şaftların eylemesizlik momentleri, şanzıman mimarisinde yer alan kavrama ve frenler için sürtünme kayıpları, araç motoruna ait parametreler, tork dönüştürücü, tekerlekler ve diferansiyel gibi birçok güç aktarım organının özellikleri yer almaktadır.

Model öngörülü kontrol için gerekli sistem modellemesi üçüncü bölümde yer almaktadır. Kavrama, fren ve dişli setlerinin yer aldığı mimarinin bond graph methodolojisi kullanılarak oluşturulmuş modeli bu bölüm içerisindedir. Bond graph methodolojisi sistemin sahip olduğu türevsel ve integral nedensellikleri ortaya çıkarmış ve tüm şanzıman modelinin 3. dereceden bir sistem modeli olduğunu ortaya koymuştur. Sistemin modellenmesi esnasında elde edilen tüm denklemler de bu kısımda yer almaktadır. Kavramaların modellenmesi esnasında da bu zamana kadar yapılmış farklı kavrama sürtünme modellemeleri dikkate alınmıştır. Literatür araştırması bond graph methodolojisi, farklı kavrama sürtünme modellerinin araştırılması ve model öngörülü kontrolün gerekliliklerinin, sınırlarının, hangi sistemlere ne şekilde uygulanacağının örneklerden yola çıkarak anlaşılması şeklinde yapılmıştır.

Dördüncü bölümde model öngörülü kontrol ile ilgili genel bilgiler, temel yaklaşımlar ve otomotiv sektöründeki uygulama alanlarından bahsedilmektedir. Kontrol tasarımına ait bu bilgilerden sonra tork dönüştürücü faz için oluşturulan kontrolör tasarımı ve içeriği bu bölümde yer almaktadır. Kontrolörün şanzıman yazılımı içerisinde tork aktarımının hız senkronizasyonu fazını gerçekleyen bileşen içerisinde yer aldığı ve bu sistemdeki kontrol uygulamasında kısıtlayıcı parametrenin kavramaların aktarabileceği maksimum tork değeri olduğu bilgisi yer almaktadır. Ayrıca şanzıman ve araç modelinin birlikte yer aldığı yazılım içerisinde olmasından dolayı örnekleme zamanı seçiminde de bu yazılıma bağlı kalınmıştır. Gerçek şanzıman üzerinde ölçülebilen ve ölçülemeyen parametreler de kontrolör içerisinde yer almaktadır. Kontrolör tasarımı için hedeflenen maliyet fonksiyonu, hız ve tork değişimlerinin yumuşak, hızlı ve sürücü tarafından hissedilmeyecek şekilde olmasını içermektedir. Bu doğrultuda parametre ağırlıkları değiştirilerek farklı sonuçlar elde edilmişir.

Model öngörülü kontrol, bond graph methodolojisi kullanılarak elde edilmiş denklemleri kullanarak, sistemin gelecekteki cevaplarını öngörerek sisteme kontrol girişi verir. Sisteme gönderdiği kontrol çıktısı, sistem için istenilen iyileştirmeleri içeren optimum çözümü sağlayacak şekildedir. Sistem için istenilen iyileştirmeler; vites geçişleri esnasında devreye giren ve devreden çıkacak kavramalardaki tork aktarımlarının sürücü tarafından olabildiğince az hissedilmesi, dolaylı yoldan kavramalarda tork geçişleri esnasındaki hız senkronizasyonun hızlı ve yumuşak bir şekilde gerçekleşmesidir. Uygulanan kontrol algoritması kavramaların taşıyabileceği maksimum tork değerleri kısıtları içerisinde yukarıda içeriği belirtilen maliyet fonksiyonunu sağlayacak şekilde oluşturulmuştur.

Son bölümde, bu çalışmadan yola çıkarak model öngörülü kontrol uygulamasının sadece üçüncü vitesten dördüncü vitese geçiş esnasındaki hız senkronizasyonu fazına etki etmesinin yanı sıra, tork aktarımı esnasında diğer fazları da kapsayacak şekilde uygulanabilmesinin gelecekte yapılabilecek iyileştirmeler arasında yer almasından bahsedilmektedir. Şanzımanın bond graph methodolojisi kullanılarak elde edilen model denklemleri hem vites büyültmek hem vistes küçültmek için oluşturulabilecek kontrol algoritması için kullanılabilmektedir. Bu çalışmada vites büyültme esnasında kullanılan algoritma dördüncü vitesten üçüncü vitese geçme gibi vites küçültme durumlarında da kullanılabilir. Ayrıca bu modelleme yöntemi ile farklı şanzıman tipleri, motor, güç aktarım mekanizması, süspansiyon sistemi gibi çeşitli dinamik sistemler de modellenebilmektedir. Araç içerisinde optimum çözüme ulaşılmak istenen herhangi bir problem karşısında sistem modelinden yararlanarak model öngörülü kontrol algoritması oluşturulabilir. Bu çalışmadan yola çıkarak gelecekte otomotiv sektörünün farklı uğraş sahalarında yapılabilecek iyileştirmelerden de bahsedilmiştir.

1. INTRODUCTION

1.1 Introduction and Aim of Study

The aim of this Master's thesis is to improve the torque tracking without slipping in the clutches and responding driver torque request as fast as possible during gear shift. Depending upon the gear shift request from driver, deciding which clutches to engage or disengage, calculating how much torque will be transferred through clutches, thus the hydraulic oil pressure are regulated via driving control. The main purpose here is to ensure that the clutches can follow the calculated speed profile quickly and smoothly while gear shift is occuring.

Automatic transmissions have been produced and developed to meet the needs of the industry for over fifty years. While all kinds of transmissions carry out specific functions, the different types of transmission with different components have some advantages with respect to the others. For example, gear shift is executed with automatic control systems at the automatic transmissions with the torque converter or fluid coupling included in drivetrain whereas the gear shift is carried out by the driver in manual transmissions. However, both types of gearboxes deliver power to the wheels from engine by increasing torque and reducing speed corresponding to its gear ratios. The losses caused by slipping have been prevented with the control of fluid coupling in automatic control systems. The other difference is the usage of planetary gear sets in automatic transmissions. With different combinations of planetary gear sets, different vehicle speed values can be obtained. In each combination, due to engaged clutches no slipping occurs. Thus, there is no loss of speed and torque.

In the automotive industry, model predictive control algorithm is applied for various problems in order to meet different needs. For example, explicit and hybrid model predictive control is applied in various applications such as traction control, active steering, and semiactive suspensions in vehicle dynamics. In addition, MPC is also developed for direct injection engine control, A/F ratio control, magnetic actuators, and cabin heat control in HEVs applications [1].

1.2 Literature Review

In this study, there are two important issues to be implemented title apart from the software of the transmission and architecture of transmission to be controlled. One of them is the bond graphic method, which is the method of modeling of the transmission, and the other is the model predictive controller design, which will be applied in gear shift.

Before applying model predictive control in 7-speed automatic transmission, modeling of the existing automatic transmission will be achieved using the bond graph modeling method. The modeling of the transmission, which is a high order system, due to many reasons such as the fact that the power flow from the input shaft to the output shaft of the transmission can be clearly shown during modeling, the relationship between torque and speed can be easily understood, mathematical equations can be obtained systematically, parameters that can be written in terms of each other can be easily reduced, is made by the bond graph modelling. In the third section, the definition of the bond graph model, the acquisition of the torque and angular velocity equations on the simple planetary gear set and the combined ravigneaux planetary gear set, and the modeling of the gear sets are presented.

In the fourth section, the model predictive control approach, the terms that are used, the factors to be considered such as weight parameter and cost function in the controller design are mentioned.

When the graphs of the torque handover phase are examined, it is decided that a different control type should be applied in order to eliminate irregularities in the system where the PID controller is located in. This controller is expected to respond to different requirements and requests of the system. MPC controller gives different weight values to different parameters of the system by taking into account the cost function. In this study, during the torque handover phase of the transmission, parameters were determined with regards to the torque values transmitted by the clutches and the speeds that these clutches have in the speed synchronization phase and these data were used in the cost function.

2. TRANSMISSION STRATEGY

2.1 Transmission Architecture

The gearbox consists of a combined ravigneaux planetary gear set and a planetary gear set with clutches and brakes that connects various components to each other and the chassis respectively. The seven-speed (7 forward and reverse) transmission has 4 clutches and 2 brakes located in gearbox.

The output shaft of the torque converter is connected to the ring gear of the third planetary gear set and the compound ravigneaux planetary gear set which is consists of first and second planetary gear set via first clutch.

The output of the first clutch is connected to the planetary gear of first/second planetary gear set. This planetary gear can be connected to the chassis by controlling the second brake.

The sun gear of third planetary gear set is always attached to the chassis. As a result, the input torque from the torque converter is transmitted through the ring gear to the carrier at a constant gear ratio. The output torque is transmitted to the system via the carrier. This planetary gear is connected to the sun gear in the first planetary gear set with the second clutch and to the sun gear in the second planetary gear set with the third clutch. This sun gear of the second planetary gear set can be connected to the chassis by controlling the first brake.

The planetary gear of the first planetary gear set is connected to planetary gear of the second planetary gear set directly. At the same time, first and second planetary gear sets a common ring gear. The ring and planetary gear of the second planetary gear set is connected via the fourth clutch.

The output torque of gearbox is taken from the ring gear of first/second planetary gear set. After the multiplication of torque and differential ratio, torque is transmitted to the wheels.

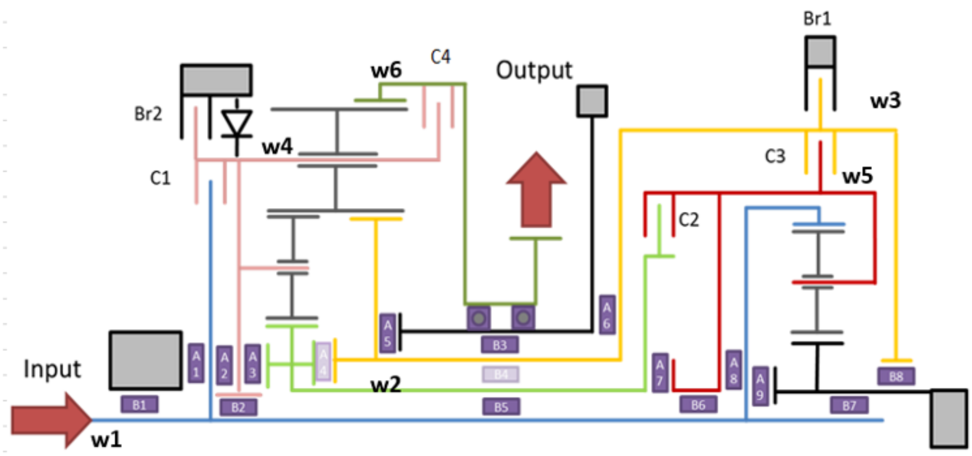

Figure 2.1 : Schematic representation of transmission architecture.

Table 2.1 : Combinations of clutches and brakes.

GEAR	C1	C2	C3	C4	BR1	BR2
Gear 1		X				X
Gear 2		X			X	
Gear 3		X	X			
Gear 4	X	X				
Gear 5	X			X		
Gear 6	X		X			
Gear 7	X				X	
Gear R			X			X

Table 2.2 : Gear ratios.

GEAR	GEAR RATIO
1^{st}	4.67
2^{nd}	2.54
3^{rd}	1.71
4^{th}	1.18
5^{th}	1
6^{th}	0.8
7^{th}	0.72
Reverse	3.78

2.2 Software Architecture

The level of transmission strategy where all clutches are controlled included in the software architecture of the system includes components of different subsystems together with torque handover. All of the continuous, oncoming and offgoing clutches and lockup clutch are controlled via signals from each subsystems in transmission strategy.

The behavior of the clutches and the torque transfer are determined in transmission strategy block according to the signals from the blocks where the input signals are processed and the gear selection is made. Transmission control of the vehicle in the case of driving is carried out at this stage. The transmission strategy block determines which the clutch is to be engaged, the clutch is to be disengaged and the the clutch is continued according to the condition of the vehicle, actual gear, requested gear by driver.

2.3 Torque Handover Component

It is assumed that the torque demand from the driver and the engine torque do not change during the shift. In this direction, four different shift types can occur in the direction of the power flow and direction of gear shift.

- Power On Upshift (POUS)
- Power On Downshift (PODS)
- Power Off Upshift (PFUS)
- Power Off Downshift (PFDS)

All of the shift types consist of 4 main phases: preparation, torque handover, speed synchronization and reinstatement phase. On the shift diagrams x-axis represents time and y-axis represents speed or torque.

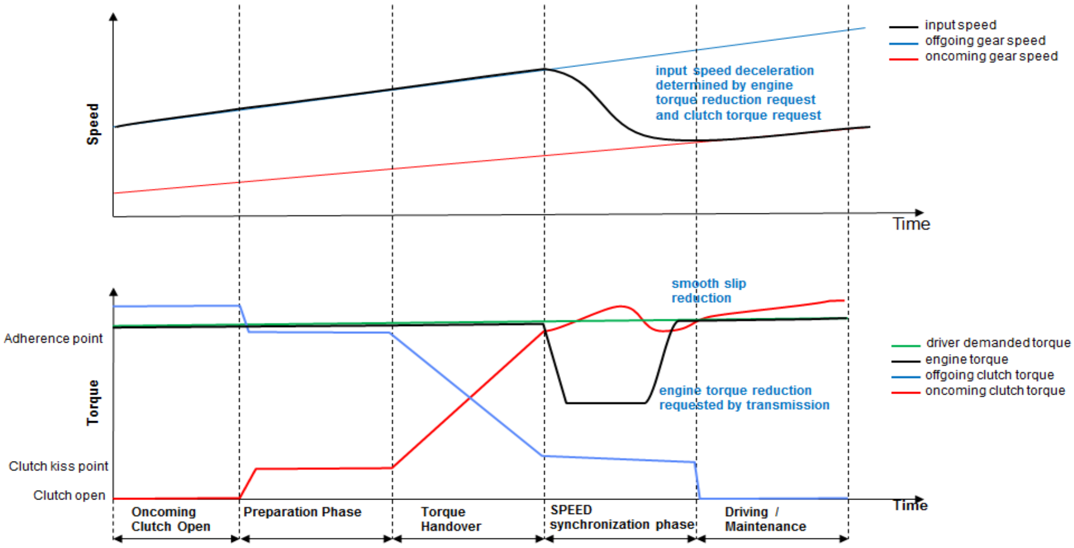

Figure 2.2 : Shift diagram of POUS.

Power on upshift (POUS): During a power on upshift, the power flows toward wheels from engine and the accelerator pedal is pressed. At the same time, an upper gear is targeted. Speed synchronization phase occurs after the torque handover phase. At speed synchronization phase, the engine torque is reduced and the input speed is decreased to oncoming gear speed quickly. Consequently, the time is required for speed synchronization becomes shorter (Figure 2.2). The shift has torque speed sequence (**TSS:** 1-PREP, 2-TORQUE, 3-SPEED, 4-REINST).

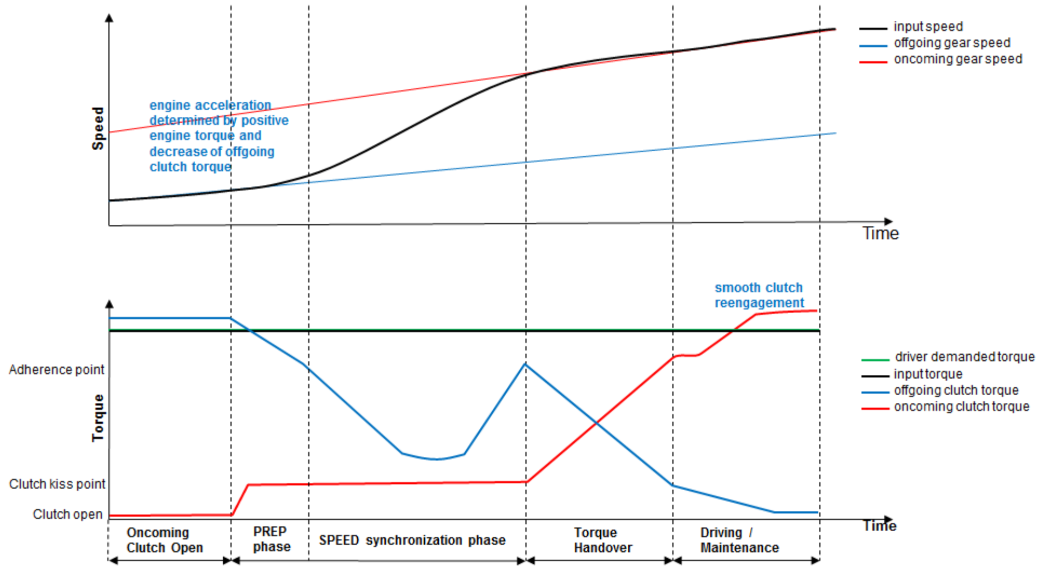

Figure 2.3 : Shift diagram of PODS.

Power on downshift (PODS): During a power on downshift, the power flows toward wheels from engine and the accelerator pedal is pressed. At the same time, a lower gear is targeted. Torque handover phase occurs after the speed synchronization phase (Figure 2.3). The shift has speed torque sequence (**STS:** 1-PREP, 2- SPEED, 3-TORQUE, 4-REINST). At speed synchronization phase, the input speed is increased [2].

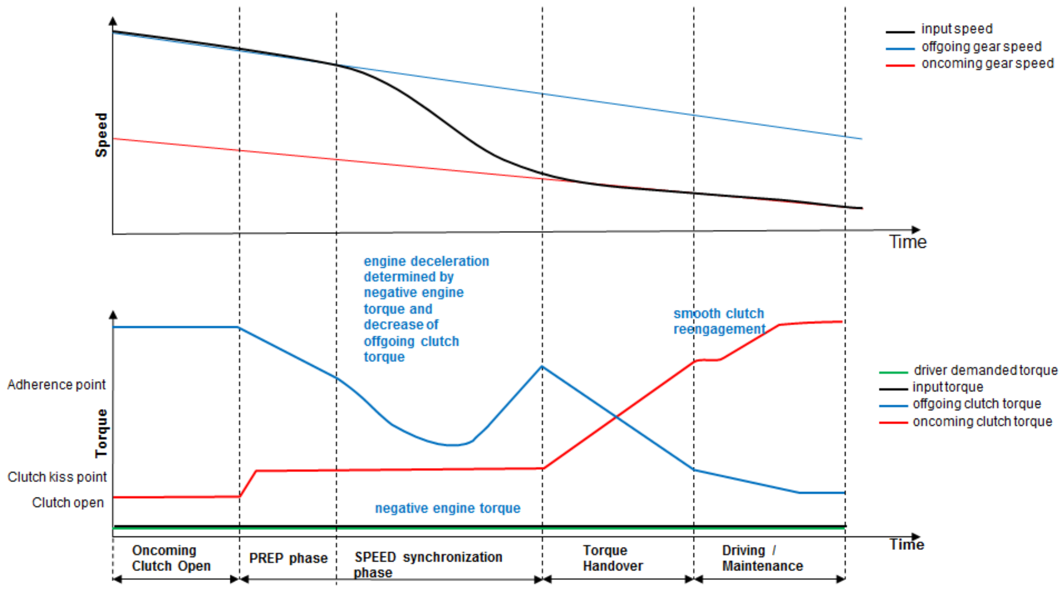

Figure 2.4 : Shift diagram of PFUS.

Power off upshift (PFUS): During a power off upshift, the power flows towards the engine from wheels and the accelerator pedal is released. At the same time, an upper gear is targeted. Torque handover phase occurs after the speed synchronization phase (Figure 2.4). The shift has speed torque sequence (**STS:** 1-PREP, 2- SPEED, 3-TORQUE, 4-REINST).

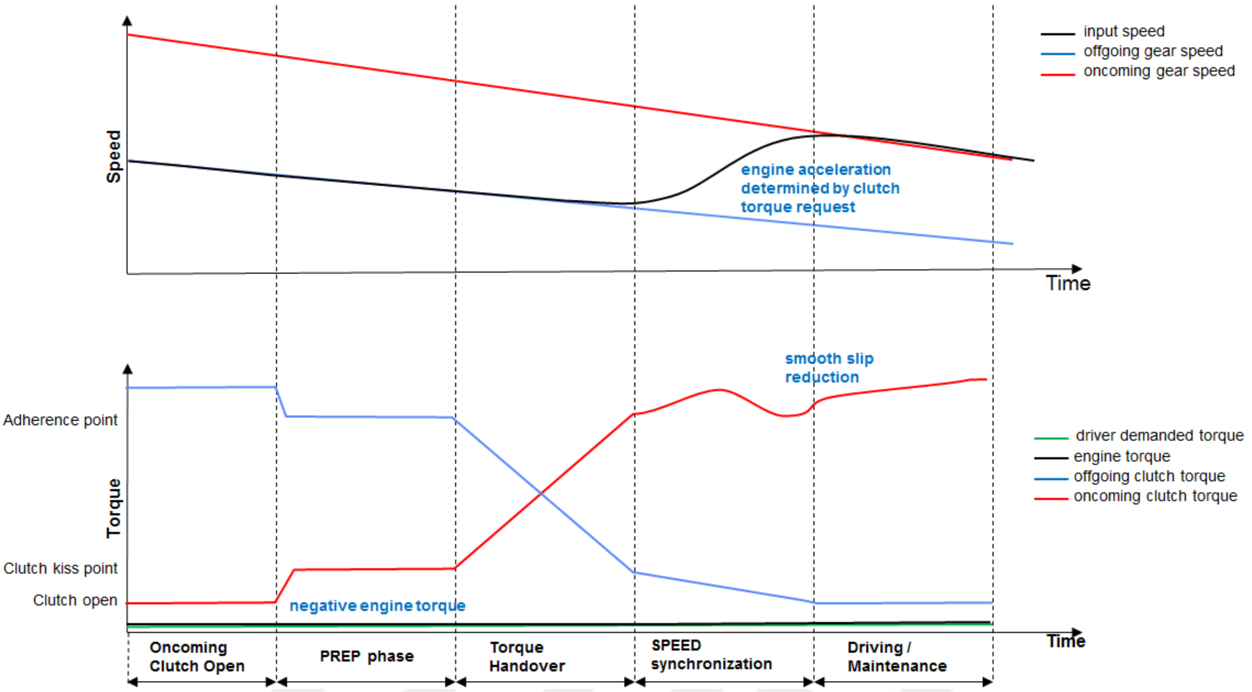

Figure 2.5 : Shift diagram of PFDS.

Power off downshift (PFDS): During a power off downshift, the power flows towards the engine from wheels and the accelerator pedal is released. At the same time, a lower gear is targeted. Torque handover phase occurs after the speed synchronization phase (Figure 2.5). The shift has torque speed sequence (**TSS:** 1-PREP, 2- TORQUE, 3-SPEED, 4-REINST) [2].

The purpose is to ensure that the torque transferring between the engaged clutch and the disengaged clutch during torque handover is not felt by the driver. Therefore, both the ramp function and the pid controller are used in the torque handover phase of the transmission strategy component in software. In the case where the ramp function is included in the software, the reference speed has been followed by transmission input shaft speed with high accuracy. However, in cases where the controller is active, the difference between the reference speed and transmission input shaft speed continuously changes. Therefore, the torque transfer between the oncoming clutch and offgoing clutch does not occur smoothly (Figure 2.6).

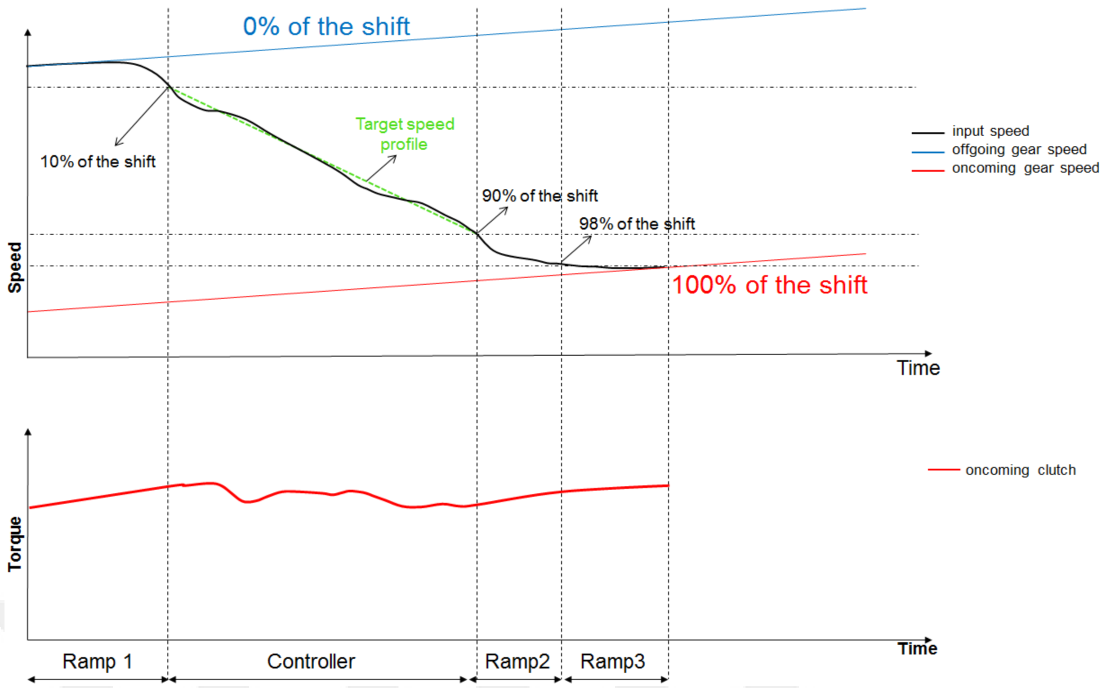

Figure 2.6 : Torque and speed results of POUS.

2.4 AVL Cruise™ Simulation

AVL Cruise™ is a simulation and test platform that can integrate with the Simulink environment and allows user to test a software made in a Simulink environment and examine the results (Figure 2.7). AVL Cruise™ models can act as the plant models and can be used to develop feedback control algorithms for any powertrain control application. PoET Tester is added to Simulink environment via block similar to a Simulink block. Test platform sends the input and supplementary signals into groups of vectors to the Simulink environment according to test cases that is defined within the block. The resulting output signals are sent to the PoET tester block, and review and visualization of the results are provided at this block.

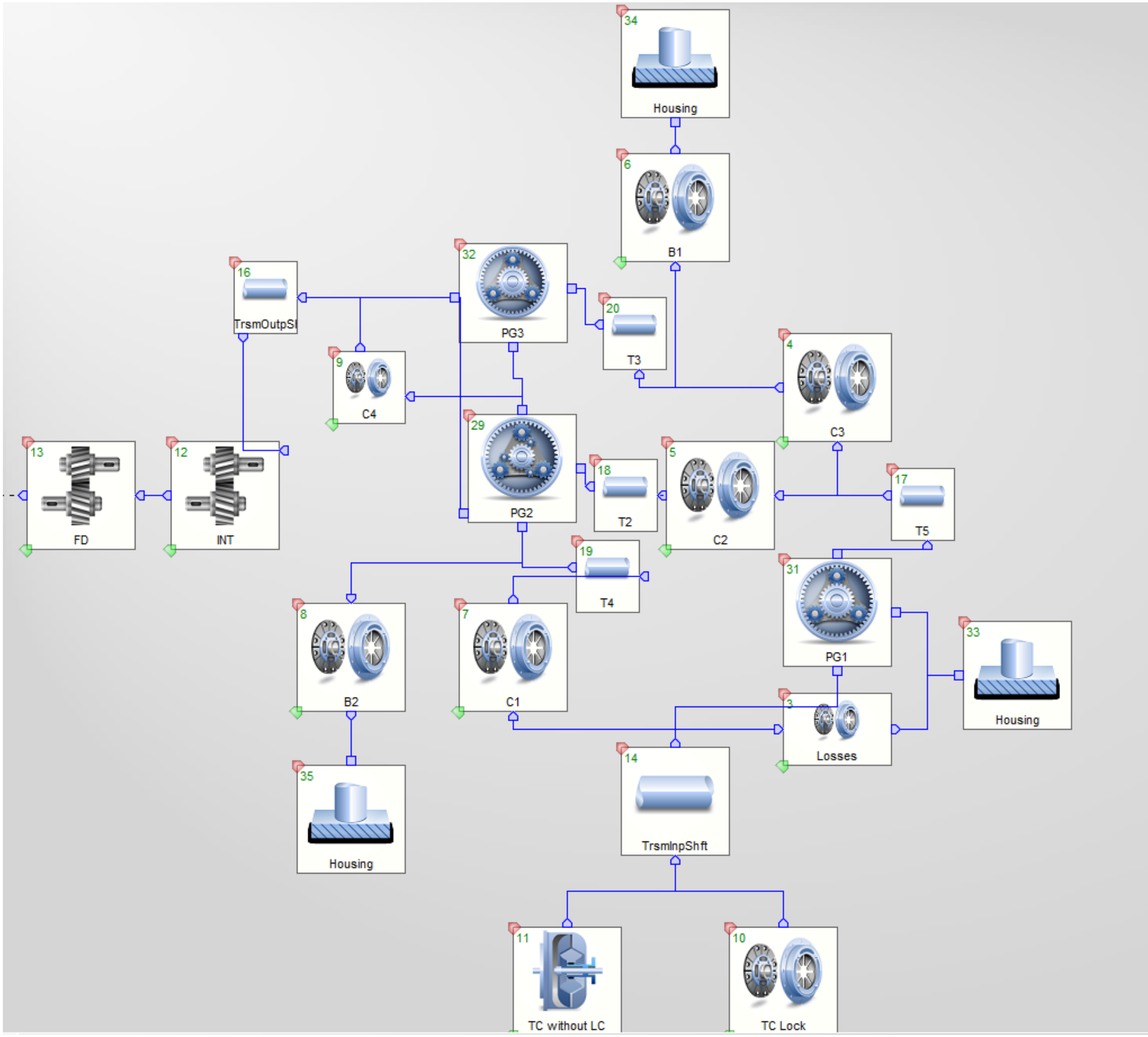

Figure 2.7 : AVL Cruise™ transmission model.

3. BOND GRAPH MODELING OF GEAR SET

3.1 Bond Graph Modeling Methodology

Bond graph modeling is used for illustrating the energy structure of mechanical/electrical/hydraulic systems. A half arrow sign represent the way of the flow and is named as bond. The bond identify the effort and flow elemets of power elements. The power is defined with multiplying of effort and flow element on the bond.

$$P(t) = e(t)\, f(t) \tag{3.1}$$

Resistance, capacitance and inertia elements like energy storage elements and effort or flow source elements take place in the bond graph modeling. For example, energy structure of transmission contain S_e elemet called effort source and this element indicates that transferring of torque from the engine to the transmission. In addition, since the sun gear of simple planetary gear set is connected to the chassis, the torque applied by the sun gear to the planetary gear is indicated by the S_f element [3].

3.1.1 Planetary gear set modeling

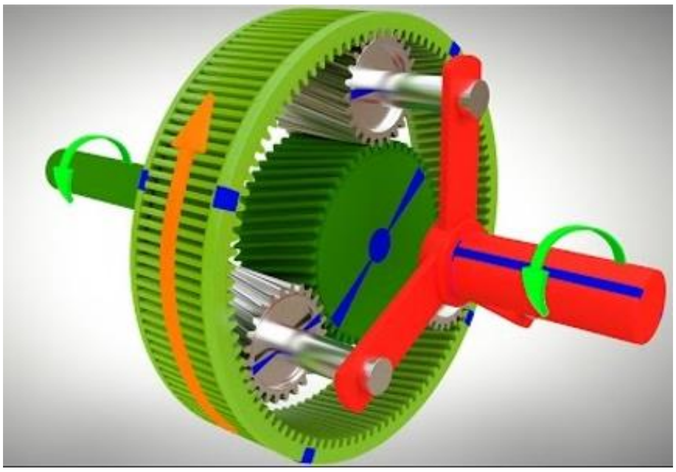

Figure 3.1 : Simple Planetary Gear Set.

Before modeling of the planetary gear sets, torque and speed relationship between gears as obtained. First, the planetary carrier is kept constant and velocity equations are constituted as follows .

$$w_s = -w_p * \frac{z_p}{z_s} \tag{3.2}$$

$$w_r = w_p * \frac{z_p}{z_r} \tag{3.2}$$

From equation 3.1 and 3.2,

$$\frac{w_s}{w_r} = -\frac{z_r}{z_p} \tag{3.3}$$

When the relative speed of the planetary carrier is added to the previous equations, the new velocity equations are as follows.

$$w_s = -w_p * \frac{z_p}{z_s} + w_c \tag{3.4}$$

$$w_r = w_p * \frac{z_p}{z_r} + w_c \tag{3.5}$$

From equation 3.4 and 3.5,

$$\frac{w_s - w_c}{w_r - w_c} = -\frac{z_r}{z_s} = -i_0 \tag{3.6}$$

The torque relationship between the gears is subtracted from the torque equations of the system in equilibrium

$$T_s + T_r + T_c = 0 \tag{3.7}$$

and

$$T_p = 0 \tag{3.8}$$

Since the planetary gear must be in balance, the following relation is obtained from the balance between sun and ring gear [4].

$$T_r * z_s - T_s * z_r = 0 \tag{3.9}$$

$$\frac{T_r}{T_s} = \frac{z_r}{z_s} = i_0 \tag{3.10}$$

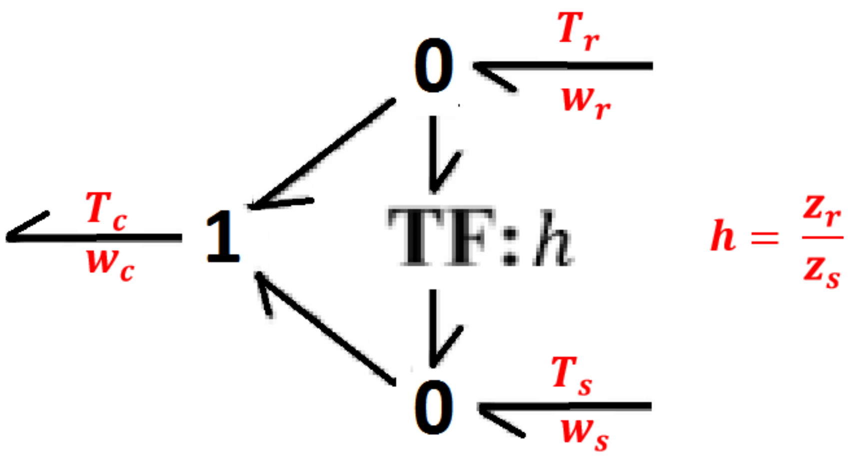

Figure 3.2 : Bond graph model of simple planetary gear set.

Angular velocity and torque equations supported by bond graph modeling.

$$w_c = \frac{w_s + w_r * h}{1 + h} \tag{3.11}$$

$$T_s = \frac{T_r}{h} \tag{3.12}$$

$$T_c = -T_r * (\frac{1}{h} + 1) \tag{3.13}$$

3.1.2 Ravigneaux planetary gear set modeling

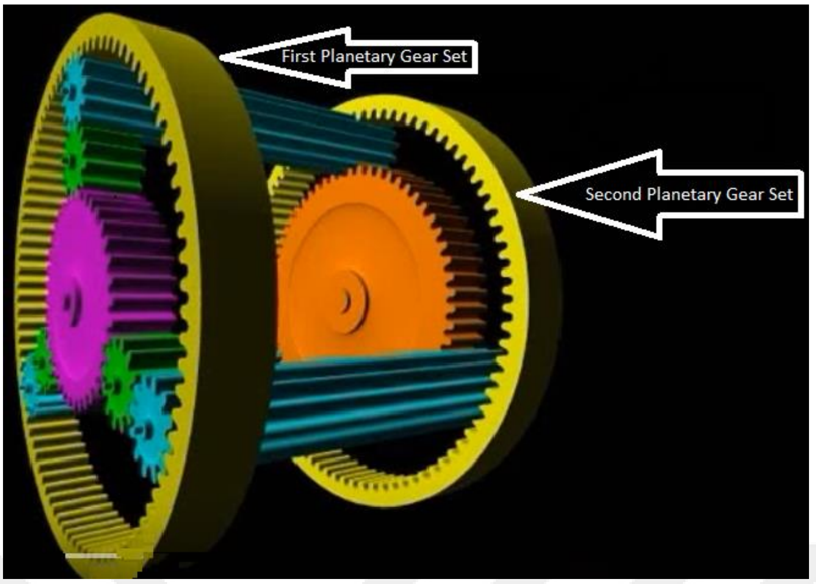

Figure 3.3 : Ravigneaux Planetary Gear Set (Retrieved from https://www.youtube.com/watch?v=Nbnyo9uUVKg.[16]).

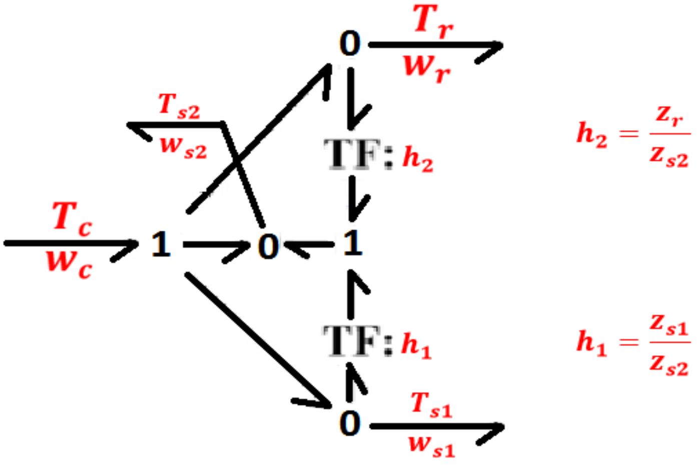

Figure 3.4 : Bond graph model of ravigneaux planetary gear set.

Angular velocity and torque equations supported by bond graph modeling [5].

$$T_{s_2} = \frac{T_r}{h_2} + \frac{T_{s_1}}{h_1} \tag{3.14}$$

$$w_{s_2} = w_{c_2} * (1 + h_2) - w_{r_2} * h_2 \tag{3.15}$$

3.1.3 Transmission modeling

The transmission is consists of a combined ravigneaux planetary gear set and planetary gear set with clutches and brakes connected to each other and the chassis. The seven-speed (7 forward and reverse) transmission is formed with 4 clutches and 2 brakes located in gearbox [6].

In the model created using the bond graph method, the I elements, shown in red, contain integral causality, while the I elements, shown in blue, contain derivative causality. The system has 6 gearbox inertia elements. Since three of them contain integral causality, the model is third order. The state space representation of the system is created by these three inertia elements. The other three angular velocities can be written in terms of angular velocities including integral causality[7].

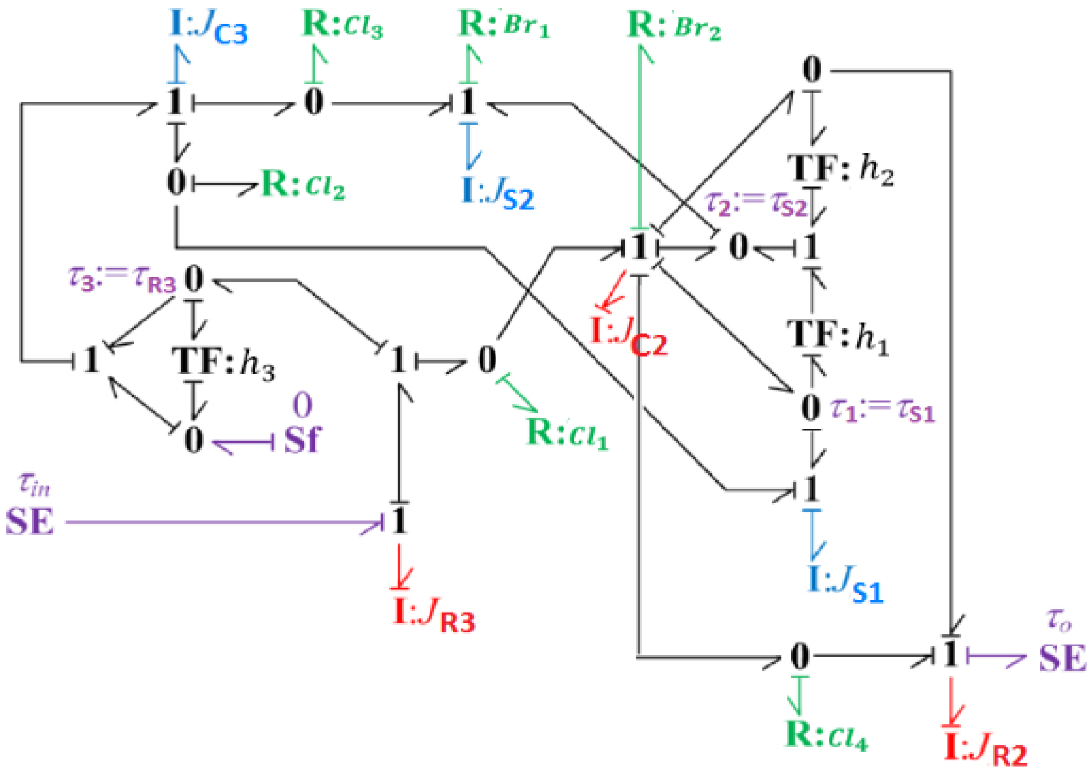

Figure 3.5 : Bond graph model for 7-speed transmission (Retrieved from reference [7]).

Parameters and obtained equations used in bond graph modeling [7]-[10]:

h_1, h_2 and h_3 : gear ratios of planetary gear set and combined ravigneaux planetary gear set

A: inertia matrix $\in \mathbb{R}^{3x3}$, B: input matrix $\in \mathbb{R}^{3x8}$

$$h_1 = \frac{z_{s1}}{z_{s2}} \tag{3.16}$$

$$h_2 = \frac{z_r}{z_{s2}} \tag{3.17}$$

$$h_3 = \frac{z_{r3}}{z_{s3}} \tag{3.18}$$

$$A = \begin{bmatrix} a_{11} & 0 & 0 \\ 0 & a_{22} & a_{23} \\ 0 & a_{32} & a_{33} \end{bmatrix} \tag{3.19}$$

$$a_{11} = I_{R3} + I_{C3}\left(\frac{h_3}{1+h_3}\right)^2$$

$$a_{22} = I_{C2} + I_{S2}\left(h_2{}^2 + 2h_2 + 1\right) + I_{S1}\left(\left(\frac{h_2}{h_1}\right)^2 - 2\frac{h_2}{h_1} + 1\right)$$

$$a_{23} = -I_{S2}\left(h_2{}^2 + h_2\right) + I_{S1}\left(\frac{h_2}{h_1} - \left(\frac{h_2}{h_1}\right)^2\right)$$

$$a_{32} = -I_{S2}\left(h_2{}^2 + h_2\right) + I_{S1}\left(\frac{h_2}{h_1} - \left(\frac{h_2}{h_1}\right)^2\right)$$

$$a_{33} = I_{R2} + I_{S2}h_2{}^2 + I_{S1}\left(\frac{h_2}{h_1}\right)^2$$

$$B = \begin{bmatrix} -1 & -\left(\frac{h_3}{1+h_3}\right) & -\left(\frac{h_3}{1+h_3}\right) & 0 & 0 & 0 & 0 & 1 \\ 1 & 1-\frac{h_2}{h_1} & 1+h_2 & -1 & -h_2-1 & -1 & 0 & 0 \\ 0 & \frac{h_2}{h_1} & -h_2 & 1 & h_2 & 0 & -1 & 0 \end{bmatrix} \tag{3.20}$$

$$U = \begin{bmatrix} 1 & -1 & 0 \\ \left(\frac{h_3}{1+h_3}\right) & \frac{h_2}{h_1} - 1 & -\frac{h_2}{h_1} \\ \left(\frac{h_3}{1+h_3}\right) & -1-h_2 & h_2 \\ 0 & 1 & -1 \\ 0 & 1+h_2 & -h_2 \\ 0 & 1 & 0 \\ 0 & 0 & 1 \\ 1 & 0 & 0 \end{bmatrix} \tag{3.21}$$

$$L = A^{-1}B \tag{3.22}$$

$$\dot{w}_S = \begin{bmatrix} \dot{w_{R3}} & \dot{w_{C2}} & \dot{w_{R2}} \end{bmatrix}^T \tag{3.23}$$

$$w_{clutch} = U\,\dot{w}_S \tag{3.24}$$

$$A * \dot{w}_s = B * [\tau_{Cl_1}\tau_{Cl_2}\tau_{Cl_3}\tau_{Cl_4}\tau_{Br_1}\tau_{Br_2}\tau_{out}\tau_{in}]^T \tag{3.25}$$

$$\begin{aligned} &[\tau_{Cl_1} \quad \tau_{Cl_2} \quad \tau_{Cl_3} \quad \tau_{Cl_4} \quad \tau_{Br_1} \quad T_{Br_2}] \\ &\qquad = f\left(\left[\left[w_{Cl_1} \quad w_{Cl_2} \quad w_{Cl_3} \quad w_{Cl_4} \quad w_{br_1} \quad w_{br_2}\right]\right]\right) \\ &= f\left(\left[\left[w_{R3} - w_{C2} \quad w_{C3} - w_{S1} \quad w_{C3} - w_{S2} \quad w_{C2} - w_{R2} \quad w_{S2} \quad w_{C2}\right]\right]\right) \end{aligned} \tag{3.26}$$

3.2 Clutch Friction Modeling

In this section, different clutch friction models are developed which are derived from the friction models. These friction models used during modeling of dynamic systems have different effects on the process load, the accuracy of modeling, and the closeness of the results to reality.

In general there is a state of friction between the two function surfaces which are in contact with each other and make relative sliding or rolling motion [8].

The friction equation which expresses the friction state is as follows;

$$T_{fmax} = N_f F_{app} \mu r_e \tag{3.27}$$

N_f: the number of friction surfaces

F_{app}: force applied to clutch surfaces

μ: surface friction coefficient

r_e: clutch geometry coefficient

$$r_e = \frac{2}{3}\left(\frac{{R_o}^3 - {R_i}^3}{{R_o}^2 - {R_i}^2}\right) \tag{3.28}$$

R_o: outer radius

R_i : inner radius

Static and dynamic friction torque values respectively T_{fs} and T_{fd} are obtained by normalized command signal (com) according to maximum friction torque.

$$T_{fd} = com \, . \, T_{fmaxd} \tag{3.29}$$

$$T_{fs} = com \, . \, T_{fmaxs} \tag{3.30}$$

In many cases, the numerical value of friction can be very small, but practically always present. The most common static friction models are as follows.

3.2.1 Coulomb friction model

The coulomb friction model, one of the fundamental friction models, is used to obtain the friction torque that occurs in clutches.

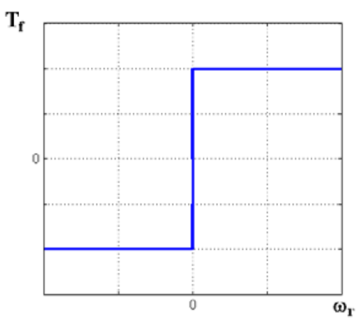

Figure 3.6 : Coulomb friction model.

The clutch output torque obtained with the Coulomb friction model is as follows.

$$T_c = \begin{cases} T_{fd}.\,\mathrm{sgn}(w_r) & \text{if } w_r \neq 0 \\ T_{app} & \text{if } w_r = 0 \end{cases} \tag{3.31}$$

T_c: torque transmitted via clutch

w_r: relative speed between the clutch surfaces

T_{app}: applied torque to clutch plates

The friction torque equation is modeled together with the switching structures in clutch models since it is undefined when the speed is zero.

3.2.2 Combined coulomb and viscous friction model

The viscous friction model can be preferred in dynamic systems because of the nonlinearity of the motion equations. Although the viscous friction model is advantageous in terms of simulation, cannot fully reflect the clutch friction model. For this reason, it is more advantageous to use the combined Coulomb-viscous friction model. As in the Coulomb friction model, the combined coulomb and viscous friction model can be easily modelled using a saturation block at simulink.

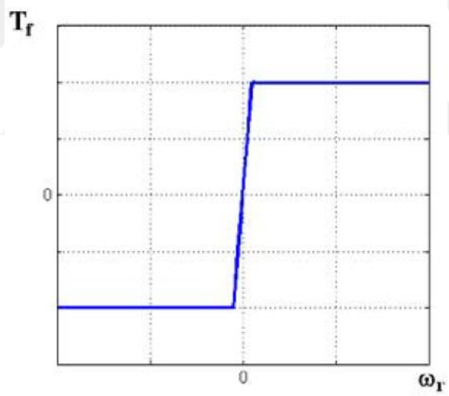

Figure 3.7 : Combined Coulomb and viscous friction model.

$$T_c = \begin{cases} T_{fd}\,.\min(2w_r/w_0, 1) & if\ w_r \geq 0 \\ T_{fd}.\max(2w_r/w_0\,, -1) & if\ w_r\ < 0 \end{cases} \tag{3.32}$$

w_0: the parameter that determines the speed of the transition form to -1 to 1

3.2.3 Static friction model

In engineering problems, the friction torque is takes any value without exceeding the breakaway torque around zero speed during modelling of friction force. If the torque around the zero speed gets values on a linear graph, this is avoided and simulink modeling can be performed. The static friction model is used for this purpose [9].

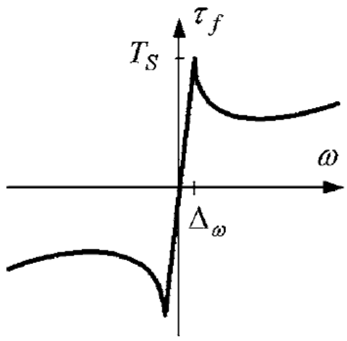

Figure 3.8 : Static friction model.

$$T_f(w) = \left[T_c + (T_s - T_c)e^{-\left|\frac{w}{w_s}\right|^{\delta}} + K_v|w|\right] sgn(w) \tag{3.33}$$

3.2.4 Karnopp friction model

The uncertainty state where is occured when the angular velocity is equal to zero in the static friction model disappears in the Karnopp friction model. To remove this uncertainty, the friction force takes on the values on a straight line with a specific slope as in the classical friction model, or it takes a value which is not depend on the speed in the $\boldsymbol{\Delta_w}$ range but saturated by the breakaway torque by depend on applied force [10].

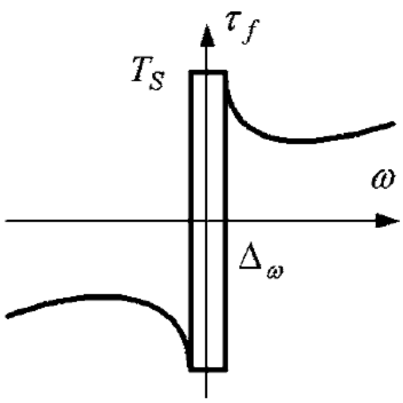

Figure 3.9 : Karnopp friction model.

$$T_f(w) = \begin{cases} T_{fslip}(w) = \left[T_c + (T_s - T_c)e^{-\left[\frac{|w|-\Delta_w}{w_s}\right]^{\delta}} + K_v|w|\right] sgn(w), & for\ |w| \geq \Delta_w \\ T_{fstick}(w) = sat(T_{stick}, \pm T_s) & otherwise \end{cases}$$

4. MODEL PREDICTIVE CONTROL

Model predictive control is one of the important technical methods used to solve an optimization problems encountered during control of physical systems. Especially, it has great advantages in the control of multi input multi output processes. Unlike existing control systems, the controller design is implemented by foreseeing the variables to be obtained after a few steps to be taken from the system outputs.

4.1 Model Predictive Control Approach

The main purpose of model predictive control is to send the next control signal according to this result by predicting the future response of the system to which the control signal sent system. There are several important terms are required (Figure 4.3). One of them is a moving horizon window. The length of this time-dependent window is constant and it indicates how long prediction horizon in the future is. The control signal generated for the prediction horizon that has $\mathbf{T_p}$ length is only applied to the first sampling time/sample and the remaining trajectory of the prediction horizon is ignored. The control horizon is always chosen to be less than or equal to the prediction horizon. In order to produce the optimum solution, the controller generates the control signal by means of cost function to minimize the difference between the desired values and the system responses [11].

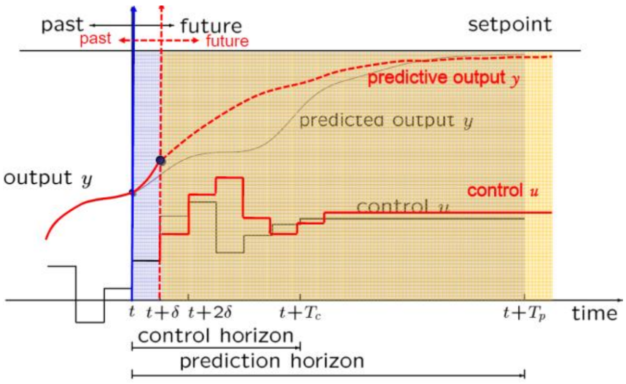

Figure 4.1 : Key terms of model predictive control.

4.2 MPC Application in Automotive Systems

There are constant improvements in the automotive industry in terms of reliability, fuel consumption, drivability, comfort etc. In order to fulfill these requirements, it is desirable that complex systems like vehicle, engine, transmission, suspension system respond as closely as possible to real physical systems. These systems contain nonlinear behaviors like static and dynamic friction forces, uncertainty on tires-ground contact under different constraints. These are multi input multi output control problems by their nature. While the model predictive control responds to these type of requirements, but also yields results quickly. In automotive applications, there are difficulties such as the fact that the systems should work at certain ranges in terms of physical constraints, the dynamic systems working together and interacting with each other and the systems have uncertainties and disruptive inputs affecting them. Automotive cases where model predictive control is used are emission regulation, drivability, driving comfort, safety constraints, active steering, vehicle tilting dynamics, traction control, semiactive suspensions, engine control and transmission gear shifting.

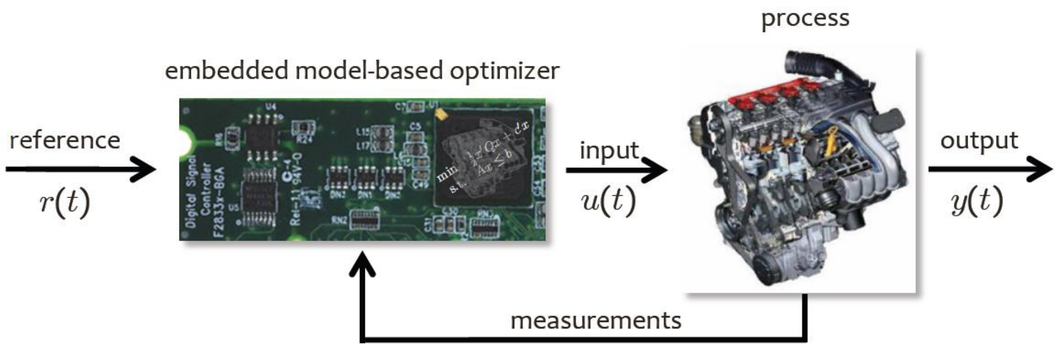

Figure 4.2 : MPC control loop.

4.3 Model Predictive Control of Torque Handover Phase

Particularly during gear shifting, the necessary torque and pressure transitions are realized with the aid of numerical maps during the engagement and disengagement of the oncoming and offgoing clutches. This requires an enormous calibration effort. For this purpose, by using model predictive control, it is desired to remove the generated torque-pressure maps and to design a suitable controller that responds well to physical gear shifting models. In this study, during the 3-4 upshift, power on upshift process is analyzed and it is desirable to ensure that the torque and speed transitions are smooth in the speed synchronization phase. There are speed torque sequence (STS) or torque speed sequence (TSS) in the four different gear types mentioned above. In the gear shift that has torque speed sequence, the controller requests torque for the oncoming clutch, otherwise it demands torque for the offgoing clutch. The desired torque values are necessary for the hydraulic system for opening and closing of the oncoming and offgoing clutches. It is expected that the transmission input shaft speed converge to the desired speed profile with the movements of the clutches.

While describing the events that occurred during the power on up shift type, the gear shifting from the third gear to the fourth gear was taken. In addition, they are explained through the oncoming and offgoing clutches of this transmission model involved in this shift (Figure 4.3).

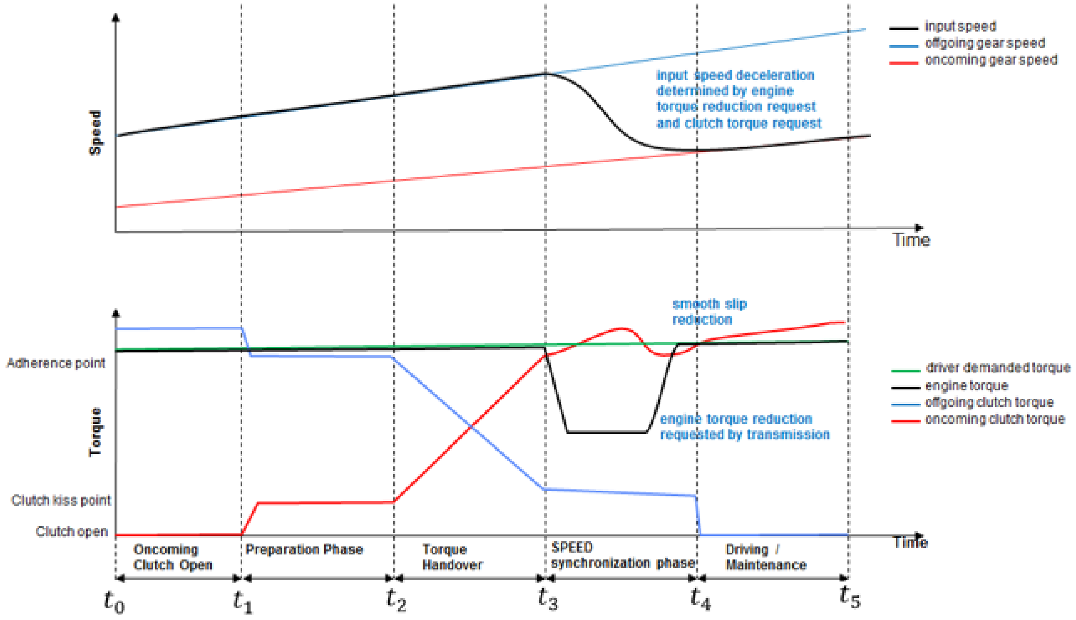

Figure 4.3 : Shift diagram of POUS with intervals.

Preparation Phase: This phase occurs at the interval t_1 - t_2 seconds. During this period clutch C_3 is engaged and torque is transmitted through this clutch and clutch C_1 is disengaged and torque transition through this clutch does not occur. Clutch C_2 is engaged and its torque value transferred during this time does not change. Engine and driver demand torque is constant and output torque is constant. There is a gear ratio between output torque and engine torque.

Torque Handover Phase: Torque phase occurs at interval t_2- t_3 seconds. The pressure of oncoming cluth C_1 starts to increase, so while the torque of the oncoming clutch increases, the torque of the offgoing clutch C_3 decreases. The plates of the oncoming clutch C_1 start to open and the torque transmitted through this clutch decreases. The torque from the input shaft is transferred to the oncoming and offgoing clutches depending on the gear ratios [12].

Speed Synchronization Phase: It is also called inertia phase. Inertia phase occurs at corresponding to interval t_3 - t_4 seconds. As the torque of the oncoming clutch C_1 continues to increase, the plates of the offgoing clutch C_3 open and its torque goes down during this period. Depending on the torque of the oncoming clutch, engine torque is intervented by control algorithm to provide transmission input torque and motor torque decreases. Before demanding of gear shift, transmission output shaft speed is proportional to the gear ratio of the third gear. During gear shifting, the transmission output shaft speed approaches the speed of the output shaft speed is proportional to the gear ratio of the fourth gear [13].

At the end of the speed synchronization phase, the torque of the offgoing clutch reduces to zero and the clutch is fully opened. At the same time, the torque of the oncoming clutch is increased (Figure 4.4).

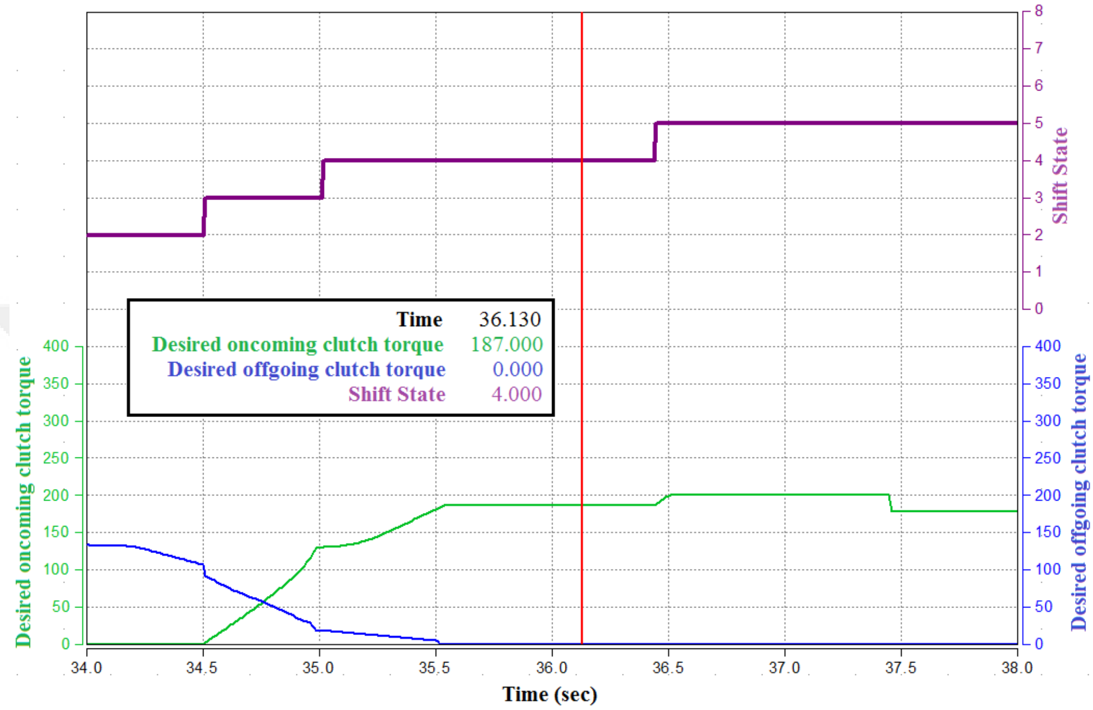

Figure 4.4 : Torques of clutches at speed synchronization phase.

The software with model predictive control or PID control runs a test include upshift and downshift stages respectively in Simulink environment in both cases. When the percentage of accelerator pedal position is increased or decreased, the vehicle has executed both upshift and downshift states (Figure 4.5).

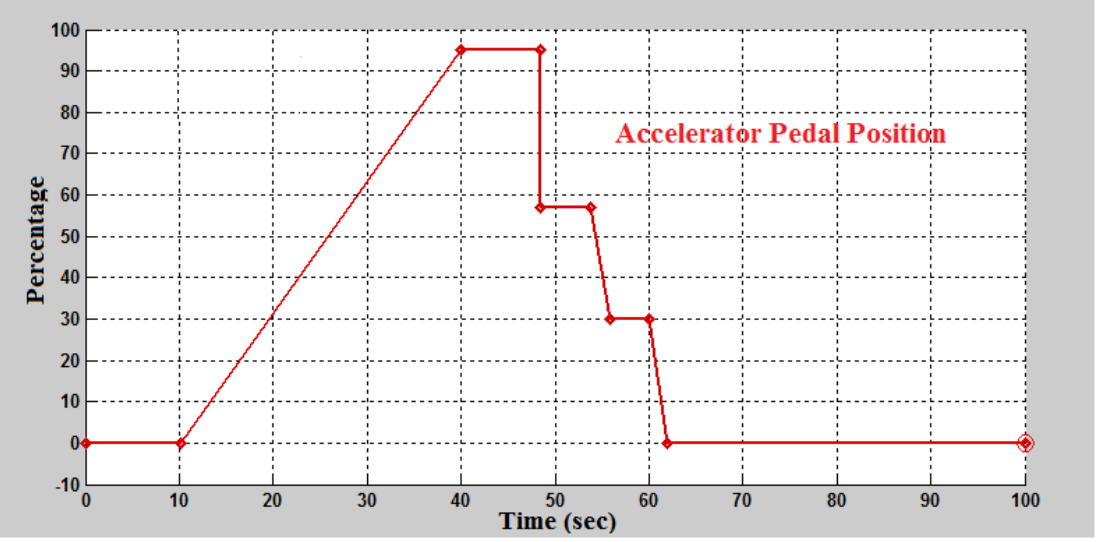

Figure 4.5 : Accelerator pedal position.

When simulation tests are performed,shift types like power on upshift and downshift and power off downshift are occured. Each shift states within the transmission strategy occurs in each shift types (Figure 4.6).

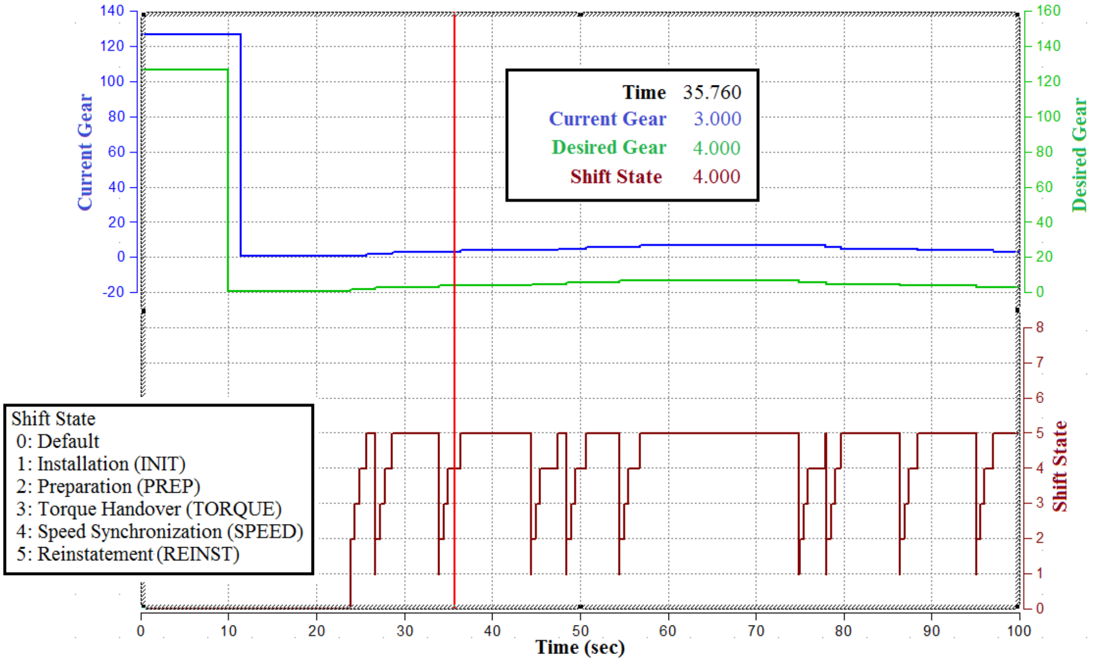

Figure 4.6 : Shift states.

As mentioned above, there are two different situations in the phase sequence during gear shifting in torque handover strategies. In one of them, the speed synchronization comes before the torque transfer phase, while in the other comes after.

PID Control for Gear Shifthing:

While the vehicle is passing from third gear to fourth gear, the results of the applied PID control application is as follows. In this case, the vehicle is in the power on upshift (POUS) gear type and torque speed sequence (TSS) is valid. If shift state is equal to 4, the speed synchronization phase is occured. At the beginning of this phase, the oncoming clutch is still slipping. For the torque value that should have for the next gear, the oncoming clutch needs to be closed. In the speed synchronization phase, as soon as the plates of the clutch get closer to each other, the torque value to be transmitted by the oncoming clutch is reached. At the same time, the offgoing clutch is opened and the transmitted torque value by the offgoing clutch goes down to zero (Figure 4.7).

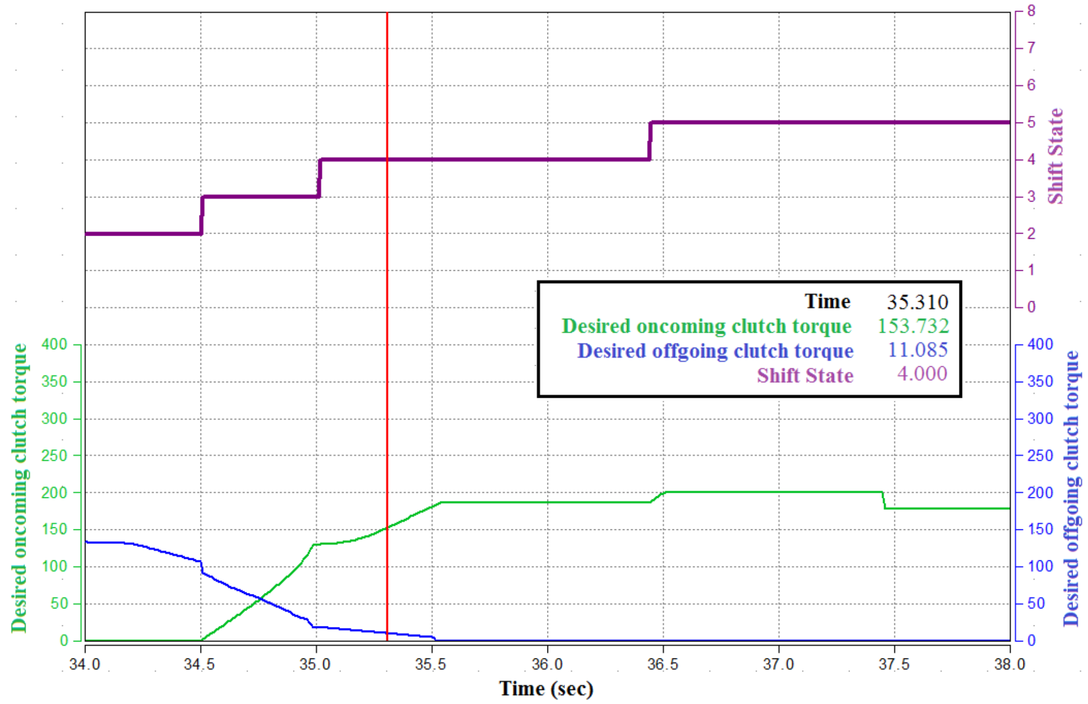

Figure 4.7 : Torques of oncoming and offgoing clutches.

While the transition to the fourth gear, first clutch is the oncoming clutch. One side of this oncoming clutch is connected to the transmission input shaft, while the other side is connected to the fourth shaft. The purpose of the PID controller is to ensure that the input shaft speed approaches the required transmission input shaft speed for the next gear quickly (Figure 4.8).

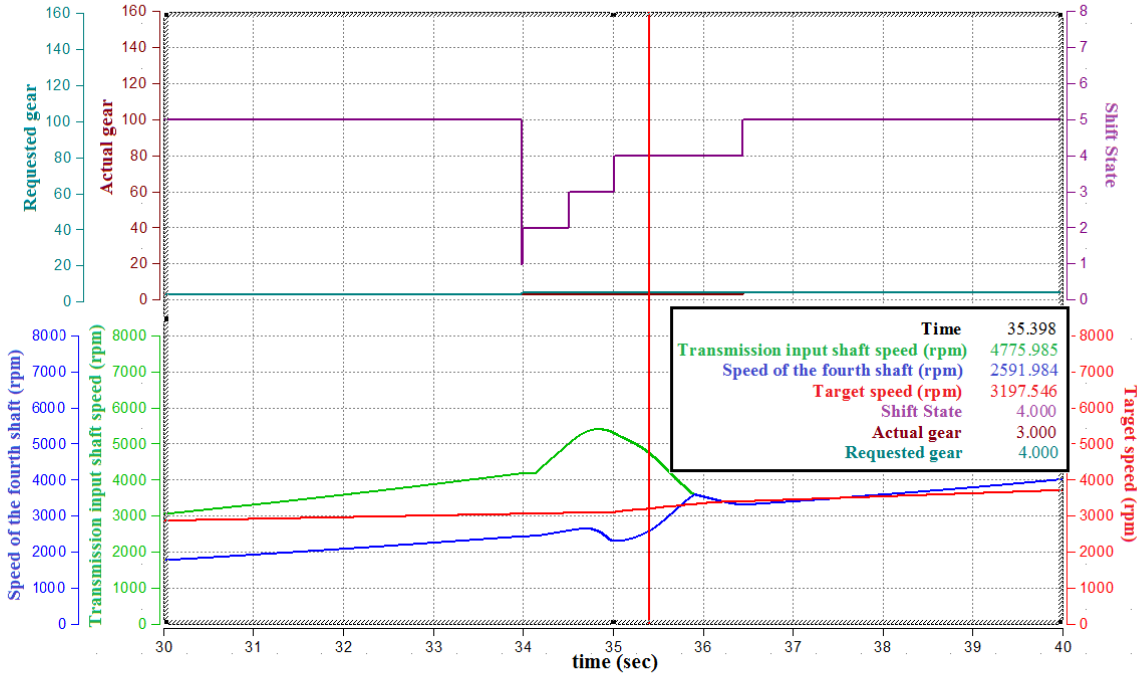

Figure 4.8 : Speeds of oncoming and offgoing clutches.

Model Predictive Control for Gear Shifthing:

Standart Cost Function:

$$J(z_k) = J_y(z_k) + J_u(z_k) + J_{\Delta u}(z_k) + J_\varepsilon(z_k)$$

The model predictive control is concerned with a second order optimization problem. Main elements of the cost function are output reference tracking, manipulated variable tracking, manipulated variable move suppression and constraint violation and their weight factors are also different [17]. According to the MPC design and system requirements, output weights are set to zero by adjusting the weights of the components other than the output reference tracking, which is one of the components of the cost function and output reference tracking is done.

Output Reference Tracking:

$$J_y(z_k) = \sum_{j=1}^{n_y} \sum_{i=1}^{p} \left\{ \frac{w_{i,j}^{y}}{s_j^{y}} [r_j(k+i|k) - y_j(k+i|k)] \right\}^2$$

Manipulated variable tracking:

$$J_u(z_k) = \sum_{j=1}^{n_u} \sum_{i=0}^{p-1} \left\{ \frac{w_{i,j}^u}{s_j^u} [u_j(k+i|k) - u_{j,\text{target}}(k+i|k)] \right\}^2$$

k — current control interval

p — Prediction horizon (number of intervals)

n_y — Number of system output variables

z_k — quadratic problem decision, given by,

$$z_k{}^T = [u(k|k)^T \; u(k+1|k)^T \ldots \ldots u(k+p-1|k)^T \;\; \varepsilon_k]$$

$y_j(k+i|k\,)$ — predicted value of jth plant output at ith prediction horizon step

$r_j(k+i|k)$ — reference value for jth plant output at ith prediction horizon step

s_j^y — scale factor for jth plant output

$w_{i,j}^y$ — tuning weight for jth plant output at ith prediction horizon step (dimensionless)

The gearbox model, which is third order, has 3 states. These are the angular speed of the shaft that connect the first clutch and the second planetary gear set, and the angular speeds of the input and output shafts. Speeds in other shafts can be written by these states. These;

w_{R3} : number of revolutions of 3rd ring gear (x_1)

w_{C2} : number of revolutions of the second carrier (x_2)

w_{R2} : number of revolutions of 2nd ring gear (x_3)

Manipulated variable (mv): Torque value/request for oncoming clutch

Manipulated disturbance (md): torque values of clutch other than oncomig clutch

System output (y = x_1): angular velocity of transmission input shaft

q: weight factor of output reference tracking

r: weight factor of manipulated variable tracking

Controller parameters: sampling time (T_s), prediction horizon (N_p) and control horizon (N_c).

The sampling time, which is one of the required parameters for the controller design in the model predictive control application, is the same as the sampling time of the software. Control horizon (N_c) is selected as 3. Prediction horizon (N_p) is selected as 10. The control horizon must be smaller than or equal to the prediction horizon.

When only the output reference tracking is considered in the cost function, that is, when the manipulated variable tracking weight is zero, the following results are observed. Synchronization of the speed of the fourth shaft with the transmission input shaft, which should occurs in speed synchronization phase, occurs in the reinstatement phase. In the case where the PID control application is involved, the speed equalization occurs in the speed synchronization phase,but it doesn't occur in the MPC application, which only takes into account the output reference tracking. Because the time required for speed synchronization has increased. So, the gear shifting time increases (Figure 4.9). Since the amount of required torque for the oncoming clutch is not sent to the transmission software, the necessary clutch conditions for the next gear have not occurred (Figure 4.10).

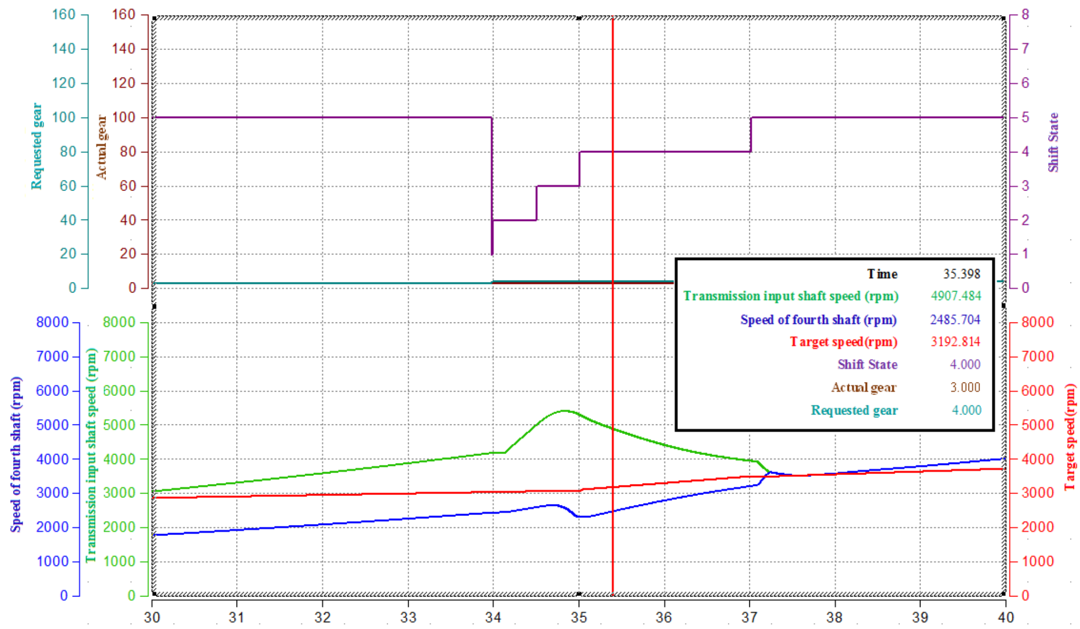

Figure 4.9 : Speeds of oncoming and offgoing clutches (q:0.001, r:0).

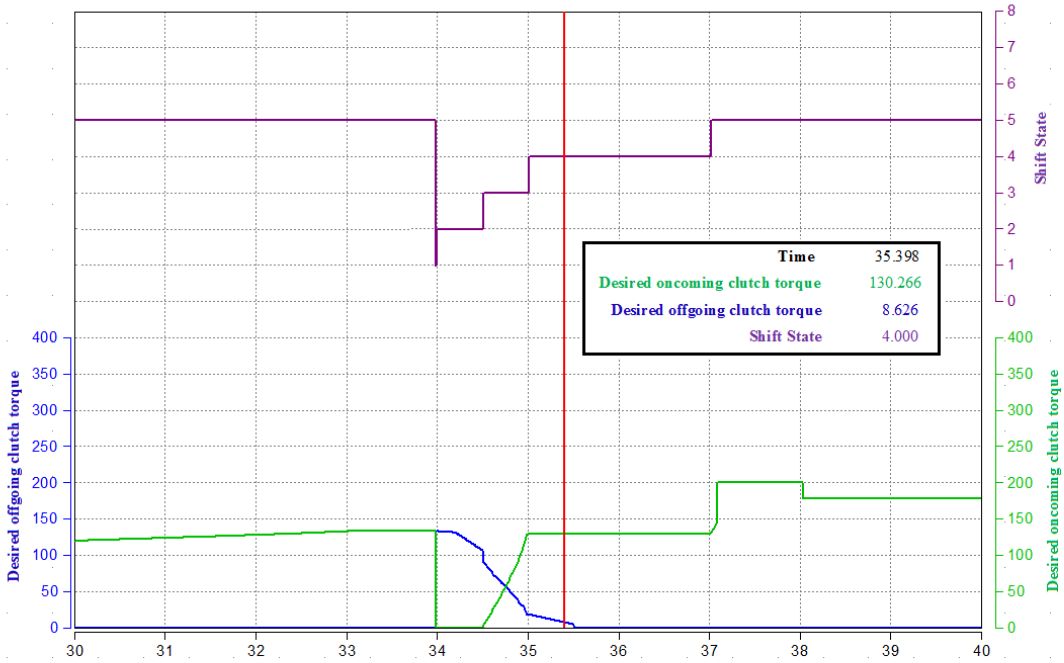

Figure 4.10 : Torques of oncoming and offgoing clutches (q:0.001, r:0).

While manipulated variable tracking performance is not considered, the weight of output reference tracking performance is increased. Decrease in gear shift time was not observed (Figure 4.11). At the same time, the transmission input shaft speed could not follow the desired speed profile. In addition, speed synchronization has not been achieved smoothly. Requested torque was sent to the system for oncoming clutch.

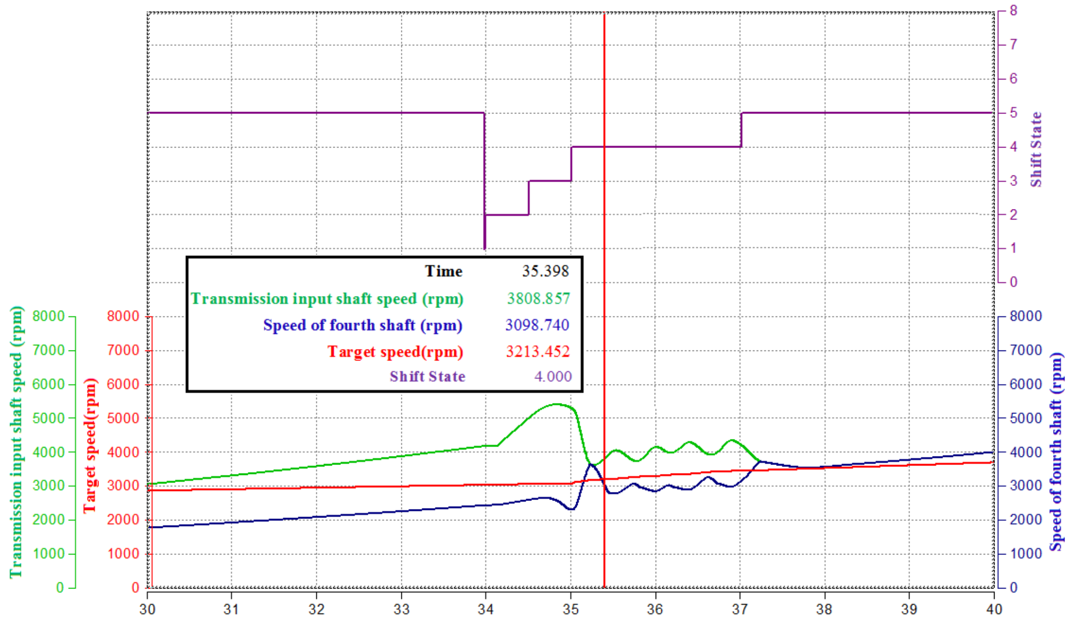

Figure 4.11 : Speeds of oncoming and offgoing clutches (q:1, r:0).

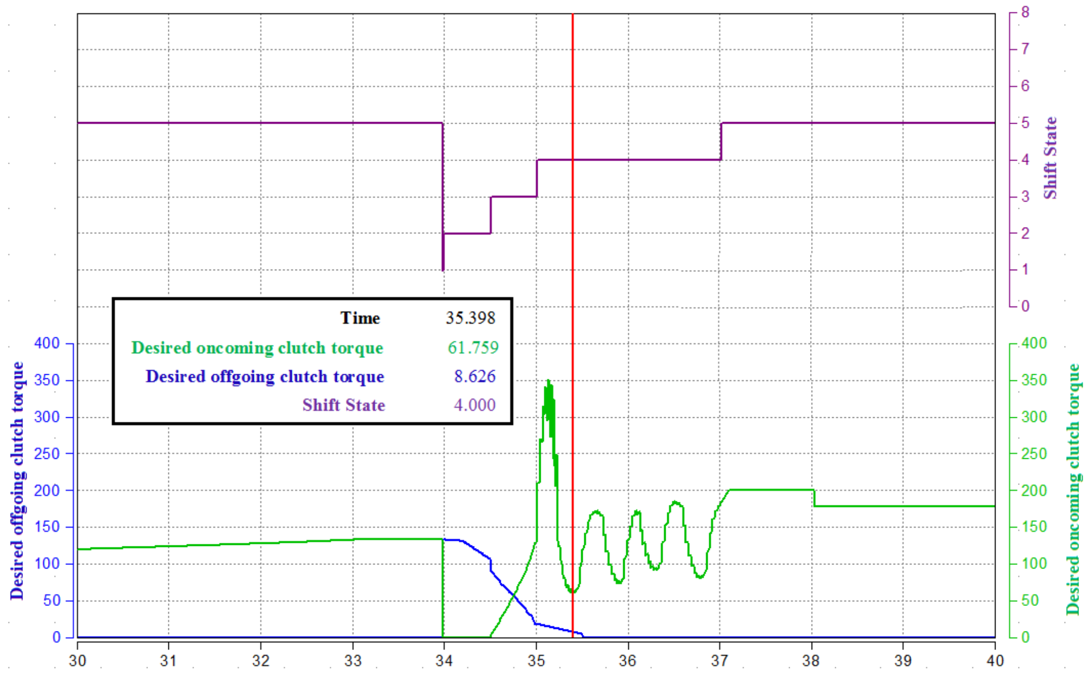

Figure 4.12 : Torques of oncoming and offgoing clutches (q:1, r:0).

Reducing the changes in torque values has not been observed during speed synchronization phase, it has increased on the contrary (Figure 4.12). These changes in torque are undesirable results for driving comfort [14].

When both of the output reference tracking and manipulated variable tracking are considered in the cost function, the results obtained with different weight parameters were evaluated. When the manipulated variable tracking performance is taken into consideration, the changes in torque are reduced. But the gear shift time is still long and speed synchronization is occurred in the next reinstatement phase [15].

When the speed reference tracking weight factor is higher than the manipulated variable tracking weight factor, the obtained speed and the changes in the desired torque values are as follows. The changes in the torque value of the oncoming clutch are high (Figure 4.13). Speed synchronization phase duration is long and speed equality occurs in the next phase (Figure 4.14). In the software where the PID controller is included, the speed synchronization phase duration is shortened by changing Kp, Ki values. In the software where the MPC application is located in, period can be shortened to 1.4 seconds. While the fluctuations in the desired torque value affect the clutch speeds, the fluctuations at clutch speed also cause the difference between the requested torque value and the clutch torque at that moment to be excessive.

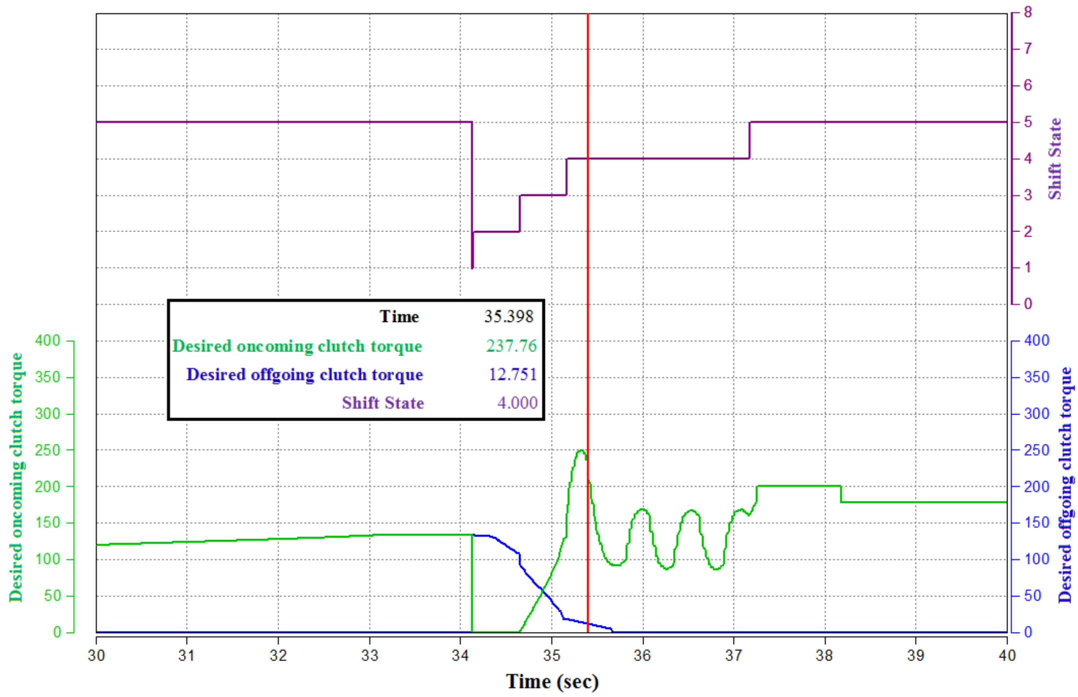

Figure 4.13 : Torques of oncoming and offgoing clutches (q:0.9, r:0.1).

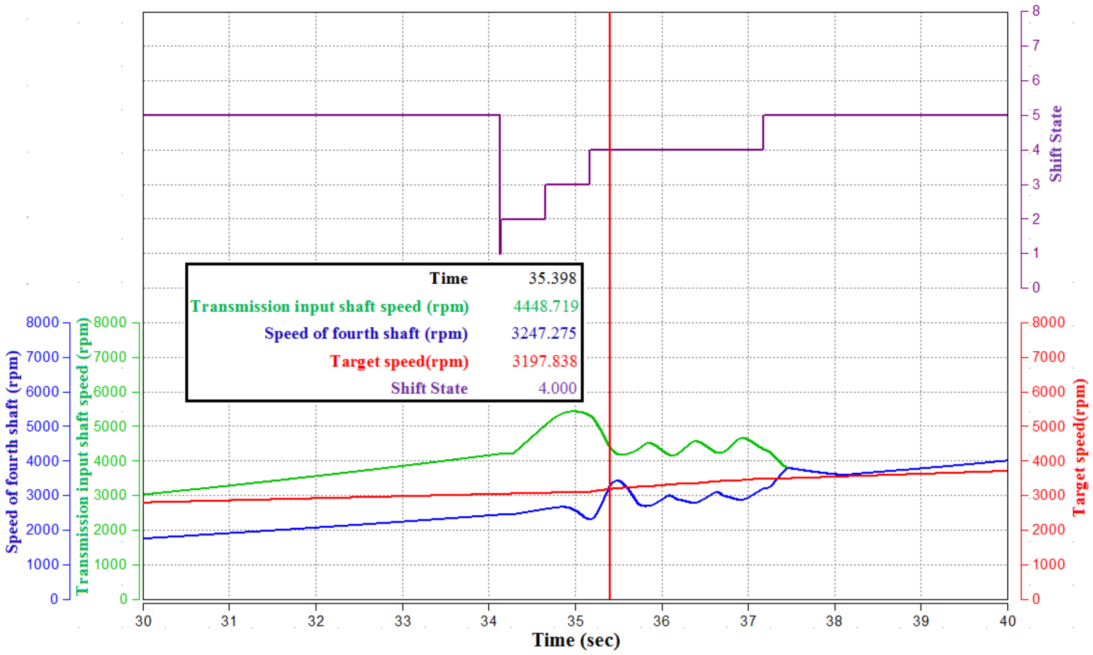

Figure 4.14 : Speeds of oncoming and offgoing clutches (q:0.9, r:0.1).

5. CONCLUSIONS AND RECOMMENDATIONS

In this reseach, the goal was to achieve important objectives such as the automatic transmission modelling by the bond graph method and the model predictive control being applied to control gear shift. The desired control objectives of the model predictive controller are such that the driver does not feel the torque transitions that occur during gear shifting and that the speed change should be as smooth as possible, which also constitute the constraints in the cost function. Modeling of the vehicle has been accomplished successfully by using AVL Cruise™ software. The controller was only applied during the gear shift from third to fourth gear, and at this interval speed and torque values of the clutches were taken into account. In this study, the controller applied during a single gear shift can also be applied in all other situations, so driving comfort can be achieved during each torque transfer in the vehicle. Furthermore, control algorithm based on oncoming clutch can also be applied to the offgoing clutch. The switched model predictive control algorithms can be applied to all gear shift. Furthermore, if the control application which is effective only in the speed synchronization phase is applied in the torque handover phase in which the torque transfer between the oncoming and offgoing clutch, both the manipulated variable and the output reference tracking performance will be effective on the model predictive control. Since the sampling time of the system is fixed, the same value is used in this controller design. Different results can be obtained by changing the controller parameters such as sampling time, control horizon or prediction horizon. In addition, incorrect results in the software can be corrected by calibrating the look up tables, maps and parametrers in the current software model. Then PID and MPC applications can be reapplied by changing their own parameters. The desired results have been approached by changing the PID parameters, but the intended results have not been achieved in the MPC application. The controllers were only active in a certain part of the phase synchronization phase. In other parts of the same phase, ramps are used to approximate the clutch torque to the desired torque values. In the entire speed synchronization phase, the MPC can be applied to track target speed profile and send the necessary clutch torque to the system at that moment.

In this study, the controller was used in the case of shifting from third gear to fourth gear, in which only the power on up shift type. In this gear shift which has a torque speed sequence (TSS), only the desired torque is sent to the system for the oncoming clutch. The same controller can request torque to the system for offgoing clutch in the other gear shifts which have speed torque sequence (STS).

The model predictive control application is a subject that is being worked on in order to solve different problems encountered in the automotive sector as well as in other fields, and is open to improvement. The uncertainties arising from the complexity of the systems and the existence of nonlinear systems necessitate the development of different controller designs and applications. The model predictive control has been used in automotive sector recently by applying different methods to different parts of the vehicle.

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APPENDICES

APPENDIX A: Derivation of AT Gearbox Model

The equations are obtained from the bond graph model by using single planetary gear set and combined ravigneaux planetary gear set architectures are given below.

$$w_{R3} = \frac{P_{R3}}{I_{R3}}$$

$$w_{C2} = \frac{P_{C2}}{I_{C2}}$$

$$w_{R2} = \frac{P_{R2}}{I_{R2}}$$

$$\dot{P_{R3}} = T_{in} - T_1$$

$$\dot{P_{C2}} = T_{Cl_1} - T_{Br_2} - T_{Cl_4} - T_{R_2} - T_{S_2} - T_{S_1}$$

$$\dot{P_{R2}} = T_{Cl_4} + T_{R_2} - T_0$$

$$T_1 = T_{R_3} + T_{Cl_1}$$

$$\dot{P_{R3}} = T_{in} - T_{R_3} - T_{Cl_1}$$

$$I_{R3}\dot{w_{R3}} = T_{in} - \frac{I_{C3}\frac{\dot{w_{R3}}\, h_3}{1+h_3} + T_{Cl_3} + T_{Cl_2}}{1+\frac{1}{h_3}} - T_{Cl_1}$$

$$\begin{aligned} I_{C2}\dot{w_{C2}} = {} & T_{Cl_1} - T_{Br_2} - T_{Cl_4} - T_{Br_1}(h_2+1) - I_{S2}\dot{w_{C2}}\left({h_2}^2 + 2h_2 + 1\right) \\ & + I_{S2}\dot{w_{R2}}\left({h_2}^2 + h_2\right) + I_{S1}\dot{w_{C2}}\left(-\left(\frac{h_2}{h_1}\right)^2 + 2\left(\frac{h_2}{h_1}\right) - 1\right) \\ & + I_{S1}\dot{w_{R2}}\left(\left(\frac{h_2}{h_1}\right)^2 - \frac{h_2}{h_1}\right) + T_{Cl_3}(h_2+1) - T_{Cl_2}\left(\frac{h_2}{h_1} - 1\right) \end{aligned}$$

$$\begin{aligned} I_{R2}\dot{w_{R2}} = {} & T_{Cl_4} + T_{Br_1}h_2 + I_{S2}\dot{w_{C2}}\left({h_2}^2 + h_2\right) - I_{S2}\dot{w_{R2}}{h_2}^2 + I_{S1}\dot{w_{C2}}\left(\left(\frac{h_2}{h_1}\right)^2 - \frac{h_2}{h_1}\right) \\ & - I_{S1}\dot{w_{R2}}\left(\frac{h_2}{h_1}\right)^2 - T_{Cl_3}h_2 + \frac{h_2}{h_1}T_{Cl_2} - T_0 \end{aligned}$$

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