Control-Oriented Automatic Transmission-Based Powertrain Modeling and Simulation with Judder

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CONTROL- ORIENTED AUTOMATIC TRANSMISSION- BASED POWERTRAIN MODELING AND SIMULATION WITH JUDDER

by

Harshal B. Kundale

A thesis submitted to the Graduate College in partial fulfillment of the requirements for the degree of Master of Science in Engineering Mechanical Engineering Western Michigan University August 2018

Thesis Committee:

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This work presents automatic transmission-based powertrain modeling. The powertrain consists of an engine, torque converter with lock up clutch, transmission gearbox, propeller shaft, and vehicle body. Simplified powertrain component models are developed for vehicle powertrain dynamic response analysis and future control work. The powertrain components are modeled with algebraic and first order non-linear differential equations. A MATLAB-based powertrain simulation system is developed to investigate the transient characteristics during lock up of torque converter. Simulation results are used in the determination of effects of judder on torque and angular velocity. Clutch judder is a self-excited vibration that occurs during the clutch engagement process. It causes vibration in the drivetrain and poor driver feel. Effects on judder of throttle ramp time, final throttle opening, and clutch clamping force are investigated. A Honda CR-V is the basis for the powertrain analyzed. Results from simulation show that a negative friction gradient of clutch material causes self-excitation during engagement process and these torsional vibration causes fluctuation in torque in drivetrain. Fluctuation in torque results in bad shift quality. Final throttle opening more than 80% and clutch clamping force 1.25 times of original clutch clamping force caused judder and it affected angular speed and torque at turbine. In order to choose appropriate clutch clamping force with final throttle opening, control need to be developed. Also, coordination of engine torque control and clutch torque control is a viable strategy to improve shift quality.
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I would like to begin by acknowledging my thesis advisor Dr. Richard Meyer of College of Engineering & Applied Science at Western Michigan University. His enthusiasm for support of academic work in system modeling for control, optimal control theory and hybrid systems inspired me to work contained in this thesis. The door to Prof. Meyer office was always open whenever I ran into a trouble spot or had a question about my research or writing.

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Finally, I must express my very profound gratitude to my parents and to my brother for providing me with unfailing support and continuous encouragement throughout my years of study and through the process of researching and writing this thesis. This accomplishment would not have been possible without them. Thank you.

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It is observed that the driveline is a major source of noise and vibration inside the vehicle of all ranges from passenger cars to heavy trucks [1]. These vibrations have adverse effect in the passenger comfort. These vibrations further lead to poor drivability and smoothness. Removing these noise and vibrations from the driveline can improve passenger comfort and manufacturer’s success [2].

Driveline vibrations can happen when the vehicle is in motion at different running conditions. Rabeih [1] described different passenger car vibrations caused by the driveline system and are summarized in Table 1.1.

Table 1.1: Classification of vibration noises caused by driveline system [1]

<table>
<thead>
<tr>
<th>Frequency range (Hz)</th>
<th>Exciting force</th>
<th>Type of vibration</th>
<th>Linear/non-linear</th>
<th>Phenomenon</th>
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1.1 Engine

It is observed that crankshaft vibrations are the major excitation forces in the engine structure and driveline. Different vibrations like bending, torsional and axial along with displacement in all six degrees of freedom make crankshaft vibrations complex [2]. Mourelatos [3] used Ritz method to predict dynamic response of an engine crankshaft. Finite element method is used to present structural analysis using dynamic sub structuring. Mourelatos calculated the dynamic response of engine crankshafts. Calculations were performed to show accuracy of presented method. Mourelatos referred Figure 1.1 to represents crankshaft-engine block system.

Figure 1.1: Schematic of the crankshaft-engine block system [3]
Nagmatsu et al. [5] used reduced impedance method to analyze natural frequencies, natural modes and dynamic response of a crank shaft of an internal combustion engine. They expressed the dynamic behavior of the straight shaft part by the transfer matrix, which is transformed into the reduced impedance matrix. They combined the reduced impedance matrices of all components to obtain a global equation of motion of the crank shaft, and this equation is solved to get natural frequencies, natural modes and dynamic response. The calculated results shows good agreement with the experimental results.

Okamura et al. [7] described a crankshaft system under firing conditions with three-dimensional free and forced vibrations. They have modeled crank pin and crank journal as round bar while crank arm and counter weight as jointed structure with simple beam. Flywheel, timing gear, and front pulley are considered as a set of moment of inertia and masses. They calculated excitation forces from the gas force and inertia force due to reciprocating masses. Finally, calculated values and experimental results compared.

1.2 Torque Converter Clutch (TCC)

[Diagram of torque converter clutch]

Figure 1.2: Schematic of torque converter clutch [12]

Clutch judder/shudder is a friction-induced vibration between masses in sliding contact that occurs in clutch engagement [9-12]. When the judder appears, there are self-excited vibrations which cause fluctuation in torque and angular speed at turbine. The self-excited vibration is caused by a negative friction gradient appearing in the clutch system. Negative friction gradient leads
negative damping in the system, which further leads to self-excited vibrations in the system. Figure 1.2 shows schematic diagram of mechanical wet friction clutch.

Duan et al. [26] presented the non-linear behavior of torque converter clutch. They developed a linear system analysis procedure to determine the pure stick to stick-slip motions. They showed this procedure can efficiently and accurately identify the frequency ranges where linear or non-linear studies are needed. They observed stick-slip behavior as a result of the engine torque irregularity and nonlinear friction characteristics. Also, the effect of the friction disc inertia is studied. Their analytical and numerical results show that this inertia significantly affects the system dynamics. Also, their predictions compared well with prior measurements on a passive vibration absorber experiment.

Li et al. [31] presented systematical approach to investigate the dynamic response characteristic of the clutch engagement process. They developed the model for dynamic analysis of driveline system including friction clutch. In this model, formula to calculate friction torque is derived. Also, based on numerical solutions, the influence of the stiffness of driven disc in torsional and axial directions on the starting judder of a vehicle is analyzed. Two real vehicle models are presented. For starting judder problem of these two vehicles, their damper torsional stiffness of clutch driven disc and axial stiffness of waveform cushion are adjusted separately based on the proposed model and the optimization result, which validated the proposed model and analytical method. They concluded that increasing the torsional stiffness, moment of inertia and viscous damping of the clutch driven disc, or decreasing the axial stiffness of the waveform cushion, or increasing the static friction coefficient can all effectively reduce the judder vibration. Since, the restriction of the installation space of the clutch drive disc, the moment of inertia is difficult to adjust. Further, larger static friction coefficient always causes bigger shock effect. Thus, the torsional stiffness of the clutch driven disc and axial stiffness of the waveform cushion are relatively easy to adjust to improve the judder influence.

Kani et al. [27] have proposed that judder is significantly related to the $\mu-\nu$ characteristic (where $\mu$ is the coefficient of friction and $\nu$ is the slip speed) of an interface friction material. For a constant clamp load they have shown that the $dM/dn$ (where $M$ is the torque and $n$ is the relative revolution speed between the driving and driven discs) is directly proportional to $d\mu/d\nu$. The clamp load is exerted by the diaphragm to clamp the clutch disc between the pressure plate and the
flywheel [52]. Kani et al. [27] explained that clutch judder is dependent on the slope of the friction coefficient. To support their explanation, they performed stability analysis and the analysis is verified with numerical simulations. An algorithm for modelling stick-slip is developed and is used in numerical simulations which show that the likelihood of stick-slip is increased by clutch pressure fluctuations, judder approaching engagement, and external torque fluctuations.

Drexl [9] presented a driveline torsional model with inertia of the driveline components considered. It was found that when the model is excited at the lowest natural frequency when judder occurs in the system. Simulation results shows that negative friction gradient creates self-excited vibrations. Also, changing negative friction gradient into positive gradient provides dampening in the system. Results obtained in the study are said to be useful in the design of a clutch system with minimum judder.

Centea et al. [12] presented a model that comprises a cable operated clutch as well as driveline and vehicle body inertia components. Non-linear characteristics of clamp load during clutch engagement and stick-slip process are provided. Results shows friction materials with different \( \mu-\nu \) characteristics produces torsional vibrations in the system. Also, loss in clamp load during clutch engagement enhance tendency of judder in the system. It is observed that experimental results and simulation results are in good agreement.

Maucher [23] presented a theoretical study of the clutch judder and the effect of various parameters on the judder. Reducing friction coefficient leads to judder in the system. Also, insufficient damping, clamp load fluctuations and engine torsional irregularity causes judder. Maucher presented frictional vibration model for motor vehicle drive train. Based on theoretical studies, Maucher has revealed that frictional vibrational occurs essentially in the presence of low drive train damping values and a negative gradient of the friction coefficient.

Li et al. [50] presented effect of judder on shift quality. They mentioned judder as jitter. They developed a judder dynamic model. They established the state–space equation with controlling variables of motor torque and clutch friction torque. The torsion angle, torsion angular velocity, and shift jerk are selected as optimization targets. To determine the dynamic responses in clutch engagement process, the optimal control in different conditions of weight coefficients, initial torsion angles, and resistance torques was studied. Their results showed that the optimal control strategy could reduce the judder.
Higashimata et al. [51] discussed slip-control of lock-up clutch to avoid vibration and noise during locking up transmission at low speed. In this study, this complex non-linear system is modeled. A robust control algorithm is implemented for taking into considerations different vibrations. Two degree of freedom control system used which provides slip control at low speed. It is observed that the transmission locked up before 20km/hr and there is not an uncomfortable feeling to the driver.

1.3 Torque Converter

![Torque Converter Diagram](image)

Figure 1.3: Cross-section of a torque converter [32]

In an automatic transmission, torque converter multiplies torque generated by the engine, absorbs torsional vibration of the engine and powertrain, and smooths out the combustion pulses associated with engine rotation [32]. Figure 1.3 shows that the torque converter has three main components: the pump/impeller, the turbine, and the stator. The pump is connected to the engine shaft. The turbine is connected to the transmission input shaft. The stator is placed between the pump and the turbine. The stator redirects fluid from the turbine to the pump. The stator has one-way clutch. The torque converter also has lock-up clutch. The lock-up clutch locks the engine and transmission during higher gear ratios. During lock up clutch operations, pump and turbine shafts
are mechanically connected at higher gear ratios to improve the fuel economy and increases the efficiency.

Hrovat et al. [33] described torque converter dynamics with a set of four first-order nonlinear differential equations, along with the corresponding bond graph representation. The torque converter model and equations presented in the paper provides future validation and experiments. This model reveals underlying physical structure of the torque converter. Pohl [34] investigated the transient performance of a torque converter. The investigation included experiments in the lab as well as numerical simulations. Experiments carried out in the lab included different torque converter transient characteristics. And numerical simulations included solving a set of non-linear differential equations. Pohl created computer model that determines the transient behavior of the torque converter. For the validation of the computer model, simulation results are compared with test data for three different torque converters. It is observed that there is excellent agreement between simulation results and experimental data.

Asl [32] presented torque converter modelling and simulation. Modeling and simulation covered forward flow mode and reverse flow mode operations of the vehicle. The torque converter model is validated with the experimental results of the Honda CRV torque converter during the forward flow mode operation. The aim of this paper is to research reverse flow mode simulation, and the application of the proposed torque converter model to evaluate damping characteristics of a torque converter due to undesired disturbances generated either from engine combustion pulsations or from road bumps and potholes. The simulation results show that a torque converter damps high frequency disturbances introduced from the engine shaft to the transmission side and vice versa.
Gearbox is another most common source of vibration of driveline systems. Daly and Smith [14] determined the transmission error of gears to identify the noise and vibration in drivelines. They looked at the gear set in Figure 1.4. They attached circular gratings at input and output of the gear drive and then turned generated photocell signals into pulse trains. Authors showed results from gearbox and back axle measurements and discussed advantages and limitations of this measurement. Their results are useful to determine noise and vibration of gear systems.

Feki et al. [28] described the complex vibrations generated by a two-stage gearbox. To identify frequency of vibration signals from two-stage gearbox, two methodologies are discussed: lumped-parameter modeling and phenomenological modeling. In phenomenological model, vibrations are measured by sensor. This sensor is fixed outside fixed ring gear with respect to inertial reference frame. In lumped parameter modeling, results from lumped parameter model are referenced with respect to a rotating frame and then transferred into an inertial reference frame. Inertial reference frame is fixed to the carrier plate and vibrations in all degrees of freedom are referenced to inertial reference frame. Gear mesh frequencies and carrier rotation frequencies are observed. It is observed that vibration behavior and related spectrum for two-stage gearbox shows close agreement for the two models described in the study.

Brosey et al. [25] showed the detrimental effect of high effective transmission inertia upon gear rattle and shift effort. A unique test stand which can evaluate the gear rattle tendency of a
transmission is described. They presented a computer model to calculate the angular acceleration at the transmission input given the spring rate, damping, and inertia of the driveline components. An optimum arrangement of the transmission gear train is described. Their results showed that this arrangement lowers the effective inertia of the transmission, thereby decreasing the tendency for gear rattle and reducing shift effort.

Fudala et al. [15] studied the vibration level of the system and gear rattle with respect to system component inertias, spring rates, and damping effective drive, coast and neutral gear rattle. They stated that, if torsional vibration at gear/spline affects oil drag effect and inertia then gear rattle occurs. The effect of gear rattle increases with an increase in torsional vibration at gear location. In addition, clutch stiffness and damping effect minimizes torsional vibration, which further minimizes gear rattle.

1.5 Propeller Shaft

![Driveline layout](image)

Figure 1.5: Driveline layout of vehicle [29]

The propeller shaft along with its joints are a source of driveline vibration. Kato et al. [16] analyzed the possibility that a flexible rotating shaft driven by a universal joint will make lateral-torsional coupled vibrations. They considered a rotating shaft has a uniform circular section and is supported by a universal joint at one end and ball bearing at the other end. Their analysis revealed the stability of the driven shaft and showed that unstable vibrations occurs when the drive speed nearly coincides with the arithmetic mean of two natural angular frequencies about bending and torsion.
Xia et al. [29] presented control method for the system torsional resonance. A RWD driveline dynamic model considered in the study is shown in Figure 1.5. The natural frequencies and modal shapes are calculated. Torsional vibration responses determined based on forced vibration analysis. Analysis of system sensitivity and DOE are based on the parameterized stiffness, inertia and damping. The results obtained for second and third order modal shows that the transmission shaft possesses the maximum amplitudes and its corresponding modal frequencies vary with different gear position. Also torsional vibrations reduce with increasing propeller shaft stiffness, increasing input shaft inertia, increasing clutch damping, increasing half shaft stiffness and reducing clutch stiffness. The results from DOE analysis shows that propeller shaft stiffness, clutch stiffness, transmission input shaft inertia and the inertia of the axle pinion shaft are important factor for reducing torsional vibrations of the transmission gear shafts. It is observed that, there is consistency between analysis results and test results.

Otake et al. [18] presented a simulation from the flywheel to the differential pinion. They measured backlash and spring characteristics of the spline. They found that there are two stages of spring stiffness. In first stage spring rate lowered and in second stage spring rate increased. They stated that these fluctuations at Hooke’s joint determines gear rattle in final gear at differential.
Axle pitch/tramp is a rotary oscillation of rigid axles about the axis of rotation parallel to the longitudinal axis of the vehicle [1]. Sharp [19] described axle pitch with an analogue computer simulation of vibrations occurring with automobile rear axles under braking and accelerating conditions. He demonstrated these vibrations are self-excited and existence of limit cycles. Then the author described the mode shapes corresponding to these limit cycles and also deduced the mechanisms through which the self-excitation is possible. He concluded that axle tramp can be eliminated by suitable attention to the design of the axle to body mounting. The axle pitching vibrations occurring with vehicle rear axles under braking and accelerating conditions are shown in Figure 1.6.

Zargartalebi et al. [30] derived a mathematical model for prediction of the pitch/tramp. Sport car has considered to research axle pitch behavior. A dynamic model considered to reveal different dynamic behavior of the system. Their study shows that, torque from the engine, moment of inertia of moving parts, wheel weight and suspension materials are important parameters to control the axle pitch. It is seen that with increase in engine torque, axle pitch critical region increases.

Sharp [20] presented vehicle rear axle model and studied axle pitch under accelerating and braking condition. This model consisted of an engine gearbox with an engine mounting which was restrained with longitudinal oscillations of engine gearbox. The author identified that axle
mounting stiffness has a major influence on stability and concluded that there needs to be more concentration on design of axle mounting to remove axle pitch from the system

1.7 Summary of Literature Survey

It is observed that many of the torsional vibrations problems and refinement in drivelines have been solved through simple, lumped mass models combined with experimental measurements. Multi body system dynamics codes are developed for more sophisticated models to assist and understanding in ensuring high levels of refinement. However, some problems remain. More understanding of the driveline system behavior during clutch engagement is still required. There are very few publications and literature available which covers behavior of the driveline system under the transient operating condition. Therefore, more attention to the effects of transient behavior is needed. Many researchers have investigated clutch judder by simulations and experiments. However, there is lack of control-oriented powertrain model that is capable of capturing clutch judder and its effects.

1.8 Thesis Overview

This thesis work presents control-oriented modeling of the powertrain containing engine, torque converter, torque converter clutch, transmission gear box, propeller shaft, and vehicle body. This model is used to investigate clutch judder during torque converter clutch (TCC) engagement via simulation. The relevant modeling of the powertrain is given in chapter 2. Chapter 3 describes the simulation setup and constant used to solve the first order differential and algebraic equations. Simulation results are summarized in chapter 4 to determine effect of clutch judder on torque and angular speed at pump and turbine, difference in angular input and output speed of propeller shaft and vehicle velocity. Finally, chapter 5 summarizes the conclusion, recommendations for future research and improvements.
CHAPTER 2

MODELING OF POWERTRAIN COMPONENTS

2.1 Introduction

The objective of this study is to develop control-oriented automatic transmission powertrain model and analysis of clutch judder which affects shift quality and driver’s feel. Engine, torque converter, torque converter clutch, transmission gearbox, propeller shaft, and vehicle body are modelled in this section in preparation for the study of judder a friction-induced vibration between masses in sliding contact and occurs in clutch engagement [18]. The main components of the powertrain are shown in Figure 2.1

![Powertrain block diagram](image)

Figure 2.1: Powertrain block diagram

2.2 Engine Modeling

The engine is characterized by torque and power curves as a function of speed. In this work, fuel consumption is not considered. The engine power output dynamics are [35]:

$$\frac{dP_{ice}}{dt} = \frac{1}{T_{ice}} \left[ -P_{ice} + P_{ice,max}(\omega_{ice})u_{ice} \right]$$  (1)

where $\omega_{ice}$ is engine’s angular velocity, $P_{ice}$ is the engine mechanical power at the flywheel output, $T_{ice}$ is 0.26 s average time constant for engine power delivery accounting for combustion delay and crankshaft and flywheel inertias [35],[36],[37]. $P_{ice,max(\omega_{ice})}$ is the maximum available engine power modulated by $u_{ice}$ in [0,1]. The engine maximum output power versus speed map is shown in Figure 2.2 xbar represents $\bar{\omega}_{ice}$ and ybar represents $\bar{P}_{ice,max}$

$$\bar{\omega}_{ice} = \frac{\omega_{ice}}{\max(\omega_{ice})} \quad \text{and} \quad \bar{P}_{ice,max} = \frac{P_{ice,max(\omega_{ice})}}{\max(P_{ice})}.$$
A third order polynomial is used to describe the maximum normalized power values:

\[ \bar{P}_{ice}^{\text{max}} = p_1 \bar{\omega}_{ice}^3 + p_2 \bar{\omega}_{ice}^2 + p_3 \bar{\omega}_{ice} + p_4 \]  

(2)

2.3 Torque Converter

Torque converter is type of fluid coupling in an automatic transmission, which connects the engine to the transmission gearbox. The torque converter connects and disconnects engine to the transmission based on vehicle to run and stopped [38].

A torque converter consists of three main parts as shown in Figure 2.3. These parts are pump, stator and turbine. The pump is connected to the engine, and the turbine is connected to the transmission gearbox. Stator is attached to the transmission housing through a one-way clutch. The pump and the turbine act as a centrifugal pump. Torque converter is filled with fluid. When the pump turns faster than the turbine, oil flows from pump to turbine because pumping action at the pump is more than that of the turbine.
When the pump starts rotating, due to centrifugal acceleration, fluid entered in the pump travels radially outwards and attains angular momentum. This requires engine torque inputs. Then fluid leaves the pump and enters the slower turning turbine and is then forced to travel radially inward reducing angular momentum. This phenomenon causes torque upon the turbine to drive the load [32]. When fluid leaves the turbine, it enters the stator. If the stator blade is stationary, the blade will bend the fluid flow in the same direction as the rotation of the pump. This process reduces the angular momentum gain required from the pump. In this case, the turbine torque is greater than that of the pump. Also torque multiplication is obtained [32].

At coupling point, the turbine achieved approximately same speed of the pump. The torque multiplication becomes zero and the torque converter act as a simple fluid coupling. At the coupling point a lock up clutch locks the turbine to the pump of the converter. Thus the turbine and pump have the same speed and torque. Lock up clutch engages only when coupling point is achieved. During coupling, the stator also starts to rotate in the direction of the pump and turbine rotation [40].

However, if negative torque to be transferred from the housing to the stator through the one-way clutch, then the one-way clutch will overrun. In this case the stator will run at a speed such that no angular momentum is contributed to the oil element. Thus, the pump and turbine torques are equals and operating condition is called “above coupling point” [32].
Operation of the torque converter is also possible with turbine speed greater than pump speed. In this case, the turbine drives the pump, the stator overruns, and the pump and turbine torques are equal. This operating condition is called “engine braking or coast operation” [32].

The power available at engine flywheel is transferred directly torque converter pump.

\[ P_{ice} = P_{pump} \] (3)

Torque converters can be modeled using a moment of momentum and fluid inertia approach [32]. The resulting differential equations are suitable to show the transient effect of the torque converter and to evaluate the effects of the torque converter’s parameters on the torque converter’s performance. The model below from [32] is developed by considering all pump, turbine, and stator dynamics:

\[ I_p w_p + \rho S_p \dot{Q} = -\rho \left(w_p R_p^2 + \frac{R_p Q}{A} \tan a_p - w_s R_s^2 - \frac{R_s Q}{A} \tan a_s \right) Q + T_p \] (4)

where, \( \rho \) is fluid density, \( A \) is flow area, \( R_p \) is pump radius, \( I_p \) is moment of inertia of pump, \( a_p \) is pump exit angle, \( S_p \) is pump design constant, \( Q \) is volumetric flow, \( w_p \) is pump angular velocity, \( w_s \) is stator angular velocity, \( R_s \) is stator radius, \( a_s \) is stator exit angle and \( T_p \) is pump torque.

Equation (4) represents a first order differential equation for pump angular momentum:

\[ I_t \dot{w}_t + \rho S_t \dot{Q} = -\rho \left(w_t R_t^2 + \frac{R_t Q}{A} \tan a_t - w_p R_p^2 - \frac{R_p Q}{A} \tan a_p \right) Q + T_t \] (5)

Equation (5) represent first order differential equation for turbine angular momentum, where \( a_t \) is turbine exit angle, \( I_t \) is moment of inertia of turbine, \( S_t \) is turbine design constant.

\[ I_s \dot{w}_s + \rho S_s \dot{Q} = -\rho \left(w_s R_s^2 + \frac{R_s Q}{A} \tan a_s - w_t R_t^2 - \frac{R_t Q}{A} \tan a_t \right) Q + T_s \] (6)

Equation (6) represent first order differential equation for stator angular momentum, where \( R_s \) is stator radius, \( a_s \) is stator exit angle, \( S_s \) is stator design constant, \( I_p \) is moment of inertia of stator.

\[
\rho \left(S_p \dot{w}_p + S_t \dot{w}_t + S_s \dot{w}_s \right) + \frac{\rho L}{A} Q = \rho \left(R_p^2 w_p^2 + R_t^2 w_t^2 + R_s^2 w_s^2 - R_p^2 w_p w_s - R_p^2 w_t w_p - R_t^2 w_s w_t \right) + \frac{w_p Q}{A} \left(R_p \tan a_p - R_s \tan a_s \right) + \frac{w_t Q}{A} \rho \left(R_t \tan a_t - R_s \tan a_p \right) + \frac{w_s Q}{A} \rho \left(R_s \tan a_s - R_t \tan a_t \right) - P_L
\] (7)
Equation (7) represent conservation of energy with including power loss $P_L$, where, $L_f$ is fluid inertia length, $C_{sh}$ is shock loss coefficient, $C_f$ is frictional loss coefficient.

There are two contributors to the power loss $P_L$, modeled here. i.e. the flow losses $P_{FL}$ and the shock losses $P_{SL}$. The flow losses are due to shear stresses in the boundary layer and a contribution due to possible pressure drag. The shock losses are due to non-ideal speed conditions at the interface between any two torque converter elements [3]. The flow loss is given by following expression [32],

$$P_{FL} = \frac{1}{2} \rho \text{abs}(Q)(f_p V_p^*{}^2 + f_t V_t^*{}^2 + f_s V_s^*{}^2)$$

(8)

where, $V_p^*, V_t^*, and V_s^*$ are fluid velocities relative to pump, turbine and stator blades, $f_p$, $f_t$, $f_s$ is given fluid friction factors.

$$V^* = Vi_r + V \tan \alpha \phi$$

$V_p^*, V_t^*, V_s^*$ Obtained by substituting the exit angles $\alpha_p, \alpha_t, \alpha_s$, $r$ is radius leading to fluid particle, $i_r$ and $i_\phi$ are unit vectors corresponding to polar coordinates in the radial and tangential direction, respectively.

The shock losses are determined by following expression [32],

$$P_{SL} = \frac{1}{2} \rho \text{abs}(Q)(C_p V_p^2 + C_t V_t^2 + C_s V_s^2)$$

(9)

where, $C_p$, $C_t$, $C_s$ are shock loss coefficients ideally equally one. Shock velocities are given by

$$V_p = R_s (w_s - w_p) + \frac{Q}{A} (\tan \alpha_s - \tan \alpha_p)$$

(10)

$$V_t = R_p (w_p - w_t) + \frac{Q}{A} (\tan \alpha_p - \tan \alpha_t)$$

(11)

$$V_s = R_t (w_t - w_s) + \frac{Q}{A} (\tan \alpha_t - \tan \alpha_s)$$

(12)

where, $\alpha_s, \alpha_p, \alpha_t$ are blade angles relative to plane A as shown in Figure 2.4 at stator, pump, and turbine.
Herein, the steady state condition is applied to Equations (4) - (7) while considering torque convertor’s forward operations below coupling point. Below coupling point, the stator’s rotational speed \(w_s\) is zero and

\[
0 = -\rho \left( w_p R_p^2 + \frac{R_p Q}{A} \tan \alpha_p - w_s R_s^2 - \frac{R_s Q}{A} \tan \alpha_s \right) Q + T_p \tag{14}
\]

\[
0 = -\rho \left( w_t R_t^2 + \frac{R_t^2 Q}{A} \tan \alpha_t - w_p R_p^2 - \frac{R_p Q}{A} \tan \alpha_p \right) Q + T_t \tag{15}
\]

\[
0 = -\rho \left( w_s R_s^2 + \frac{R_s Q}{A} \tan \alpha_s - w_t R_t^2 - \frac{R_t^2 Q}{A} \tan \alpha_t \right) Q + T_s \tag{16}
\]

The available output turbine torque \(T_t\) is transferred to the transmission gearbox as input torque for different gearshift without any losses.
2.4 Torque Converter Clutch [TCC]

The torque converter clutch works on two principles. One is torque converter clutch with fully lock-up control. In this type of operating mode, a sufficiently high pressure is applied to ensure a no slip condition. As pressure is applied to the pressure plate, engine speed will decrease until it matches exactly turbine speed once the clutch capacity matches engine torque. All engine torque is then carried by the clutch and transferred to the transmission without loss. The second is to keep the clutch open when the torque converter is the power transfer device between the engine and gearbox.

The following three operating states are considered for the torque converter and torque converter lock up clutch.

2.4.1 Open Torque Converter

For open torque converter, turbine speed is less than pump speed. This torque converter’s forward operation can be explained by using the ‘coupling point’ that is based on stator’s torque. This phase of torque converter is called ‘torque multiplication range’ below coupling point where the stator is fixed by the one-way clutch and its rotational speed is zero. The pump shaft is the driving shaft and the turbine shaft is the driven shaft. In this mode of operation, the property of the fluid coupling helps to multiply the engine’s torque to accelerate the vehicle.

\[ w_t < 0.85w_p \]

2.4.2 Pre-lockup Torque Converter

In this study, pre-lock up condition considered when turbine speed is equal to or greater than 85% of pump speed and less than 99% of pump speed. Lock-up clutch engagement occurs gradually, bring the driveline through friction disc and crankshaft through the flywheel and pressure plate to the same rotational speed. During engagement process, on the friction surfaces of the clutch the friction torque acts as an engaging force for the driveline. A part of the energy transmitted through the driveline is transformed into other forms of energy by positive damping effects. If damping becomes negative, a part of the energy transmitted by the clutch could induce self-excited torsional vibrations of the driveline, contribute to judder [42].

In this study, pre-lock up condition is considered between,

\[ w_t \geq 0.85w_p \text{ and } w_t < 0.99w_p \]
To obtain dynamic response from clutch driven disc, relative angular velocity $\omega_{SL}$ considered. The friction coefficient ($\mu$) at sliding state is not a constant and it changes with the relative angular velocity $\omega_{SL} = \omega_p - \omega_t$ [49]. To obtain dynamic response of the driven system of clutch, the $\omega_{SL}$ cannot exactly equal zero in the numerical solution, which may lead to incorrect calculation because of incorrect judgment on the clutch state [31]. To solve this problem, the following condition is considered:

$$\omega_{SL} \leq 0.15$$

The coefficient of static friction ($\mu_0$) between the clutch plates and the friction facing is very crucial for the clutch performance. It determines the capability of torque transmission of a clutch as well as torsional vibration of a driveline. If the value of friction coefficient gradient ($\mu'$) is negative, the self-excited vibration occurs. The gradient of friction coefficient in this study is set to -0.0001s/rad [31]. There are many friction models to describe the relationship between the sliding friction coefficient ($\mu$) and the relative angular velocity($\omega_{SL}$), such as the Strubeck friction model. These models are usually non-linear. In this research, the following model [12] is used.

$$\mu = \mu_0 + \mu'(\omega_{SL})$$  \hspace{1cm} (18)

During the clutch engagement process, the friction torque ($T_{cl}$) between driving and driven disc will be changed in the slip states. In the slip state, the relative angular velocity, and the friction torque ($T_{cl}$) can be given by [31],

$$T_{cl} = F_c \mu R_{mc} \text{abs}(\omega_{SL})$$  \hspace{1cm} (19)

where, $F_c$ is clutch clamping force, $R_{mc}$ is the clutch mean radius and it is determined by the clutch outside radius $R_o$ and inside radius $R_i$ as [31],

$$R_{mc} = \frac{2(R_o^3 - R_i^3)}{3(R_o^2 - R_i^2)}$$  \hspace{1cm} (20)

Also, turbine torque is equal to clutch friction torque ($T_{cl}$).

$$T_t = T_{cl}$$  \hspace{1cm} (21)
2.4.3 Lock-up Torque Converter

During this mode of operation, clutch is full engaged. Speed of pump and turbine are equal. No torque multiplication takes place and available engine speed and torque transmitted to driveline through lock-up clutch.

\[ w_t = w_p \quad (22) \]
\[ T_t = \frac{P_{Ice}}{w_p} = T_p \quad (23) \]

2.5 Automatic Transmission

The automatic transmission converts input speed and torque from the engine, via the torque converter, to that required at the propeller shaft input to move the vehicle; this allows the engine to operate in its relatively narrow range of speeds while providing a wide range of output speeds. The automatic transmission uses gears to make more effective use of the engine's torque, and to keep the engine operating at an appropriate speed [43]. The planetary gear set is the device that makes this possible in an automatic transmission.

The planetary gears are an essential component of automatic transmission. In this study, a Ravigneaux type planetary gear set is used. The dynamic equations for the Ravigneaux type automatic transmission are mathematically derived. A schematic diagram of the Ravigneaux gear set is shown in Figure 2.6:

Figure 2.5: Ravigneaux gear set [44]
The Ravigneaux set has two sun gears, a large sun and a small sun, and a single planet carrier, holding two sets of planetary gears, inner planets and outer planets [43][45]. The carrier is one sub-assembly with inner and outer planet gears located on the same radial line. The two sets of planet gears rotate independently of the carrier (co-rotate) with a fixed gear ratio with respect to each other. The inner planets couple with the small sun gear and co-rotate at a fixed gear ratio with respect to it. The outer planets couple with the large sun gear and co-rotates with a fixed gear ratio with respect to it. Finally, the ring gear also couples and co-rotates with the outer planets in a fixed gear ratio with respect to them.

A 4-speed automatic transmission with Ravigneux gear set is considered that is patterned after that in the 2004 Honda CRV. The 4-speed automatic transmission consist of one Ravigneaux gear set and five clutches. The follower shaft connects to the ring gear of Ravigneaux set. Three of the forward clutches determine the gears that the base shaft connects to for power transfer.

2.5.1 Different Shift Modeling

Equations (18)-(29) below describe the relationships for torque and speed in specific gears. The equations are for a 4-speed transmission similar to that previously described.

2.5.1.1 Torque Equation

The torque at ring gear, which is transferred to propeller shaft, is equal to the multiplication of input torque available at input gear and the respective gear ratio.

\[ T_{s1} = T_t \]  

\[ \frac{T_{s2}}{s_2} + \frac{T_{s1}}{s_1} + \frac{T_c}{c_1} = T_r \left[ \frac{1}{r} - \frac{2c_2}{c_1} \right] \]  

where, \( T_{s1} \) is torque at input sun gear, \( T_{s2} \) is torque at reaction sun gear, \( T_c \) is torque at carrier, \( T_r \) is torque at ring gear, and \( s_1, s_2, c_1, r \) are radii of input sun gear, reaction sun gear, carrier and ring gear, respectively.

In the planetary gear set, if there is no power transfer from carrier, then speed of carrier is considered as zero and equations for different shifts are considered as ‘without carrier (\( \omega_c = 0 \)):’

\[ \omega_{s1}s_1 = \omega_{p1}p_1 \]  

\[ \omega_{p1}p_1 = \omega_{p2}p_2 \]  

\[ \omega_{p2}p_2 = \omega_r r \]
\[
\begin{align*}
\omega_s s_2 &= \omega_p p_2 \quad (29) \\
\omega_s s_1 &= \omega_r r \quad (30) \\
\omega_s s_2 &= \omega_r r \quad (31) \\
\omega_{s1} s_1 &= \omega_{s2} s_2 \quad (32)
\end{align*}
\]

where, \(\omega_c\) is carrier speed, \(\omega_{p1}\) and \(\omega_{p2}\) are planet gear 1 and planet gear 2 speed, \(\omega_{s1}\) is input sun gear speed, \(\omega_{s2}\) is reaction sun gear speed, \(\omega_r\) is ring gear speed, \(S_1\) is input sun gear radius, \(S_2\) is reaction sun gear radius, \(r\) is ring gear radius. When the carrier speed is nonzero, then

\[
\begin{align*}
\omega_{s1} s_1 &= \omega_c c_1 - \omega_{p1} p_1 \quad (33) \\
\omega_c c_1 + \omega_{p1} p_1 &= \omega_c c_2 - \omega_{p2} p_2 \quad (34) \\
\omega_{s2} s_2 &= \omega_c c_2 - \omega_{p2} p_2 \quad (35) \\
\omega_c c_2 + \omega_{p2} p_2 &= -\omega_r r \quad (36)
\end{align*}
\]

For each selection, Equations (24) - (36) are elaborated on in the following sections.

2.5.2 First Gear Steady State Equation

Figure 2.6 shows the transmission in the first gear configuration. During the first gear, The UD clutch (under drive clutch) is fully engaged and in lock up condition. The UD clutch connects turbine with input sun gear, the speed of both input sun gear and turbine are same. The L/R clutch (Low/Reverse) is also in lock-up condition. As L/R clutch hold carrier, speed of carrier is zero; 2\textsuperscript{nd} clutch and OD clutch are open.

![Figure 2.6: First gear connection](image-url)
The following are first gear operating conditions:

\[ \omega_{p1} = \omega_{p2} = 0 \]  
\[ \omega_{s1} s_1 = \omega_r r \]  
\[ \omega_{s2} s_2 = \omega_r r \]  
\[ \omega_{s1} s_1 = \omega_{s2} s_2 \]  
\[ T_r \left[ \frac{1}{r} - \frac{2c_2}{c_1} \right] - \left[ \frac{T_{s_2}}{s_2} \right] - \frac{T_{s_1}}{s_1} = 0 \]

2.5.3 Second Gear Steady State Equation

Figure 2.7 shows second gear arrangements. During second gear, UD is fully engaged and so input sun gear is equal to turbine speed. The 2nd clutch is now fully engaged so that reaction sun gear is not moving. L/R clutch and OD clutch are open.

Therefore, following conditions are satisfied for second gear:

\[ \omega_{s2} = 0 \]  
\[ \omega_{s1} s_1 = \omega_{p1} p_1 \]  
\[ \omega_{p1} p_1 = \omega_{p2} p_2 \]  
\[ \omega_{p2} p_2 = \omega_r r \]  
\[ \omega_{s1} s_1 = \omega_r r \]  
\[ \omega_{s1} s_1 = \omega_c c_1 - \omega_{p1} p_1 \]  
\[ \omega_c c_1 + \omega_{p1} p_1 = \omega_c c_2 - \omega_{p2} p_2 \]  
\[ \omega_c c_2 + \omega_{p2} p_2 = -\omega_r r \]
\[ T_r \left[ \frac{1}{r} - \frac{2c_2}{c_1} \right] - \frac{T_{s1}}{s_1} - \frac{T_c}{c_1} = 0 \] (51)

2.5.4 Third Gear Steady State Equation

Figure 2.8 shows third gear arrangements. During third gear, both UD clutch and OD clutch are fully engaged. So, input sun gear speed and the carrier speed are the same as the turbine speed. The 2\textsuperscript{nd} clutch is not engaged. 2\textsuperscript{nd} clutch and L/R clutch are open.

Thus, the following conditions are satisfied in third gear:

\[ \omega_{s2} = \omega_{p1} = \omega_{p2} = \omega_c = 0 \] (52)

\[ \omega_{s1}s_1 = \omega_r r \] (53)

\[ T_r \left[ \frac{1}{r} - \frac{2c_2}{c_1} \right] - \frac{T_{s1}}{s_1} = 0 \] (54)

2.5.5 Fourth Gear Steady State Equation

Figure 2.9 shows fourth gear arrangements. During fourth gear, the OD clutch and 2\textsuperscript{nd} clutch is fully engaged resulting in the speed of carrier being the same as the turbine speed. Also, the reaction sun gear is not moving. The UD clutch is not engaged and no torque dynamics acting on input sun gear. UD clutch and L/R clutch are open.
Figure 2.9: Fourth gear connection

Thus, following conditions are satisfied for fourth gear:

\[
\begin{align*}
\omega_{s2} &= 0 \\
\omega_t &= \omega_c \\
\omega_{s1}s_1 &= \omega_{p1}p_1 \\
\omega_{p1}p_1 &= \omega_{p2}p_2 \\
\omega_{p2}p_2 &= \omega_r r \\
\omega_{s1}s_1 &= \omega_r r \\
\omega_{s1}s_1 &= \omega_c c_1 - \omega_{p1}p_1 \\
\omega_c c_1 + \omega_{p1}p_1 &= \omega_c c_2 - \omega_{p2}p_2 \\
\omega_c c_2 + \omega_{p2}p_2 &= -\omega_r r \\
Tr\left[ \frac{1}{r} - \frac{2c_2}{C_1} \right] - \left[ \frac{Ts_2}{S_2} \right] - \left[ \frac{Ts_1}{S_1} \right] - \left[ \frac{7c}{C_1} \right] &= 0
\end{align*}
\]
2.6 Propeller Shaft

The propeller shaft connects the transmission output to the differential. The shaft consists of a flexible material like steel that twists in a response to an applied torque [46]. The twisting action delays power transmission between the shaft ends, changing the dynamic response of the driveline system. It is assumed, the shaft twists but does not bend. The model contains a finite number, \( N \), of lumped inertia elements connected in series by parallel spring-damper, plus a final inertia [47]. The result is a series of \( N+1 \) inertias connected by \( N \) rotational springs and \( N \) rotational dampers. Figure 2.10 shows propeller shaft with four elements plus a final inertia.

![Figure 2.10: Propeller shaft block diagram [47]](image)

First order non-linear differential equations are used to calculate input and output angular difference of propeller shaft. The input of the propeller shaft is connected to the transmission output shaft while the output of the propeller shaft is connected to the differential.

The shaft stiffness and inertia are computed from the shaft dimensions and material properties by the following relationships:

\[
J = \frac{n}{32} (D_o^4 - D_i^4) \quad (65)
\]

\[
V = \frac{n}{4} (D_o^2 - D_i^2) (1) \quad (66)
\]

\[
M = V \cdot \rho_p \quad (67)
\]

\[
I = (0.5)M(R_o^2 + R_i^2) \quad (68)
\]
where, \( J \) is polar moment of inertia, \( V \) is volume of shaft, \( M \) is mass of shaft, \( I \) is moment of inertia, \( \rho_p \) is density of shaft. Hoop moment of inertia is considered with very small wall thickness. Stiffness of propeller shaft \( K_p \) is given by,

\[
K_p = \frac{G J}{l}
\]  

(69)

The critical damping \( C_p \) is

\[
C = 2\xi (KI)^{0.5}
\]  

(70)

where \( \xi \) is a damping ratio, \( C_p \) is damping

In this study, two element model of propeller shaft is considered. The input of the propeller shaft is connected to transmission output shaft while output of the propeller shaft is connected to the differential.

The torque at input of the propeller shaft is equal to multiplication of torque at the turbine and gear ratio.

\[
T_{pin} = T_t R
\]  

(71)

Torque at transmission output shaft is then transferred to which is input of the propeller shaft and given by,

\[
\left(\frac{I_p}{2}\right) \frac{d\omega_{pin}}{dt} = k_p (-\Delta \theta_p) - C_p (\omega_{pin} - \omega_{pout}) + T_{pin}
\]  

(72)

where \( \Delta \theta_p = \theta_{pin} - \theta_{pout} \) and \( \frac{d\Delta \theta_p}{dt} = \omega_{pin} - \omega_{pout} \).

Torque at output of the propeller shaft is calculated by following equation,

\[
\left(\frac{I_p}{2}\right) \frac{d\omega_{pout}}{dt} = k_p (\Delta \theta_p) + C_p (\omega_{pin} - \omega_{pout}) - T_{pout}
\]  

(73)
2.7 Vehicle Body Dynamics

The vehicle dynamics describe the longitudinal motion of the vehicle given some torque from the propeller shaft. Figure 2.11 shows various forces acting on a vehicle:

![Figure 2.11: Forces acting on vehicle body](image)

The forces opposing vehicle motion are

\[ F_{\text{aero}} = -\frac{1}{2} \rho_a C_d A v^2 \]  \hspace{1cm} (74)
\[ F_{\text{roll}} = -n_r m g \cos(\theta) \]  \hspace{1cm} (75)
\[ F_{\text{grad}} = -m g \sin(\theta) \]  \hspace{1cm} (76)

where, \( F_{\text{aero}} \) is aerodynamic force, \( F_{\text{roll}} \) is rolling force, \( F_{\text{grad}} \) is gradient force, \( \rho_a \) is ambient air density, \( C_d \) is aerodynamic drag coefficient, \( A \) is vehicle frontal area, \( v \) is vehicle speed, \( n_r \) is rolling friction coefficient, \( m \) is mass of vehicle, \( g \) is acceleration due to gravity, and \( \cos(\theta) \) is angle of inclination.

Vehicle movement is given by,

\[ \frac{dv}{dt} = \frac{1}{m} \left( F_{\text{aero}}(v) + F_{\text{roll}}(v, \theta) + F_{\text{grad}}(\theta) \right) + \frac{1}{m} T_v \]  \hspace{1cm} (77)

where, \( T_v \) is calculated from \( T_{pout} \) as given by,
where $FR$ is final drive ratio.

2.8 Summary

Chapter 2, describes first order non-linear differential equations and algebraic equation. These equations are used in MATLAB for simulation. The constants used in the equations are given in chapter 3. MATLAB ODE23 solver and FSOLVE are used to solve above equations. Results and MATLAB simulation are discussed in chapter 4.
CHAPTER 3

SIMULATION SETUP

3.1 Introduction

A 2004 Honda CR-V RWD model is considered here as the example powertrain. The data parameters for the torque converter, propeller shaft and gear ratios for transmission gearbox was available for Honda CR-V in the literature. Components of powertrain considered are the engine, torque converter, automatic transmission gearbox, propeller shaft, and vehicle body. After the various model parameters are defined, the solution algorithm is presented.

3.2 Model Parameters

The following subsections describe the model parameters used in the study.

3.2.1 Engine

Engine map with power versus angular speed is generated from Honda CR-V data [53]. MATLAB curve fitting toolbox is used to get third order polynomial fit for engine maximum power versus speed mapping. Fit parameters are summarized in Table 3.1.

<table>
<thead>
<tr>
<th>Sr. No.</th>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>max($\omega_{ice}$)</td>
<td>6400 rpm</td>
</tr>
<tr>
<td>2</td>
<td>max($P_{ice}$)</td>
<td>119.019kw</td>
</tr>
<tr>
<td>3</td>
<td>$T_{ice}$</td>
<td>0.26s</td>
</tr>
<tr>
<td>4</td>
<td>$p_1$</td>
<td>-1.605</td>
</tr>
<tr>
<td>5</td>
<td>$p_2$</td>
<td>1.781</td>
</tr>
<tr>
<td>6</td>
<td>$p_3$</td>
<td>0.8671</td>
</tr>
<tr>
<td>7</td>
<td>$p_4$</td>
<td>-0.07342</td>
</tr>
<tr>
<td>8</td>
<td>Coefficient of determination ($R^2$)</td>
<td>0.9994</td>
</tr>
</tbody>
</table>
3.2.2 Torque Converter

For the torque converter modeling following constant values used. The constant values considered for torque converter in this section 3.2.2, represents for Honda CR-V model and these values became available from available literature for Honda CR-V [32-34].

Table 3.2: Torque converter constants for Honda CR-V [32-34]

<table>
<thead>
<tr>
<th>Name</th>
<th>Value</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>Pump fluid friction factor</td>
<td>0.197</td>
<td></td>
</tr>
<tr>
<td>Turbine fluid friction factor</td>
<td>0.197</td>
<td></td>
</tr>
<tr>
<td>Stator fluid friction factor</td>
<td>0.197</td>
<td></td>
</tr>
<tr>
<td>Pump shock loss coefficient</td>
<td>1.011</td>
<td></td>
</tr>
<tr>
<td>Turbine shock loss coefficient</td>
<td>1.8</td>
<td></td>
</tr>
<tr>
<td>Stator shock loss coefficient</td>
<td>0.773</td>
<td></td>
</tr>
<tr>
<td>Flow area</td>
<td>0.0107</td>
<td>m²</td>
</tr>
<tr>
<td>Fluid density</td>
<td>840</td>
<td>kg/m³</td>
</tr>
<tr>
<td>Pump radius</td>
<td>0.0991</td>
<td>M</td>
</tr>
<tr>
<td>Turbine radius</td>
<td>0.0735</td>
<td>M</td>
</tr>
<tr>
<td>Stator radius</td>
<td>0.0665</td>
<td>M</td>
</tr>
<tr>
<td>Pump exit angle</td>
<td>16.21</td>
<td>Degree</td>
</tr>
<tr>
<td>Turbine exit angle</td>
<td>-53.14</td>
<td>Degree</td>
</tr>
<tr>
<td>Stator exit angle</td>
<td>55.62</td>
<td>Rad</td>
</tr>
<tr>
<td>Pump design constant</td>
<td>-0.001</td>
<td>m²</td>
</tr>
<tr>
<td>Turbine design constant</td>
<td>-0.00002</td>
<td>m²</td>
</tr>
<tr>
<td>Stator design constant</td>
<td>0.002</td>
<td>m²</td>
</tr>
<tr>
<td>Pump inertia</td>
<td>0.092</td>
<td>kg/m²</td>
</tr>
<tr>
<td>Turbine inertia</td>
<td>0.026</td>
<td>kg/m²</td>
</tr>
<tr>
<td>Stator inertia</td>
<td>0.012</td>
<td>kg/m²</td>
</tr>
<tr>
<td>Fluid inertia length</td>
<td>0.28</td>
<td>M</td>
</tr>
<tr>
<td>Pump inlet angle</td>
<td>-40.70</td>
<td>Degree</td>
</tr>
</tbody>
</table>
3.2.2 Torque Converter Clutch

During pre-lock up, clutch clamping force \( (F_c) \) is clutch clamping has baseline magnitude of 4400 N [31]. The coefficient of static friction \( (\mu_0) \) between the clutch plates and the friction facing is very crucial for the clutch performance. It determines the capability of torque transmission of a clutch as well as torsional vibration of a driveline. If the value of friction coefficient gradient \( (\mu' \) ) is negative, the self-excited vibration occurs. The gradient of friction coefficient in this study is set to -0.0001 rad/s and the static friction coefficient is set to 0.28 [31]. In this research, the following model [12] is used.

\[
\mu = \mu_0 + \mu'(w_{SL})
\]  

(79)

Also, the spring constant for torque lockup clutch disc is considered 1500 Nm/rad.

3.2.3 Transmission Gearbox

Table 3.3: represents gear ratios for the four speed 2004 Honda CR-V [32].

Table 3.3 Gear ratios of the Honda CR-V [32].

<table>
<thead>
<tr>
<th>Gear Ratios</th>
<th>Gear Ratio</th>
<th>Vehicle velocity (km/hr)</th>
</tr>
</thead>
<tbody>
<tr>
<td>( \frac{r}{s_1} ) = 2.8</td>
<td>1</td>
<td>( v &lt; 15 )</td>
</tr>
<tr>
<td>( \frac{r\epsilon_1}{s_1\epsilon_2} ) = 1.6</td>
<td>2</td>
<td>( 15 &lt; v &lt; 30 )</td>
</tr>
<tr>
<td>( \frac{r}{2\epsilon_2-2\epsilon_1+s_1} ) = 1.1</td>
<td>3</td>
<td>( 30 &lt; v &lt; 50 )</td>
</tr>
<tr>
<td>( \frac{r}{2\epsilon_2} ) = 0.8</td>
<td>4</td>
<td>( 50 = v )</td>
</tr>
</tbody>
</table>
3.2.4 Propeller Shaft

Propeller shaft in this study is split between two equal elements. Spring mass damper system considered between two elements. The constant values used in this study are summarized in Table 3.4.

<table>
<thead>
<tr>
<th>Name</th>
<th>Value</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>Mass of propeller shaft ($M_p$)</td>
<td>5.08</td>
<td>kg</td>
</tr>
<tr>
<td>Outer radius of propeller shaft ($R_{op}$)</td>
<td>0.03175</td>
<td>m</td>
</tr>
<tr>
<td>Inner radius of propeller shaft ($R_{ip}$)</td>
<td>0.0301</td>
<td>m</td>
</tr>
<tr>
<td>Modulus of rigidity (steel) ($G$)</td>
<td>80</td>
<td>GPa</td>
</tr>
<tr>
<td>Length of propeller shaft ($L$)</td>
<td>2.1463</td>
<td>m</td>
</tr>
</tbody>
</table>

3.2.5 Vehicle Body

The vehicle constant values used are summarized in Table 3.5 represent for 2004 Honda CR-V model [37-39].

<table>
<thead>
<tr>
<th>Name</th>
<th>Value</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>Vehicle mass ($m$)</td>
<td>1452</td>
<td>kg</td>
</tr>
<tr>
<td>Vehicle frontal area ($A$)</td>
<td>2.49</td>
<td>m²</td>
</tr>
<tr>
<td>Ambient air density ($\rho_a$)</td>
<td>1.225</td>
<td>kg/m²</td>
</tr>
</tbody>
</table>
3.3 Simulation Algorithm

- Simulation has run to observe effect of clutch judder on torque, angular velocity and vehicle velocity.
- During pre-lock up, clutch-clamping force \( F_c \) has baseline magnitude of 4400 N [7].
- The gradient of friction coefficient \( \mu' \) in this study is set to -0.0001 rad/s [7].
- The coefficient of static friction \( \mu_0 \) is set to 0.28 [7].
- Spring constant for torque lock up clutch disc is considered at 1500 Nm/rad [7].
- Input values are vehicle speed at 2 m/s and first gear ratio of 2.8 used.

Step1:

- Powertrain equations are solved at an initial steady state condition.
- Set the initial time \( t(0) \) is zero, the time index \( (k) \) is zero. Choose the final throttle opening \( u_{ice,final} \) and throttle ramp time \( \Delta t \) values.

Step2:

Calculate the current throttle opening according to

\[
 u_{ice} = \min \left( u_{ice,initial} + \frac{(t(k+1))(u_{ice,final}-u_{ice,initial})}{\Delta t}, u_{ice,final} \right)
\]

where current simulation is from \( t(k) \) to \( t(k+1) \).
Step 3:

- Gear ratios are selected based on the current vehicle speed \( v \) in (m/s):

  
  \[
  \begin{align*}
  &\text{if } v < 4.1666 \text{ m/s} \\
  &\quad \text{G}=1 \\
  &\text{else if } v < 8.3333 \text{ m/s} \\
  &\quad \text{G}=2 \\
  &\text{else if } v < 13.889 \text{ m/s} \\
  &\quad \text{G}=3 \\
  &\text{else} \\
  &\quad \text{G}=4
  \end{align*}
  \]

Step 4:

Three operating conditions of torque converter unlock, pre-lockup and full lockup are used to solve equations. Based on turbine speed, an operating condition is selected.

  
  \[
  \begin{align*}
  &\text{if } w_t \geq 0.85w_p \text{ and } w_t < 0.99w_p \\
  &\quad K_p = \frac{1500(\frac{gL}{L})}{1500 + (\frac{gL}{L})} \\
  &\quad \text{Torque converter pre-lock is ON.}
  \end{align*}
  \]

  
  \[
  \begin{align*}
  &\text{else if } \text{abs}(w_{SL}) \leq 0.15 \\
  &\quad K_p = (\frac{gL}{L}) \\
  &\quad \text{Torque converter lockup is ON.}
  \end{align*}
  \]

  
  \[
  \begin{align*}
  &\text{else} \\
  &\quad K_p = (\frac{gL}{L}) \\
  &\quad \text{Torque converter is open.}
  \end{align*}
  \]

Step 5:

Simulate the system given the gear ratio and torque converter operating condition.

Step 5A:
If the torque converter is open, then simulate powertrain that includes the steady-state torque converter equations. The solution is over a time increment of 0.5 s.

Step 5B:
If the torque converter is in pre-lockup, then simulate powertrain that includes the clutch torque transfer equations. The solution is over a time increment of 0.02 s.

Step 5C:
If the torque converter is in lockup, then simulate powertrain that includes turbine speed and torque equal to the pump speed and torque. The solution is over a time increment of 0.5 s.

Step 6:
Update the time index \( k = k + 1 \).
If the final time is reached, then stop. Otherwise, go to Step 2.

3.4 Summary
Chapter 3 represented simulation setup for MATLAB. The constants are listed in this chapter, which are used in differential and algebraic equations. Finally, simulation algorithm is set forth to describe how MATLAB programming is done.
CHAPTER 4
SIMULATION OF A HONDA CR-V

4.1 Introduction

MATLAB simulations were performed for 2004 Honda CR-V. The objective of this study is to develop control-oriented automatic transmission powertrain model and simulation with judder for future control work. This study focused on the demonstration of clutch judder which affects shift quality, and evaluate judder severity with regard to several parameters. Simulation results are compared based on following parameters.

<table>
<thead>
<tr>
<th>Scenario 1</th>
<th>Parameter</th>
<th>Varying</th>
<th>Fixed</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>$u_{ice, final}$</td>
<td></td>
<td>X</td>
</tr>
<tr>
<td></td>
<td>$\Delta t$</td>
<td>X</td>
<td></td>
</tr>
<tr>
<td></td>
<td>$F_c$</td>
<td></td>
<td>X</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Scenario 2</th>
<th>Parameter</th>
<th>Varying</th>
<th>Fixed</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>$u_{ice, final}$</td>
<td>X</td>
<td></td>
</tr>
<tr>
<td></td>
<td>$\Delta t$</td>
<td></td>
<td>X</td>
</tr>
<tr>
<td></td>
<td>$F_c$</td>
<td></td>
<td>X</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Scenario 3</th>
<th>Parameter</th>
<th>Varying</th>
<th>Fixed</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>$u_{ice, final}$</td>
<td>X</td>
<td></td>
</tr>
<tr>
<td></td>
<td>$\Delta t$</td>
<td></td>
<td>X</td>
</tr>
<tr>
<td></td>
<td>$F_c$</td>
<td></td>
<td>X</td>
</tr>
</tbody>
</table>
Self-excitation from clutch (judder) during engagement process is expected to cause fluctuation in torque and subsequently reduce driver comfort. Therefore, torque and angular velocity at each component of drivetrain is measured to determine effect of judder. Also, vehicle velocity is measured to determine effect of judder.

The results obtained in this study are validated with literature referred for this study. Torque converter steady state equations calculations are in good agreement with SAE J643 test data mentioned in Pohl [34]. Also, data for engine map is validated with Meyer et al. [37].

4.1 Simulation Profile with Different Throttle Ramp Time- Scenario 1

The first simulation conducted with changing throttle ramp time. Simulation with ramp time 5s, 10s and 20s carried out. The final throttle \( u_{ice,final} \) is 0.4 or 40% of fully open.

![Graph showing simulation profile](image)

Figure 4.1 Simulation profile for (- - -) pump angular velocity at 5s, ( - - - ) pump angular velocity at 10 s, (- - - ) pump angular velocity at 20 s, ( blue ) turbine angular velocity at 5 s, ( green ) turbine
angular velocity at 10 s, (---) turbine angular velocity at 20 s, with final throttle opening is 40% and clamping force is 1.25\(F_c\)

It is observed that pump angular speed is greater than turbine angular speed for all three throttle ramp times until clutch engagement. After 70 s, pre-lock up condition starts, and pump and turbine speed become equal after a short time. It is seen that during clutch engagement, turbine angular speed (\(w_t\)) is not smooth which leads to vibration in drivetrain. With an increase in throttle ramp time 5 s, 10 s, and 20 s onset of clutch judder is delayed, however still judder is observed. For throttle ramp time of 5 s, turbine angular speed (\(w_t\)) fluctuates between 71 s and 73 s. For throttle ramp time 10 s, turbine angular speed (\(w_t\)) fluctuates between 74 s and 76 s. For throttle ramp time 20 s, turbine angular speed (\(w_t\)) fluctuates between 80 s and 82 s. Hence, it is observed that onset of clutch judder is delayed with increasing throttle ramp time but fluctuation takes place in turbine angular velocity due to judder. Also, for three different throttle ramp time similar fluctuations are observed in turbine angular velocity between 173 rad/s and 159.5 rad/s.

Pump torque (\(T_p\)) and turbine torque (\(T_t\)) are also plotted to observe effect of judder with change in throttle ramp time. Figure 4.2 represents pump and turbine torque plot versus time with 5 s, 10 s and 20 s throttle ramp time.
Figure 4.2: Simulation profile for (---) pump torque at 5s, (-----) pump torque at 10s, (-----) pump torque at 20 s, (---), turbine torque at 5 s, (----) turbine torque at 10 s, (-----) turbine torque at 20 s, with final throttle opening is 40% and clamping force is 1.25\(F_c\)

From Figure 4.2, it is observed that clutch engagement starts after 70 s causes turbine torque fluctuates due to self-excitation of clutch. Turbine torque profile for all three-throttle ramp times are not smooth and fluctuate before full lockup. It is observed that with an increase in throttle ramp time fluctuation time in turbine torque is delayed. This is because onset of clutch judder is delayed. However, judder is still observed with an increase in throttle ramp time. The turbine torque fluctuates between 147 Nm and 65 Nm for all three different throttle ramp times. Magnitude of fluctuation in turbine torque for three different throttle ramp times is same. After, 80 s full lockup condition occurs and turbine and pump torque become equal.

Propeller shaft input and output angular difference are plotted in Figure 4.3. It shows that input and output angular difference of propeller shaft drops from 0.052 rad/s to 0 rad/s during clutch engagement.
Velocity profile in Figure 4.4 shows that during clutch engagement judder has no discernible effect of velocity with 40% final throttle opening and $1.25F_c$ clutch clamping force. Velocity profile obtained is smooth for all three throttle ramp times of 5 s, 10 s and 20 s. The dotted lines represent gear ratios for different throttle ramp time.
Figure 4.4 Simulation profile for vehicle velocity with throttle ramp time 5 s, 10 s and 20 s, final throttle opening 40% and clutch clamping force $1.25F_c$.

4.2 Simulation with Different Final Throttle Opening- Scenario 2

Simulations are also performed to run the system with 40%, 60%, 80% and 100% final throttle command. This time, throttle ramp time is kept at 10 s and a clutch clamping force of $1.25F_c$. Figure 4.5 shows pump angular velocity with different throttle opening. It is observed that there is drop in pump angular velocity during clutch engagement between for 40%, 60% and 80% and 100% final throttle opening. Also, with final throttle opening more than 80%, pump angular velocity drops between 800 rad/s and 250 rad/s. This large drop takes place due to judder.
Figure 4.5: Simulation profile for pump angular velocity ($\omega_p$) with 40%, 60%, 80% and 100% final throttle opening, 10 s throttle ramp time and $1.25F_c$ clutch clamping force.

Simulation run again for pump angular velocity with different final throttle opening, throttle ramp time of 10s and clutch clamping force increased from $1.25F_c$ to $1.5F_c$. From Figure 4.6, it is observed that pump angular velocity profile for 80% final throttle opening is improved compared to pump angular velocity profile for 80% final throttle opening in Figure 4.5. However, with 100% final throttle opening judder effect is still observed causing pump angular velocity to drop from 752 rad/s to 300 rad/s.
Figure 4.6: Simulation profile for pump angular velocity with 40%, 60%, 80% and 100% final
throttle opening with 10s throttle ramp time and 1.5$F_c$ clutch clamping force

From Figure 4.5 and Figure 4.6, it is observed that with increasing clutch clamping force
from 1.25$F_c$ to 1.5$F_c$ judder effect can be minimized.

Figure 4.7 represents turbine angular velocity profile for different final throttle opening
with clutch clamping force of 1.25$F_c$. Each curve shows fluctuation in turbine angular velocity
during clutch engagement. Also, fluctuations are more with final throttle opening of 80% and
above. Turbine angular velocity with final throttle opening of 80% and above fluctuate between
800 rad/s and -250 rad/s due to judder.
Figure 4.7: Simulation profile for turbine angular velocity ($\omega_t$) with 40%, 60%, 80% and 100% throttle opening with 10s throttle ramp time and $1.25 F_c$ clutch clamping force.

To minimize effect of judder, simulation run with increasing clutch clamping force from $1.25 F_c$ to $1.5 F_c$ for turbine angular velocity for different final throttle opening. Results are summarized in Figure 4.8. From Figure 4.8, it is observed that turbine angular velocity profile for final throttle opening of 80% is now improved compared to Figure 4.7 but still turbine angular velocity profile with final throttle opening of 100% fluctuates between 750 rad/s and -150 rad/s during clutch engagement due to judder.
From Figure 4.7 and Figure 4.8, it is observed that judder appears during clutch engagement for different final throttle opening. However, effect of judder can be minimized with increasing clutch clamping force to appropriate value based on final throttle opening.

Figure 4.9 represents simulation profile for pump torque for final throttle opening of 40%, 60%, 80% and 100% with 10 s throttle ramp time and 1.25\(F_c\) clutch clamping force. Peak is observed for each final throttle opening during clutch engagement.
Figure 4.9: Simulation profile for pump torque with 40%, 60%, 80% and 100% final throttle opening with 10 s throttle ramp time and 1.25$F_c$ clutch clamping force.

Figure 4.10 represents, pump torque profile for different final throttle opening, 10 s throttle ramp time and clutch clamping force of 1.5$F_c$. 
Figure 4.10: Simulation profile for pump torque with 40%, 60%, 80% and 100% final throttle opening with 10 s throttle ramp time and $1.5F_c$ clutch clamping force

From Figure 4.10, it is observed that peak is still observed for each pump torque curve. However, with increasing clutch clamping force from $1.25F_c$ to $1.5F_c$, pump torque curve with 80% final throttle opening is improved compared to pump torque for 80% final throttle opening in Figure 4.9. It is observed that, fluctuation in pump torque can be minimized with increase in clutch clamping force to appropriate value based on final throttle opening.
Figure 4.11: Simulation profile for turbine torque with 40%, 60%, 80% and 100% final throttle opening with 10 s throttle ramp time and $1.25F_c$ clutch clamping force

From Figure 4.11, it is observed that, fluctuation in turbine torque takes place during clutch engagement for different final throttle opening. Also, with final throttle opening of 80% and above, torque at turbine fluctuate between 140 Nm and -150 Nm.
Figure 4.12: Simulation profile for turbine torque with 40%, 60%, 80% and 100% final throttle opening with 10 s throttle ramp time and $1.5F_c$ clutch clamping force.

Figure 4.12 represents, turbine torque curve with different final throttle opening, 10 s of throttle ramp time and clutch clamping force of $1.5F_c$. It is observed that, judder appears in each curve and effect of judder is more with final throttle opening of 100%. However, with increasing clutch clamping force from $1.25F_c$ to $1.5F_c$ torque curve at 80% final throttle opening is improved compared to the torque curve at 80% final throttle opening in Figure 4.11. Thus, it is required that appropriate selection of clutch clamping force is necessary depend on final throttle opening to minimize judder effect and fluctuation in turbine torque.

The propeller shaft input and output angular difference simulation carried out for the different final throttle openings, 10 s throttle ramp time and $1.25F_c$ clutch clamping force. Results are summarized in Figure 4.13.
Figure 4.13: Simulation profile for input and output angular difference of propeller shaft for 40%, 60%, 80% and 100% final throttle opening, 10s throttle ramp time and 1.25$F_c$ clutch clamping force.

From the Figure 4.13, it is observed that, for each final throttle opening, angular difference start reducing to zero during clutch engagement process and once, full lockup clutch condition occurs, difference becomes zero. It is observed that, for 40% of final throttle opening, pre-lock up starts at 75 s compared to the other three final throttle openings where pre-lock up starts between 30 s and 42 s.

The simulation for angular difference of propeller shaft also carried out for 1.5$F_c$ clutch clamping the different final throttle openings and results are summarized in Figure 4.14. It is observed that, with increasing clutch clamping force from 1.25$F_c$ to 1.5$F_c$ similar curves observed compared to Figure 4.13 for different final throttle opening.
Thus, from Figure 4.13 and Figure 4.14, it is concluded that with increasing clutch clamping force has no effect in input and output angular difference of propeller shaft. Also, during clutch engagement, angular difference drops to zero for different final throttle opening.

Velocity profile for 40%, 60%, 80% and 100% final throttle opening, 10s throttle ramp time and $1.25F_c$ clutch clamping force is simulated and results are shown in Figure 4.15 and Figure 4.16.
Figure 4.15: Simulation profile for vehicle velocity for 40%, 60%, 80% and 100% final throttle opening, 10 s throttle ramp time and $1.25F_c$ clutch clamping force.
From Figure 4.15, it is observed that velocity profile for 80% and 100% throttle opening is not smooth and it gives jerks to the driver during clutch engagement. Figure 4.16 shows closer view for 80% and 100% final throttle opening curve in Figure 4.15. Vehicle velocity with 80% and above final throttle opening fluctuate between 38 s and 60 s which indicate effect of judder.

Simulation has also been conducted for velocity profile with different final throttle opening and 1.5$F_c$ clutch clamping force. From Figure 4.17 it is observed that, with 100% final throttle opening, velocity is not smooth and gives jerk to driver during clutch engagement. Figure 4.18 shows closer view for 100% final throttle opening in Figure 4.17.
Figure 4.17: Simulation profile for vehicle velocity for 40%, 60%, 80% and 100% final throttle opening, 10 s throttle ramp time and $1.5F_c$ clutch clamping force
However, it is observed from Figure 4.16 that with increasing clutch clamping force from $1.25F_c$ to $1.5F_c$ velocity curve with 80% final throttle opening is improved compared to the velocity curve of 80% final throttle opening in Figure 4.15.

4.3 Simulation with Different Clutch Clamping Force- Scenario 3

Clutch clamping force is normally applied to pressure plate of the clutch. For the scenario 3, clutch clamping force is increased by 25%, 50%, 75% and 100% of the original clutch clamping force of 4400 N and final throttle opening is kept constant at 80% and throttle ramp time of 10 s.

Figure 4.19 represents pump angular velocity profile with different clutch clamping forces. From Figure 4.19, it is observed that, clutch engagement starts after 35 s. Pump angular velocity for $1.25F_c$ drops during clutch engagement from 750 rad/s to 250 rad/s. Also with the other three
clutch clamping forces, pump angular velocity drops from 305 rad/s to 258 rad/s during clutch engagement.

This large drop in pump angular velocity for $1.25F_c$ compared to the other three curves takes place because there is low clutch clamping force is available for final throttle opening of 80%.

Figure 4.19: Simulation profile for pump angular velocity with $1.25F_c$, $1.5F_c$, $1.75F_c$ and $2F_c$ clamping force, 10 s throttle ramp time and 80% final throttle opening

Figure 4.20 shows turbine angular velocity curves for different clutch clamping forces, throttle ramp time of 10 s and final throttle opening of 80%
Figure 4.20: Simulation profile for turbine angular velocity with $1.25F_c$, $1.5F_c$, $1.75F_c$ and $2F_c$ clamping force, 10 s throttle ramp time and 80% final throttle opening.

From Figure 4.20, it is observed that with low clutch clamping force turbine angular velocity has more fluctuation. For, $1.25F_c$ clutch clamping force, turbine angular velocity fluctuates between 750 rad/s and -220 rad/s. With an increase in clutch clamping force, fluctuations are minimized.

Figure 4.21 shows torque at pump for different clutch clamping forces and final throttle opening of 80%. From Figure 4.21, it is observed that pump torque curve with clutch clamping force of $1.25F_c$ fluctuates more compared to other clutch clamping forces during clutch engagement.
Figure 4.21: Simulation profile for pump torque with $1.25F_c$, $1.5F_c$, $1.75F_c$ and $2F_c$ clamping force, 10s throttle ramp time and 80% final throttle opening

Turbine torque for different clutch clamping forces are summarized in Figure 4.22. From Figure 4.22, it is observed that turbine torque with clutch clamping force of $1.25F_c$ fluctuates more compared to other clutch clamping forces during clutch engagement. Thus, it shows that there needs to be appropriate clutch clamping force is necessary based on final throttle opening.
Simulation has been conducted for input and output angular difference of propeller shaft with different clutch clamping force and 80% throttle opening. Results are summarized in Figure 4.23. From Figure 4.23, it is observed that input and output angular difference of propeller shaft curve follows same path for different clutch clamping force during unlock, pre-lock and full lock up condition. During pre-lock condition, difference drops from 0.082 rad/s to 0 rad/s.
Finally, simulation for velocity profile has been conducted with different clutch clamping force. Results are summarized in Figure 4.24. From Figure 4.24, it is observed that, with $1.25F_c$ clutch clamping force and 80% throttle opening velocity profile is not smooth. During clutch engagement, velocity fluctuates. It is observed that with 80% final throttle opening, if clutch clamping force is not sufficient velocity profile is not smooth and gives jerk and poor driver’s feel. Figure 4.25 shows closer view for velocity profile for $1.25F_c$ during clutch engagement. Velocity shows fluctuations between 40 s and 60 s.
Figure 4.24: Simulation profile for velocity with different clutch clamping force, 10 s throttle ramp time and 80% final throttle opening.
Thus, from various Figures in scenario 3, it is observed that for 80% final throttle opening, if clutch clamping force is $1.5F_c$ or below, torque and angular velocity fluctuates during clutch engagement. Also, vehicle velocity gives jerks. So, there is necessity to select appropriate clutch clamping force based on final throttle opening otherwise judder can be seen during clutch engagement.
4.4 Summary

It is observed that, with changing throttle ramp time onset of judder is delayed. However, judder is still observed and fluctuation in angular velocity, as well as torque at turbine takes place. Judder has no effect on final velocity for changing throttle ramp time. With changing final throttle opening, it is observed that if clutch clamping force is below $1.5F_c$ for 80% and above final throttle opening, large fluctuation in torque and angular velocity at turbine occurred. Also, velocity profile is not smooth and provides poor driver’s feel. With changing clutch clamping force, it is observed that if final throttle opening is 80% clutch clamping force should be $1.5F_c$ and above to minimize effect of judder.
CHAPTER 5

CONCLUSIONS AND FUTURE WORK

5.1 Conclusions

This thesis has presented a control-oriented automatic transmission-based powertrain modeling and simulation. With this model, the vibration response characteristics of the clutch engagement and judder are investigated. Clutch judder is demonstrated and analyzed with MATLAB simulation. Effect of changing throttle opening, throttle ramp time and clutch clamping force is observed on clutch judder. From, simulation work it is observed that, clutch judder is self-excitation which occurs during clutch engagement. It has negative effect on drivetrain as well as driver’s feel.

Judder appeared with changing throttle ramp time. With increasing throttle ramp time, onset of judder was delayed. However, judder is observed for each throttle ramp time. With changing throttle ramp time, fluctuation of torque and angular velocity at turbine is observed. There is no effect of judder observed on vehicle velocity for changing throttle ramp time.

Judder appeared for different final throttle opening. For 10 s throttle ramp time and $1.25F_c$ clutch clamping force, angular velocity and torque at turbine above 80% final throttle opening fluctuate tremendously because of clutch judder. However, this effect can be minimized with increasing clutch clamping force from $1.25F_c$ to $1.5F_c$. Vehicle velocity with 80% and above final throttle opening fluctuates if the clutch clamping force is $1.25F_c$. With increasing clutch clamping force from $1.25F_c$ to $1.5F_c$, effect of judder on vehicle velocity can be minimized. There is possibility that with increasing clutch clamping force may lead to increased clutch disc wear and shorter life.

Judder appeared in the system if clutch clamping force is $1.25F_c$ with 10 s of throttle ramp time and final throttle opening is 80%. Torque and angular velocity at turbine fluctuates with $1.25F_c$ clutch clamping force and 80% final throttle opening. Also, vehicle velocity is not smooth and gives jerk. There is necessity to develop controls that regulates clutch clamping force based on final throttle opening.
Judder can be regulated by appropriate clutch clamping force and final throttle opening. Final throttle opening should not be more than 80% with clutch clamping force $1.25F_c$. Otherwise, it leads to fluctuating torque and angular speed at turbine. Final throttle opening of more than 80% with low judder is possible with selection of appropriate clutch clamping force.

More understanding of the driveline system behavior during clutch engagement is still required. There are very few publications and literatures are available which covers behavior of the driveline system under the transient operating condition. Therefore, more attention to the effects of transient behavior is needed. Many researchers have investigated clutch judder by simulations and experiments. However, there is lack of control-oriented powertrain model that is capable of capturing clutch judder and its effects. In this thesis work, negative friction gradient is considered. Results show that there is need to adjust friction coefficient to positive based on literature.

5.2 Future Work

Negative friction gradient is the main contributor to increase the clutch judder. So, first implementation in future work is run the model with available positive friction coefficient materials like FeramAlloy and observe the effects of judder. FeramAlloy is combination of Ceramic and Feramics. Ceramics is mixture of silicon dioxide and several additives while Feramics is blend of metals and silicon dioxide. There are some other elements in powertrain which also causes torsional vibration which lead bad shift quality. Further work is needed to analyze other contributor of torsional vibration. Additionally, the control-oriented nature of the powertrain simulator should be used in control studies. Controls are to be developed that regulate throttle level and lock up clutch clamping force to reduce judder in the system. It is observed from the results if there is lack of clutch clamping force available for final throttle opening, judder appears in the system which leads to fluctuation in torque and angular speed. So, it is necessity to choose appropriate clutch clamping force depending upon final throttle opening. In this study, designing controller that will regulate clutch-clamping force based on final throttle opening is future work.
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