Study on Automotive Driveline System with Torque Converter Clutch under Harmonically Varying Normal Load

by

Mohd Hazwan bin Md Din

Dissertation submitted in partial fulfilment of
the requirements for the
Bachelor of Engineering (Hons)
(Mechanical Engineering)

JULY 2008

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CERTIFICATION OF APPROVAL

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A project dissertation submitted to the
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BACHELOR OF ENGINEERING (Hons)
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Approved by,

(Dr. Setyamartana Parman)
Project Supervisor

UNIVERSITI TEKNOLOGI PETRONAS

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July 2008

CERTIFICATION OF ORIGINALITY

This is to certify that I am responsible for the work submitted in this project, that the original work is my own except as specified in the references and acknowledgements, and that the original work contained herein have not been undertaken or done by unspecified sources or persons.

Mohd Hazwan Bin Md Din

ABSTRACT

In this research project, an Automotive Driveline System with Torque Converter Clutch under effects of a harmonically varying normal load on transient responses of a 2-Degree Of Freedom torsional system is studied. An immediate application of this study is the slipping torque converter clutch that is employed in automotive driveline system. The purpose of Torque Converter Clutch is to transmit rotating power from the prime movers (internal combustion engine) to the rotating driven load. Up to 10 percent of the engine energy is wasted by torque converter slipping caused by negative slope introduced by substantial stick slip motion. Investigation on key parameters that control the clutch engagement rate and pronounce the stick slip motion could further attenuate such motion. The objectives of this project is to develop its mathematical model, the system is supposed to be 2-Degree of Freedom torsional system with non-linear dry friction. Under this mathematical model, the governing equation that represents the dynamics of the system and non-linear time varying dry friction formulation is established. Then the development of MATLAB programming to simulate the dynamics of the system is established. From the numerical simulation in MATLAB programming, transient responses including pure slip and stick slip transient response of the system under a harmonically varying load and sinusoidal torque excitation will be studied computationally. The effects of saturation friction torque, mean torque and engine inertia on clutch engagement rate are found significant in pure slip transient response. Furthermore, the effects of kinetic friction, saturation friction torque and phase lag on the stick-slip transient response (judder) influence the dynamics of the system by the introduction of stick slip motion.

ACKNOWLEDGEMENT

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LIST OF ABBREVIATIONS

NONMENCLATURE

C	Torsional viscous damping coefficient
DOF	Degree of freedom
Sgn	Signum
I	Torsional inertia
K	Torsional stiffness
MHBM	Multi-term harmonic balance method
N	Normal load
NLTV	Nonlinear time-varying
T	Torque

GREEK LETTERS

μ	Coefficient of friction
θ	Absolute angular displacement
δ	Relative angular displacement
ψ	Phase lag
σ	Conditioning factor
ω	Excitation frequency (rad/s)

SUBSCRIPTS

1,2,3	Inertial element indices
e	Engine
f	Friction
k	Kinetic
m	Mean
s	Static
p	Fluctuating component

SUPERSCRIPTS

$\bar{}$	Normalized value
‘	First derivative with respect to time
‘’	Second derivative with respect to time

OPERATORS

$\|\|$	Absolute value
$\langle\ \rangle_t$	Time average operator
[]	Matrix operation

CHAPTER 1
INTRODUCTION

1.1 Background of Study

Generally, Torque Converter Clutch is a significant mechanism in the automotive driveline system powered by internal combustion engine. The Torque Converter is mounted together with the flywheel between the engine and the transmission as shown in Figure 1.1. The shaft from the engine and the transmission are hooked-up to the Torque Converter to establish the complete connection in the automotive driveline system. The idea behind the Torque Converter is to transmit rotating power from the internal combustion engine to the transmission without having a clutch in the middle of power transmission. A significant application for the Torque Converter is used in every automatic transmission vehicle where it replaces the clutch application in manual transmission vehicle.

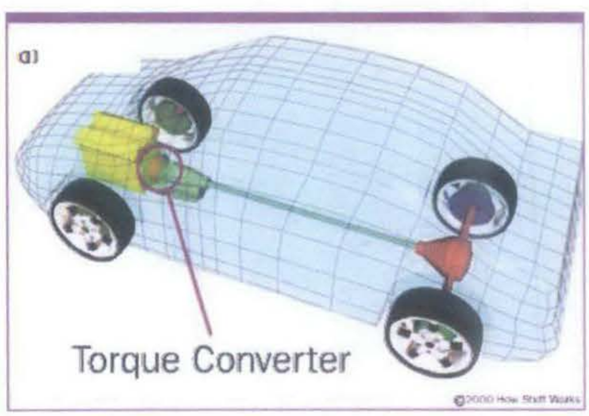


Figure 1.1: Location of torque converter in automotive driveline system

A torque converter is a modified form of fluid coupling that is used to transfer rotating power from a prime mover, such as an internal combustion engine or electric motor, to a rotating driven load. The torque converter normally takes place of a mechanical clutch, allowing the load to be separated from the power source and it's able to multiply torque provided by the engine when there is substantial difference between the output and input rotational speed, thus providing the equivalent of a reduction gear [1]. The torque converter consists of three components inside a very strong housing. The components are pump, turbine, and stator as shown in Figure 1.2. The ability of the torque converter to multiply the torque from the engine is influence by the design of the four component have been mentioned before. The pump has specially curved vanes and is driven by the engine's crankshaft. The turbine also has specially curved vanes and is connected to the input shaft of the transmission. Adding a third element, the stator (also called the reactor), gives the assembly the capability it's named for. The stator has vanes and is mounted on a one-way clutch, to allow it to freewheel in only one direction. The stator assembly is located between the impeller and turbine and redirects oil that bounces back off the turbine. The force of the redirected oil assists in rotating the turbine, resulting in torque multiplication. When the impeller's speed is high and turbine's speed is low, torque can be multiplied by as much as 2:1 [2].

Figure 1.2: Torque converter components (from left to right); turbine, stator, pump.

1.2 Problem Statement

The investigation on the effect of harmonically varying normal load on the dynamic of the 2-DOF with the nonlinear dry friction is studied. The implementation of this system is applied in Torque Clutch Converter (TCC). Up to 10 percent of the engines energy is wasted by torque converter slippage and the speed difference between the impeller and turbine during the coupling phase. In an automotive driveline system with torque converter, the negative slope in friction-velocity characteristic introduces substantial stick-slip motions. A well tuned normal load could possibly attenuate such motion with key parameters controlling the engagement rate are identified.

1.3 Objective of the Study

- To derived a mathematical model of the automotive driveline system with Torque Converter as 2-DOF torsional system with nonlinear dry friction.
- To develop a MATLAB programming with respect to characteristic of the system in order to simulate the dynamics of the system.
- To study the pure slip transient response of the system.
- To study the stick-slip transient response.

1.4 Scope of the Study

In this study, the scope should be covered comprises of 2-DOF torsional system with nonlinear dry friction and under sinusoidal torque excitation. Load (*N*) applied in this study is time-varying load (*N*) to investigate the effects on the torsional system. Previous study on the same torsional system used constant load (*N*).

Dry friction element is selected from the other types of dissipation element such as viscous damping, fluid damping and hysteric damping because the dry friction element used as a key path to transmit the mechanical power in the Torque Converter system. A fundamental of vibration engineering, mathematic engineering and computer programming using MATLAB are required in this study.

CHAPTER 2
LITERATURE REVIEW

2.1 Recent work

2.1.1 Dynamics of a torsional system with harmonically varying dry friction torque

A single-DOF torsional system with dry friction element is studied in this journal. This system could be either nonlinear time-invariant (NLTI) or nonlinear time-varying (NLTV) type depending on the nature applied normal load (*N*). However this study focused on the (NLTV) type which results from surface irregularity or contact fluctuation. This type of system is applied in automotive driveline system where the friction element functions as a power transmission path with 2-DOF torsional system. The formulating of multi-term harmonic balance method is applied to the (NLTV) system.

The problem formulation for this system involves single-DOF torsional system as shown in Figure 2.1, with $\bar{I}$ as moment of inertia, $\bar{K}$ as a torsional stiffness, $\bar{C}$ as viscous damping term and $\bar{\theta}$ represent angular displacement. The dry friction element is assumed to be subjected to periodically normal load $\bar{N}(\bar{t})$. This (NLTV) system can be described mathematically by equation (2-1).

$$\bar{I}\ddot{\bar{\theta}} + \bar{C}\dot{\bar{\theta}} + \bar{K}\bar{\theta} + \bar{T}_f\,(\dot{\bar{\theta}},\bar{t}) = \bar{T}_f(\bar{\omega}\bar{t}) \tag{2-1}$$

The nonlinear time-varying friction torque is given by speed dependent friction coefficient $\mu(\dot{\bar{\theta}})$, effective arm $\bar{r}$ and $\bar{N}(\bar{t})$ as

$$\bar{T}_f\,(\dot{\bar{\theta}},\bar{t}) = \bar{r}\;\mu\dot{\bar{\theta}}\bar{N}(\bar{t}) \tag{2-2}$$

The normal load (N) is assumed to be periodic with fundamental frequency $\bar{\omega}/\kappa$ where κ is an integer. The dry friction is described by classical coulomb law though static and kinetic friction is assumed to be identical in this study.

$$\bar{N}(\bar{t})=N_0+\Sigma_{n=1}\left(\bar{N}_{2n-1}\sin\left(\frac{n}{\kappa}\,\bar{\omega}\bar{t}\right)+\bar{N}_{2n}\sin\left(\frac{n}{\kappa}\,\bar{\omega}\bar{t}\right)\right) \tag{2-3}$$

$$\bar{\mu}\dot{\theta}=\begin{cases}\bar{\mu} & \left|\dot{\bar{\theta}}\right|>0\\ [-\bar{\mu}\;\bar{\mu}] & \left|\dot{\bar{\theta}}\right|=0\end{cases} \tag{2-4}$$

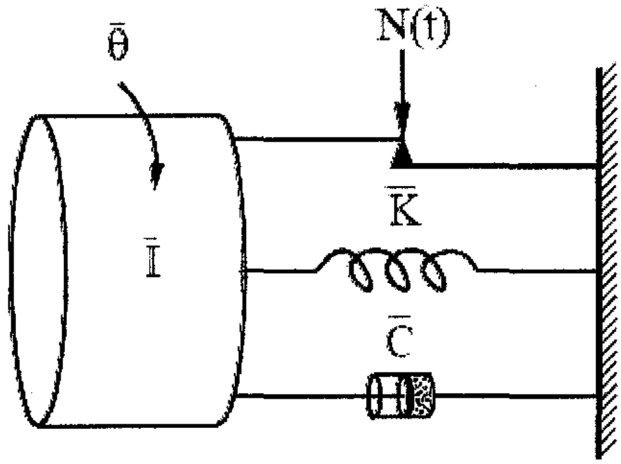


Figure 2.1: Single-DOF torsional, (NLTV) system.

Equation (2-1) can be further simplified by introducing the following dimensionless parameters.

$$\bar{\omega}_n=\sqrt{\bar{K}/\bar{I}}\,,\qquad \zeta=\bar{C}/2\sqrt{\bar{K}\bar{I}},\qquad \bar{\theta}_c=\bar{T}_p/\bar{K},$$

$$\theta=\bar{\theta}/\bar{\theta}_c,\qquad T_p=\bar{T}_p/\bar{T}_p=1.0,\qquad T_f=\bar{T}_f/\bar{T}_p,$$

$$\Omega=\bar{\omega}/\bar{\omega}_n,\qquad \tau=\bar{\omega}_n\bar{t},$$

Finally the following dimensionless equation is obtained represented by equation (2-5)

$$\theta'' + 2\zeta\theta' + \theta + T_f(\theta',\tau) = T_p(\Omega\tau) \tag{2-5}$$

Where $T_f(\vartheta',\tau)$ is nonlinear time varying term.

Multi-term harmonics balanced methods is applied to solved equation (2-5) and determine the periodic response of $\theta(\tau)$. Essentially, the MBHM construct approximate periodic response that is presented by a truncated Fourier series.

2.1.2 Vibration control to avoid stick-slip motion

Friction occurs in everyday phenomena in engineering points of view. Two different significant phenomena that can be identified is first by the resistance against the start of relative motion of the bodies which caused by the contact between those bodies. Second is the resistance against the existing relative motion of two contacting bodies. The first phenomena are called static friction and the second phenomena are called kinetic friction. Static friction and kinetic friction are express by the Coulomb's Law by equation (2-6a) and (2-6b) below

$$|F_R| \leq \mu_0 F_N \tag{2-6a}$$

$$F_R = \mu F_N \tag{2-6b}$$

where μ_0 is the coefficient of static friction, μ is the coefficient of kinetic friction and F_N is normal forces. The stick-slip motion occurs when there is an intermittent change between the static and kinetic friction. It is possible only if the dynamic properties of the regarded system such as elasticity, mass distribution and damping allow this type of motion. Stick-slip vibration appears in everyday life as well as engineering application. Examples are sound from violin string, brake groan and chatter of machine tools. Furthermore it's creates unwanted noise, diminished accuracy and increase wear [6].

In nonlinear vibration theory, stick-slip motion can be treated as a stable limit cycle of a self-excited vibration system. Such system encounter an energy source, an oscillator, and switching mechanism triggered by the oscillator that control the energy flow from the source to the vibrating system as depicted in Figure 2.2 .

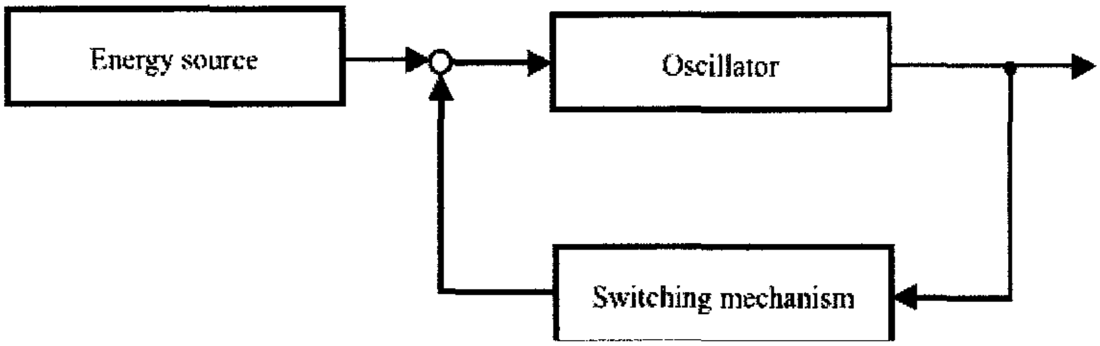


Figure 2.2: Block diagram of self-excited vibration system.

If the energy flow into a vibration system is greater or less than dissipation during one period, the equilibrium condition becomes unstable and the vibration amplitude will increase or decrease. If the energy input and output is balanced during each period, then the isolated periodic motion occur which is also known to be a limit cycle. The example for simple friction oscillator is shown in Figure 2.3.

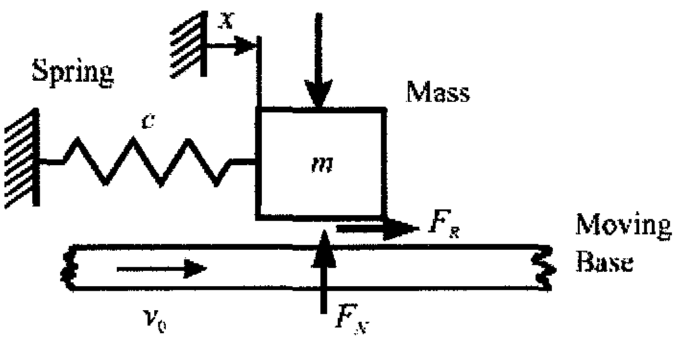


Figure 2.3: Model of simple friction oscillator.

The energy source is a moving base with constant velocity v_0=constant driving the mass of the discrete spring-mass oscillator. The friction force F_R usually depends on the relative velocity between base and mass oscillator. Different friction characteristics, $F_R = F_R(v_{rel})$ are shown in Figure 2.4.

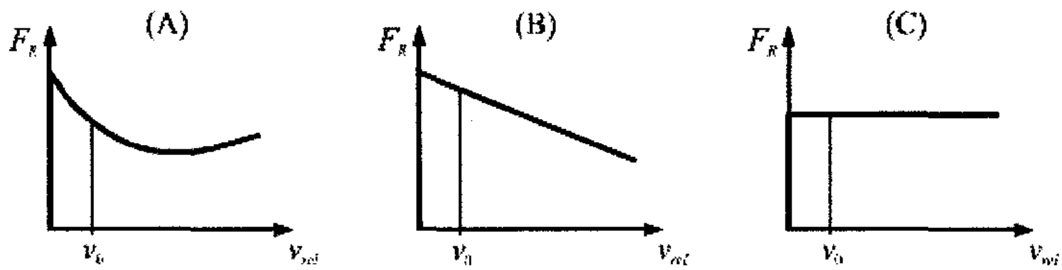


Figure 2.4: Friction characteristics :(A)generally decreasing (B)linearly decreasing (C)constant

In characteristics, (A) and (B) the limit value of the static friction force $F_R(v_{rel}\rightarrow 0)$ is greater than the kinetic friction force $F_R(v_{rel}\neq 0)$, thus the friction force characteristic is decreasing for a small value of v_{rel}. If the base velocity is adjusted so that the friction force shows a negative slope at the equilibrium state $x=x_s$, then this state become unstable. Hence, the amplitude grows and the trajectory in the x, phase plane reach fast limit cycle as depicted in Figure 2.5. From the limit cycle shown below, the slip mode A→B and the stick mode B→A can be distinguish. The physical reason for this instability is an energy transfer from the base to the mass.

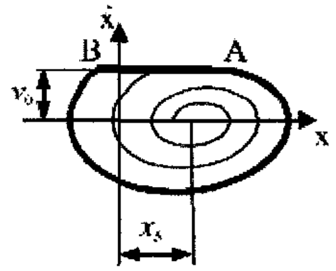


Figure 2.5: Limit cycle.

There are 3 different measures that are useful to avoid stick-slip motion and it's modelling in Figure 2.6.

(1) Effect of external damping.

An increase of external damping can compensate the negative damping induced by the decreasing friction characteristic.

(2) Effect of external excitation.

An additional external can be applied, either as harmonic excitation or as harmonic force excitation of the mass. This excitation can break up the robust stick-slip limit of the cycle.

(3) Effect of fluctuating normal forces.

Vibration control can be used to provide fluctuating normal forces that can result in a negative energy input.

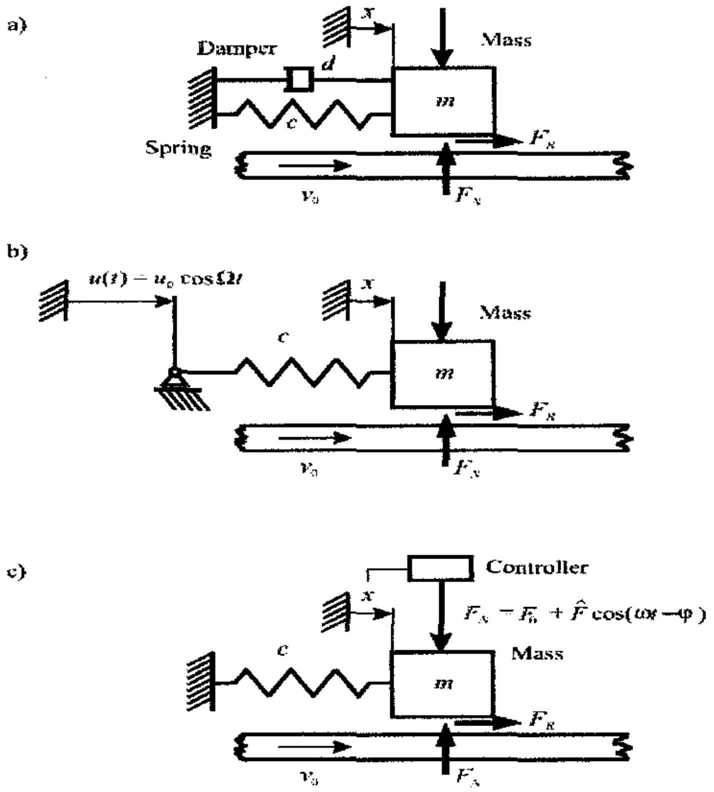


Figure 2.6: Measures to avoid stick-slip motion: (a) additional external damping (b) additional external excitation (c) fluctuating normal forces

2.2 Theory

2.2.1 Dry friction element

In this study, the investigation is focused on the effect of time varying N for the same torsional system. An immediate application of this work is the slipping torque converter clutch that is employed in an automotive driveline system, as shown in Figure 2.7. The dry friction element is used as a key path to transmit the mechanical power. In this study, actuation pressure $P(t)$ is assumed to be harmonic along with a mean term [3].

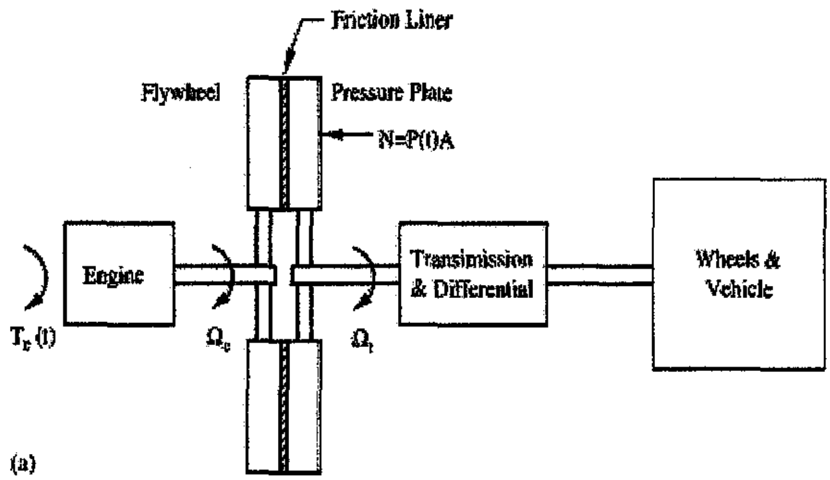


Figure 2.7: Automotive driveline system with torque converter

Dry friction elements or Coulomb damping are encountered in many mechanical and structural systems, under a variety of operational condition. This type of damping is due to the force caused by friction between two solid surfaces. The force acting on the system must oppose the motion; therefore, the sign of this force must have opposite sense (direction) of velocity. If the kinetic coefficient of friction is μ and the force compressing the surface is N, then

$$F(x') = \mu N \, sgn(x') \tag{2-7}$$

where *sgn* is the signum function, which take on the value of +1 for positive value of the argument, -1 for the negative values of the argument, and 0 when the argument is zero [4]. Although dry friction can result in loss of efficiency of internal combustion engine, wear on contacting parts, and the loss of position accuracy in servomechanism, it has been used to enhance the performance of turbo machinery blades, certain built-up structure, and earthquake isolation [5].One approach is presented for determining the governing equation of motion. The approach is based on force-balance method. The underlying principles of the force-balance method rely on the force and momentum imposed on a system to the rate of change of linear momentum and the rate of change of angular momentum.

2.2.2 Newton's second law of motion

Newton's second law of motion is applied in this study to derive the equation of motion of the system. In this study, the consideration focused on the equation of motion for forced nonlinear vibration with 2-DOF of a torsional system. In this system, the element involved including rotational mass or inertia, linear torsional spring element and lumped viscous damping element. The mass or inertia element is assumed to be a rigid body; it can gain or lose kinetic energy whenever the velocity of the body changes. From Newton's second law of motion, the product of the mass and acceleration is equal to the force applied to the mass. The viscous damping element is the most commonly used damping mechanism in vibration analysis. When mechanical system vibrate in a fluid medium such as air, gas, water and oil, the resistance offered by the fluid to the moving body caused energy to be dissipated. In viscous damping, the damping force is proportional to the velocity of the vibrating body. For a linear torsional spring, it is a type of the mechanical link that is generally assumed to have negligible mass and damping. A force is developed in the spring whenever there is relative motion between the two ends of spring. The spring force is proportional to the amount of deformation or displacement.

2.2.3 Vibration analysis of the automotive driveline system

A vibration analysis procedure is implemented to analyze a vibratory system of the automotive driveline system in this study. A vibratory system is a dynamic system for which variable such as the excitations (inputs) and responses (output) are time-dependent. The response of a vibrating system generally depends on the initial conditions as well as the external excitations. Most practical vibrating systems are very complex. Only the most important features are considered in the analysis to predict the behaviour of the system under specified input condition. Often the overall behaviour of the system can be determined by considering even a simple model of a complex physical system. Thus, the analysis of a vibrating system usually involves mathematical modelling, derivation of a governing equation, solution of the equation, and interpretation of the results [8].

2.2.4 Mathematical modelling

The purpose of the mathematical modelling is to represent all the important features of the system for the purpose of deriving the mathematical equations governing the system behaviour. The mathematical model should include enough details to describe the system in terms of equations without making it too complex. In this study the mathematical model involves nonlinear behaviour that is more complex compared to the linear behaviour. Nonlinear model sometimes reveal certain characteristics of the system that cannot be predicted using linear model. In this approach, first a very crude or elementary model is used to a insight into the overall behaviour of the system. Subsequently, the model is refined by including more component or details so that the behaviour of the system can be observed more closely [4]. To illustrate the procedure of refinement used in mathematical modelling in this study, consider the automotive driveline system with torque converter shown in Fgure 2.7. The mechanical power produce by the engine is transmitted through the flywheel to the transmission and finally to the wheel of the vehicle. The components that built-up the flywheels are pressure plate and friction liner that applied a dry friction element as a key path to the mechanical power transmission. For a refined approximation the combined torsional inertia of the engine and inertia of the friction shoe and pressure plate are represent separately with a 2-DOF model as shown in Figure 2.8.

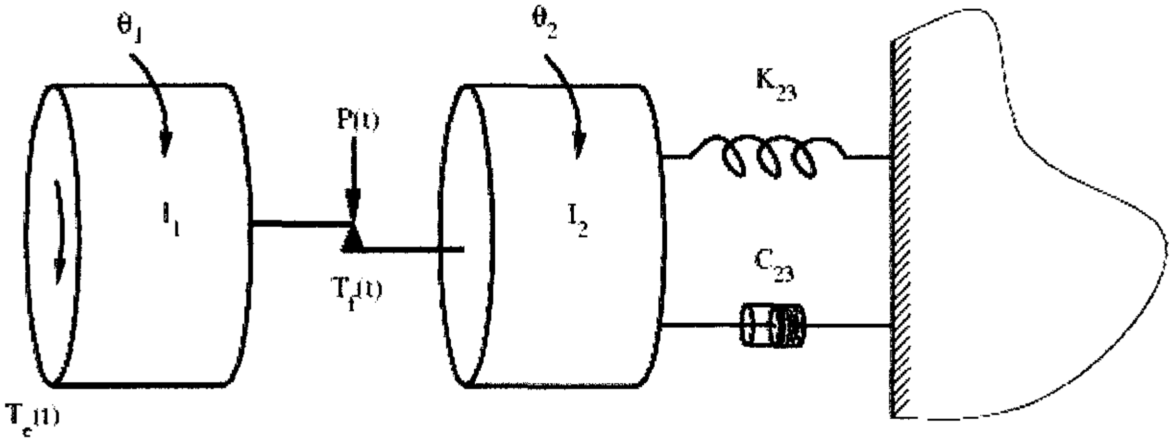


Figure 2.8: Mathematical model of 2-DOF of the system

2.2.5 Governing equation of 2-DOF torsional system

As shown in Figure 2.8, the nonlinear friction torque T_{sf} is transmitted within the TCC is applied by hydraulic actuation pressure $P(t)$, and in this study $P(t)$ is assumed to be harmonic along with a mean term. Here I_1 represent the combined torsional inertia of a flywheel, I_2 is the inertia of the friction shoe and pressure plate, and I_3 is the wheel and vehicle sub-system that is assumed to be rigid. The governing equations of this system are

$$I_1\ddot{\theta}_1 + T_f\left(\dot{\theta}_1 - \dot{\theta}_2, t\right) = T_e(t) = T_m + T_p \sin(\omega t) \qquad \text{(2-8a)}$$

$$I_2\ddot{\theta}_2 + C_{23}\dot{\theta}_2 + K_{23}\theta_2 = T_f\left(\dot{\theta}_1 - \dot{\theta}_2, t\right) \qquad \text{(2-8b)}$$

Here, θ_1 and θ_2 are absolute angular displacement. Equation (2-8a) and (2-8b) derived from the whole TCC system that consists of flywheel, friction shoe and pressure plate, and wheel. It represents the total torque produce by the engine that experience coulomb damping in the middle of power transmission to the friction shoe and pressure plate and finally to the wheel. The different speed rotation between the flywheel and friction shoe contributed to the T_f expression where

$T_f(\dot{\theta}_1 - \dot{\theta}_2, t)$ is the nonlinear friction torque which is a function of relative velocity $\dot{\theta}_1 - \dot{\theta}_2$ across the friction interface. The engine torque excitation $T_e(t)$ is composed of mean $T_m = \langle T_e \rangle_t$ and pulsating T_p component, where $\langle\ \rangle_t$ is the time average operator. The pulsating torque T_p generally contains many harmonic (or torque order component). However in this study considered only one term $T_m + T_p \sin(\omega t)$. Equation (3-1b) is derived from the $T_f(\dot{\theta}_1 - \dot{\theta}_2, t)$ expression which comprises from actuation pressure $P(t)$ applied to the system throughout the end of the system which is wheel or vehicle sub-system. C_{23} is lumped viscous damping and K_{23} is linear torsional stiffness associated with the automotive driveline. Since the wheel or vehicle sub-system is assumed rigid, it can be eliminated from the equation (2-8b). When relative motions are of interest, rewrite equation (2-8a) and (2-8b), where $\delta_1 = \theta_1 - \theta_2$ and $\delta_2 = \theta_2$ yields;

$$I_1\ddot{\delta}_1 - \frac{I_1}{I_2}C_{23}\dot{\delta}_2 - \frac{I_1}{I_2}K_{23}\delta_2 + \left[1.0 + \frac{I_1}{I_2}\right]T_f(\dot{\delta}_1, t) = T_m + T_p \sin(\omega t) \quad \text{(2-9a)}$$

$$I_2\ddot{\delta}_2 + C_{23}\dot{\delta}_2 + K_{23}\delta_2 = T_f(\dot{\delta}_1, t) \quad \text{(2-9b)}$$

2.2.6 Nonlinear time-varying dry friction formulation

The nonlinear friction torque $T_f(\dot{\delta}_1, t)$ is carried by the clutch and then it acts as an equivalent torque excitation to the downstream system. In a realistic automotive system, a pulse-width modulated solenoid valve would generate $P(t)$ by changing the command value or duty ratio [9]. That result in nonlinear time varying (NLTV) friction torque formulation, $T_f(\dot{\delta}_1, t) = \mu(\dot{\delta}_1)P(t)AR$ where μ is the velocity dependent coefficient of friction. Note that A and R are pressure area and moment arm and they are assumed to be time-invariant. Express the $P(t)$ profile in a form of a sinusoidal signal with a mean pressure P_m, amplitude P_p, starting time t_0 and the actuation pressure frequency ω_f:

$$P(t) = \begin{cases} 0, & t < t_{0-}, \\ P_m + P_p \sin(\omega_f t + \varphi), & t \geq t_{0+}. \end{cases} \tag{2-10}$$

Here, ψ ia the phase lag between the actuation pressure and $T_e(t)$. In this study, the initial engagement time is set at $t_{0+} = 0$ without lost of generality and $P(t)$ is assumed to be positive-definite ensure no separation occurs across the fractional interface, i.e. $P_p/P_m \in [0,1)$. The following friction formulation $\mu(\dot{\delta}_1)$ is employed, since the aim of this study is to examine phenomenological dynamic behaviour:

$$\mu(\dot{\delta}_1) = \begin{cases} \left[\mu_k + (\mu_s - \mu_k)e^{-\alpha|\dot{\delta}_1|}\right] sgn(\dot{\delta}_1), & |\dot{\delta}_1| > 0, \\ [0\ \mu_s], & \dot{\delta}_1 = 0. \end{cases} \tag{2-11}$$

Here, α is a positive constant that control the gradient of μ with the respect to $\dot{\delta}_1$; μ_s is the static coefficient; μ_k is the kinetic friction coefficient and *sgn* is the conventional triple-valued signum function. In this study, $\alpha = 2$ is chosen for the sake of illustration. To further reduce the system parameters, incorporate μ_k, A and R within $P(t)$ to yield the NLTV friction torque $T_f(\dot{\delta}_1, t)$ as;

$$T_f(\dot{\delta}, t) = \bar{\mu}(\dot{\delta}_1) T_s(t) \tag{2-12a}$$

$$T_s(t) = \mu_k A R P(t) = T_{sm} + T_{sp} \sin(\omega_f t + \varphi) \tag{2-12b}$$

Here, $\bar{\mu}(\dot{\delta}_1)$ has been normalized with the respect to μ_k. Further, the sign function of equation (2-11) is smoothened by a hyper-tangent to facilitate the numerical integration for the nonlinear stick-slip motion:

$$\bar{\mu}(\dot{\delta}_1) = \left[1.0 + \left(\frac{\mu_s}{\mu_k} - 1.0\right) e^{-\alpha|\dot{\delta}_1|}\right] \tanh(\sigma \dot{\delta}_1) \tag{2-13}$$

A value of 50 is chosen for a smoothening factor σ. Duan and Singh [3] has justified this choice by applying the same factor to a similar automotive drive train torsional system.

CHAPTER 3
METHODOLOGY

3.1 Process Flow

- Process flow for this study is depicted in Figure 3.1.

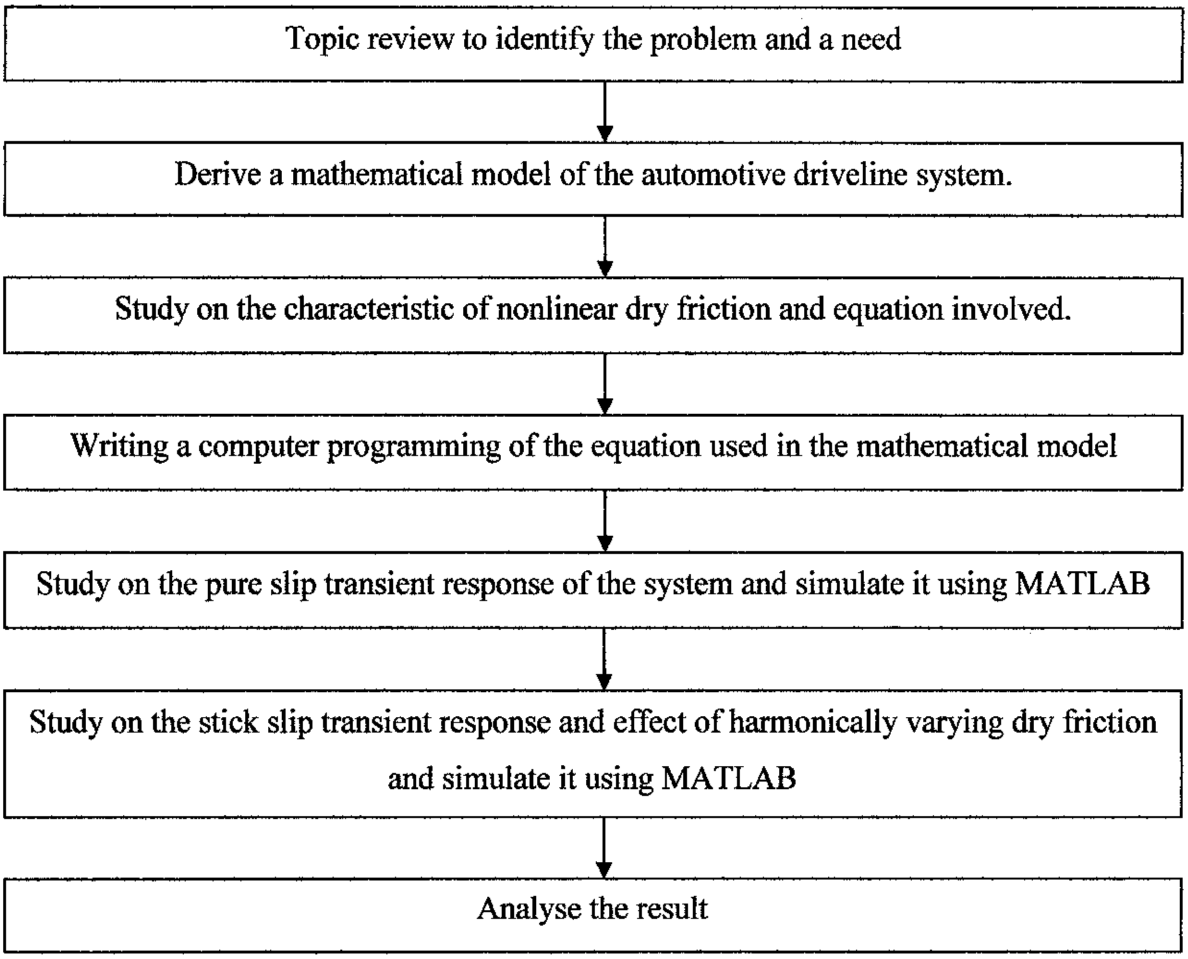


Figure 3.1: Significant step in process flow

- Gantt chart for study scheduling is depicted in Figure 3.2 and Figure 3.3.

No	Detail/ Week	1	2	3	4	5	6	7	8	9	10	11	12	13	14	SW	EW
1	Selection of Project Topic																
	-Propose Topic																
	-Topic assigned to students																
2	Preliminary Research Work																
3	Submission of Preliminary Report																
4	Project Work																
	-Reference/Literature Study																
	-Methodology																
5	Submission of Progress Report																
6	Project work continue																
7	Submission of Interim Report																
8	Oral Presentation																

SW Study Week

EW Exam Week

Figure 3.2: Gantt chart for final year project 1.

No	Detail/ Week	1	2	3	4	5	6	7	8	9	10	11	12	13	14	SW	EW
1	Project Work																
	-Programming																
	-Results Analysis																
2	Submission of Progress Report																
3	Submission of Poster																
4	Submission of Final Report																
5	Final Presentation																

SW Study Week

EW Exam Week

Figure 3.3: Gantt chart for final year project 2.

3.2 Tools and Material

- Computer Programming Software (MATLAB or FORTRAN)

3.3 Engineering Computer Programming

Computer Programming using MATLAB or FORTRAN software is required to perform the numerical simulation to investigate the characteristic of mathematical model of the automotive driveline system with torque converter clutch. The investigation must be approach methodically, applying the algorithm or step by step procedure by which one arrives at a solution. The problem solving process can be outlined as follow:

1. Define the problem
2. Create a mathematical model
3. Develop a computational method for solving a problem
4. Implement the computational method
5. Tests and assess the solution

3.4 Algorithm

To obtain the desired results, the first step is to determine the procedure of the numerical simulation. This is one of the most significant processes to ensure all experimental works completely organized. The procedures of the numerical simulation are depicted in Figure 3.4.

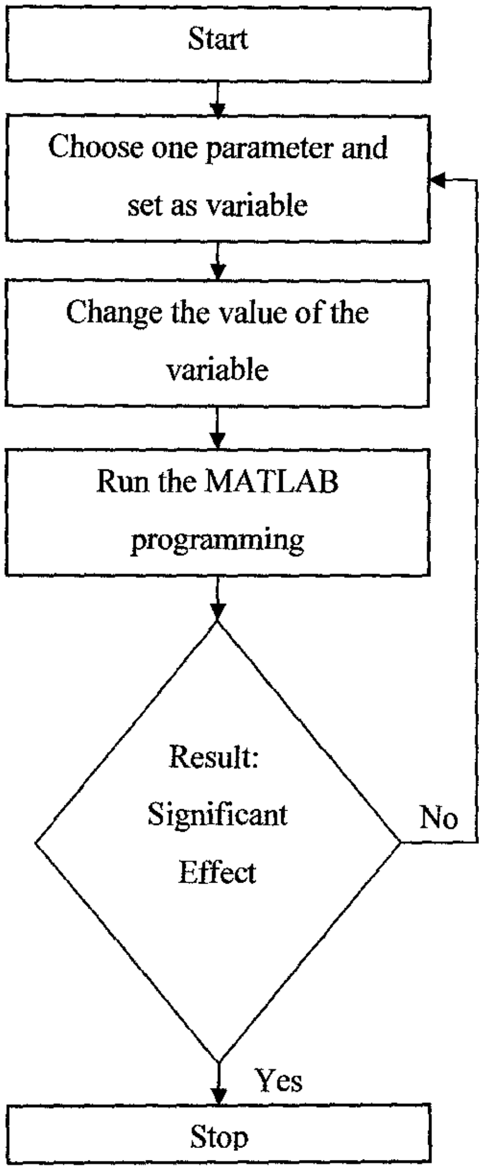


Figure 3.4: Algorithm for numerical simulation

3.5 MATLAB Programming

3.5.1 Equation Formulation

The MATLAB software is used as a computer programming tools to visualize and analyze the characteristic behaviour of the transient response of a 2-DOF torsional system with nonlinear dry friction element under sinusoidal torque excitation. To solve the nonlinear second order ODE differential equation derived from the mathematical modelling, the fourth-order Runge-Kutta method is chosen. The Runge-Kutta method is one of the numerical methods is used in computational method to solve the problem in engineering and science that can be formulated in terms of ordinary differential equations satisfying certain given condition. The method is a good choice for common purposes because it is accurate, precise and easy to program compared to other method such as Euler's method and Heun's method. The fourth order Runge-Kutta method (RK4) simulates the accuracy of the Taylor series method of order N=4. The method is based on computing y_{i+1} as follows [10]:

$$y_{i+1} = y_i + 1/6(K_1 + 2K_2 + 2K_3 + K_4)h \tag{3-1}$$

Where K_1, K_2, K_3 and K_4 have the form;

$$K_1 = f(x_i, y_i)$$

$$K_2 = f(x_i + 0.5h, y_i + 0.5K_1h)$$

$$K_3 = f(x_i + 0.5h, y_i + 0.5K_2h)$$

$$K_4 = f(x_i + h, y_i + K_3h)$$

To apply fourth order Runge-Kutta method in the system equation derived from the mathematical modelling, the first step is to develop equation (2-9a) and (2-9b) in matrix form.

$$\begin{bmatrix} I_1 & 0 \\ 0 & I_2 \end{bmatrix}\begin{bmatrix} \ddot{\delta}_1 \\ \ddot{\delta}_2 \end{bmatrix} + \begin{bmatrix} 0 & -\frac{I_1}{I_2}C_{23} \\ 0 & C_{23} \end{bmatrix}\begin{bmatrix} \dot{\delta}_1 \\ \dot{\delta}_2 \end{bmatrix} + \begin{bmatrix} 0 & -\frac{I_1}{I_2}K_{23} \\ 0 & K_{23} \end{bmatrix}\begin{bmatrix} \delta_1 \\ \delta_2 \end{bmatrix}$$

$$= \begin{bmatrix} T_m + T_p sin(\omega t) - \left[1.0 + \frac{I_1}{I_2} \right](T_{sm} + T_{sp}\sin(\omega_f t + \varphi)) \\ T_{sm} + T_{sp}\sin(\omega_f t + \varphi) \end{bmatrix} \tag{3-2}$$

Next each element in the matrix form of equation (3-2) is then express in alphabet to simplify the second order differential equation calculation.

$$[M][\ddot{y}] + [C][\dot{y}] + [K][y] = [F(t)] \tag{3-3}$$

The equation (3-3) represent a mechanical system in which a spring constant K restores a displaced mass (m). Damping is assumed to be proportional to the velocity, and the function $F(t)$ is an external forces. The second order differential equation can be reformulated as a system of two first order equations by using substitution.

$$[Z] = [\dot{y}]$$

Then $\ddot{y} = \dot{Z}$, rearrange equation (3-3).

$$[\dot{Z}] = [M]^{-1}\,[[F(t)] - [C][Z] - [K][y]] \tag{3-4}$$

The concern is on two variables which is $\dot{Z}$ and $\dot{y}$. Creates one new matrix function [u] with Z and y as variables. Rearrange equation (3-4) by placed matrix function [u] on the left hand side and the rest of the equation on the right hand side.

$$[u] = \begin{bmatrix} y \\ Z \end{bmatrix}$$

$$[\dot{u}] = \begin{bmatrix} \dot{y} \\ \dot{Z} \end{bmatrix}$$

Substitute $[Z] = [\dot{y}]$, and equation (3-4) into $[\dot{u}]$.

$$[\dot{u}] = \begin{bmatrix} \dot{y} \\ \dot{z} \end{bmatrix} = \begin{bmatrix} [Z] \\ [M]^{-1}[[F(t)] - [C][Z] - [K][y]] \end{bmatrix} \tag{3-5}$$

Rearrange equation (3-5) into first order differential equation yield,

$$[\dot{u}] = \begin{bmatrix} [0] & [1] \\ -[M]^{-1}[K] & -[M]^{-1}[C] \end{bmatrix} [u] + -[M]^{-1}F(t) \tag{3-6}$$

A numerical procedure used is Runge-Kutta method which solved equation (3-6) computationally using MATLAB software.

3.5.2 Computer Code

The algorithm in programming 2-DOF torsional system with nonlinear dry friction under harmonically varying normal load can be classified into 5 parts. The first part is the main body of the program, the second part is the function of governing equation of the system, the third part is the external force function, the fourth part is normalized friction formulation and the fifth part is nonlinear friction torque function.

- Main Function

```
clear all
I1=0.2;
I2=0.02;
K23=3000;
C23=0.6;
Tsm=500;
Tsp=0.2*Tsm;
wf=150;
psi=0;
Tm=300;
Tp=250;
w=150;

tstart=0;
d1=0;
d2=0;
d1dot=150;
d2dot=0;
% y0(1:2,1:1)=0;
```

```
% y1(1:2,1:1)=0;
% y2(1:2,1:1)=0;
% y3(1:2,1:1)=0;
% y4(1:2,1:1)=0;
% y0(1,1)=d1;
% y0(2,1)=d2;
u0(1:4,1:1)=0;
u1(1:4,1:1)=0;
u2(1:4,1:1)=0;
u3(1:4,1:1)=0;
u4(1:4,1:1)=0;
u0(1,1)=d1;
u0(2,1)=d2;
u0(3,1)=d1dot;
u0(4,1)=d2dot;
% making m
m(1:2,1:2)=0;
m(1,1)=I1;
m(2,2)=I2;
%making c
c(1:2,1:2)=0;
c(1,2)=(-I1/I2)*C23;
c(2,2)=C23;
% making k
k(1:2,1:2)=0;
k(1,2)=(-I1/I2)*K23;
k(2,2)=K23;
%making A
A(1:4,1:4)=0;
A(1,3)=1;
A(2,4)=1;
A(3:4,1:2)=-inv(m)*k;
A(3:4,3:4)=-inv(m)*c;

tend=0.3;
h=0.00001;
nstep=(tend-tstart)/h;
tn(1:nstep)=0;
u1n(1:nstep)=0;
u2n(1:nstep)=0;
u3n(1:nstep)=0;
u4n(1:nstep)=0;

itime=1;
tn(itime)=tstart;
u1n(itime)=u0(1,1);
u2n(itime)=u0(2,1);
u3n(itime)=u0(3,1);
u4n(itime)=u0(4,1);
t=tstart;
%integration
for n=1:1☹nstep-1)
   u1=u0;
   d1dot=u1(3,1);
   [Ft]=f1(t,Tm,Tp,I1,I2,Tsm,Tsp,wf,w,psi,d1dot);
   [Fx]=fx(A,u1,m,Ft);
   K1=h*Fx;

   t2=t+h/2;
   u2=u0+0.5*K1;
   d1dot=u2(3,1);
   [Ft]=f1(t2,Tm,Tp,I1,I2,Tsm,Tsp,wf,w,psi,d1dot);
   [Fx]=fx(A,u2,m,Ft);
   K2=h*Fx;

   t3=t+h/2;
   u3=u0+0.5*K2;
   d1dot=u3(3,1);
   [Ft]=f1(t3,Tm,Tp,I1,I2,Tsm,Tsp,wf,w,psi,d1dot);
   [Fx]=fx(A,u3,m,Ft);
   K3=h*Fx;

   t4=t+h;
   u4=u0+K3;
   d1dot=u4(3,1);
```

```
[Ft]=f1(t4,Tm,Tp,I1,I2,Tsm,Tsp,wf,w,psi,d1dot);
[Fx]=fx(A,u4,m,Ft);
K4=h*Fx;

t=t+h;
u0=u0+(K1+2*K2+2*K3+K4)/6;
itime=itime+1;
tn(itime)=t;
u1n(itime)=u0(1,1);
u2n(itime)=u0(2,1);
u3n(itime)=u0(3,1);
u4n(itime)=u0(4,1);

end
plot (tn,u3n)
```

- Equation (15e) Function

```
function [Fx]=fx(A,u,m,Ft)
Fx(1:4,1:1)=0;
Fx(3:4,1:1)=inv(m)*Ft;
Fx=A*u+Fx;
```

- External Force Function

```
function [Ft]=f(t,Tm,Tp,I1,I2,Tsm,Tsp,wf,w,psi)
Ft(1:2,1:1)=0;
Ft(1,1)=Tm+Tp*sin(w*t)-(1.0+I1/I2)*(Tsm+Tsp*sin(wf*t+psi));
Ft(2,1)=Tsm+Tsp*sin(wf*t+psi);
```

- $\bar{\mu}\left(\dot{\delta}_1\right)$ Function

```
function [myu]=myubar(myu_s,myu_k,alpha,d1dot,sigma)

if abs(d1dot)>0
 myu=(1+(myu_s/myu_k-1)*exp(-alpha*abs(d1dot)))*tanh(sigma*d1dot);
else
  myu=myu_s;
end
```

- $T_f\left(\dot{\delta},t\right)$ Function

```
function [tfdt]=ttf(d1dot,t,wf,psi)
alpha=2;
sigma=50;
A=0.08;
R=0.1;
myu_s=0.3;
myu_k=0.24;
pm=260000;
pp=0.2*pm;
myu=myubar(myu_s,myu_k,alpha,d1dot,sigma);
ts=myu_k*A*R*(pm+pp*sin(wf*t+psi));
tfdt=myu*ts;
```

CHAPTER 4

RESULT AND DISCUSSION

4.1 Transient Response for 2-DOF Torsional System

To analyze the characteristic of the transient response of the system, the typical key parameters of the system is first define. The key parameters are obtained from the previous study that applied the same 2-DOF torsional system with nonlinear dry friction element in steady state response [3]. The typical key parameter is depicted in Table 4.1.

Table 4.1: Key parameter used for 2-DOF system

Parameters and excitation	Values
Torsional inertias (kgm^2)	$I_1 = 0.20$, $I_2 = 0.02$
Torsional viscous damping (Nm rad/s)	$C_{23} = 0.6$
Torsional stiffness (Nm/rad)	$K_{23} = 3000$
Engine torque: mean and excitation amplitude (Nm)	$T_m = 300Nm$, $T_{sm}=500Nm$
Static and kinematic friction coefficients	$\mu_s = 0.3$, $\mu_k = 0.15\text{-}0.3$
Actuation pressure area (m^2)	$A = 0.08$
Moment arm (m)	$R = 0.1$
Actuation pressure (kPa)	$P_m = 260000$, $P_p = 0.2\ P_m$
Phase lag (radian)	$\Psi = 0, \pi/2, \pi$

In this study, only relative velocity ($\dot{\delta}_1$) and the pure positive slip motions ($\dot{\delta}_1>0$) is considered since the negative gradient dominates only when $\dot{\delta}_1$ approach zero. Thus equation (2-9a) and (2-9b) can be approximated as:

$$I_1\ddot{\delta}_1 - \frac{I_1}{I_2}C_{23}\dot{\delta}_2 - \frac{I_1}{I_2}K_{23}\delta_2 + \left[1.0 + \frac{I_1}{I_2}\right](T_{sm} + T_{sp}\sin(\omega_f t + \varphi)) = T_m + T_p sin(\omega t) \tag{4-1}$$

$$I_2\ddot{\delta}_2 + C_{23}\dot{\delta}_2 + K_{23}\delta_2 = T_{sm} + T_{sp}\sin(\omega_f t + \varphi) \tag{4-2}$$

The result produced from the MATLAB programming based on the governing equation under following initial given condition based on previous study in steady state response [3].

$$\delta_1 = 0, \quad \dot{\delta}_1 = \Omega_e = 150./s, \qquad \delta_2 = 0, \quad \dot{\delta}_2 = 0,$$

The parameters used for the reduced driveline system of Figure 4.1 based on the Table 1.The typical transient response for 2-DOF torsional system given a condition at $\omega = \omega_f$ where ω=150./s, T_{sm}=500Nm, and $T_{sp}=0.2T_{sm}$ is shown in Figure 4.1 obtained from the MATLAB programming.

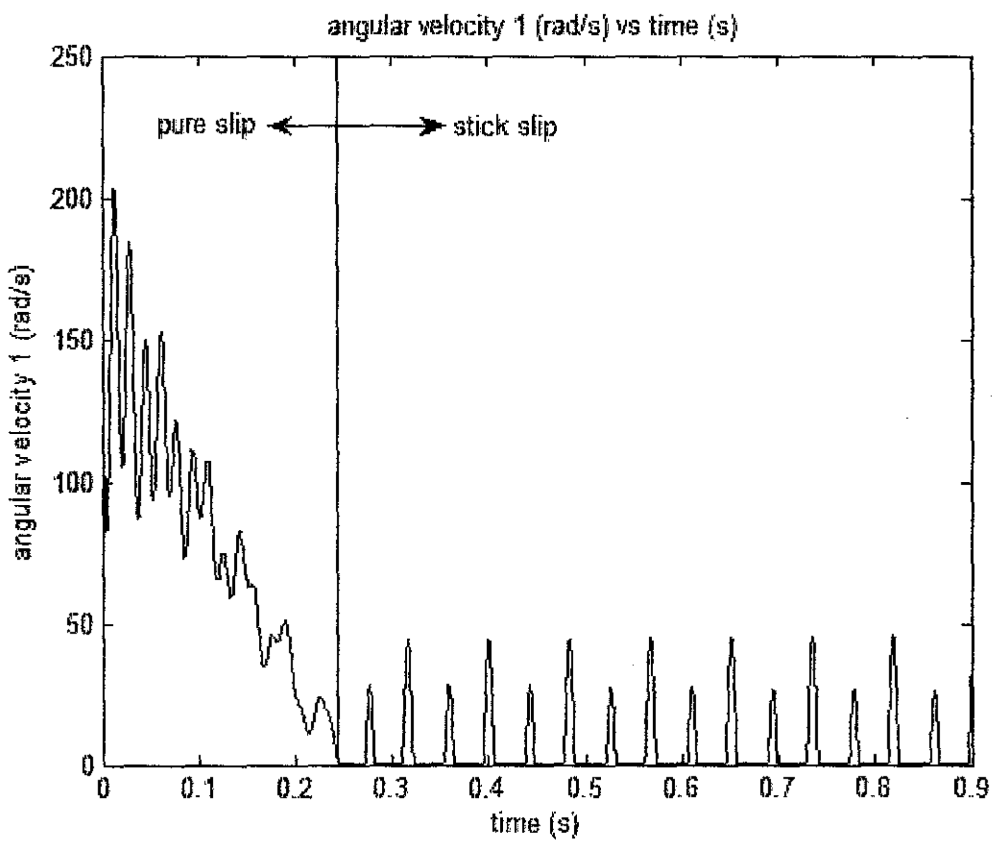


Figure 4.1: Transient response (pure slip and stick slip) of 2-DOF system.

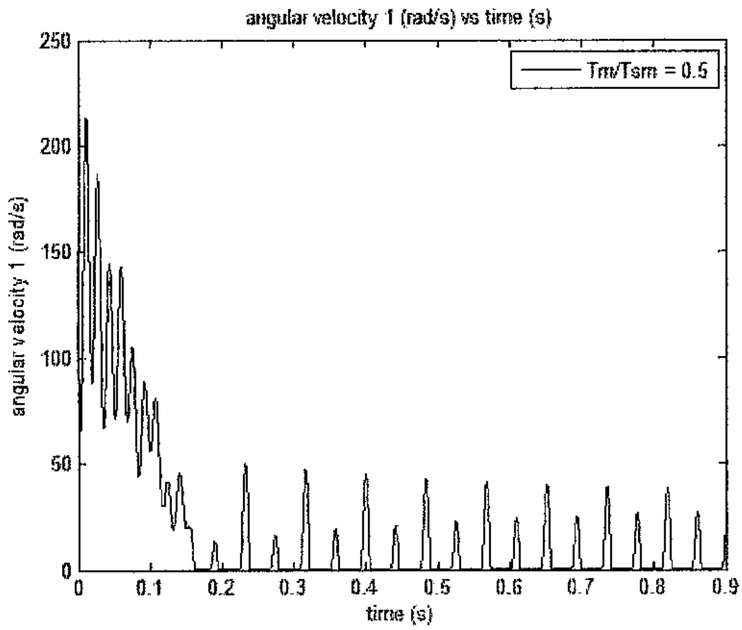
From Figure 4.1 observations, the transient response of the system can be categorized into two types of response which is time dependent during initial engagement. Those two responses are pure slip response and stick-slip response. Both of this response will discuss further in next sub-chapter. From the results shown in Figure 4.1, the pure slip transient response graph profile is not stable, inconsistence and fluctuates with time as angular velocity 1 continuously decaying with time from its initial condition at 150./s until its approach zero at approximately t=0.25. As the system approach stick slip transient response, the graph profile shown that the system experience more stable and consistence condition though in a fluctuate motion.

4.1.1 Pure Slip Transient Response

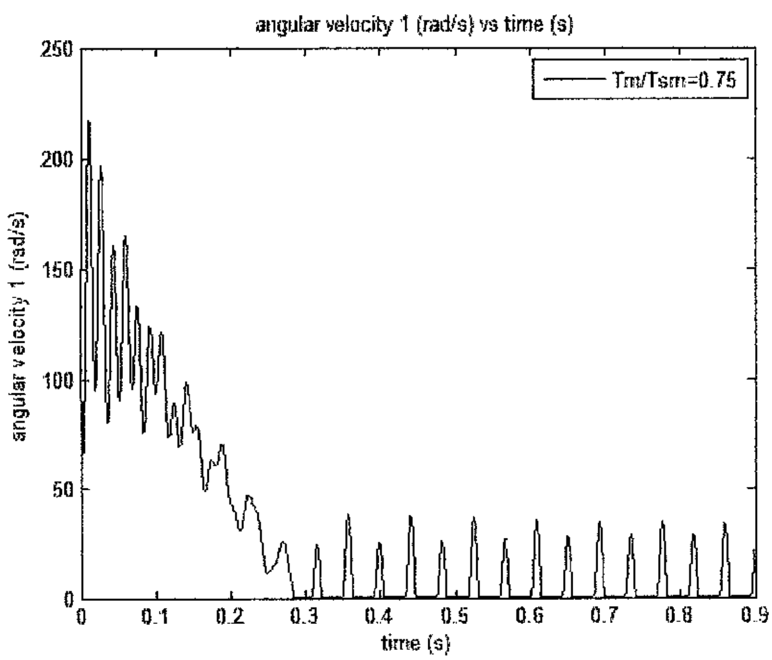
In pure slip transient response, the system is trying to achieve engagement of two surfaces with different angular velocity between them. Pure slip transient response results due to a speed different of I_1 and I_2 during initial engagement. During initial engagement, I_1 rotates at the same speed as the engine while I_2 and the rest of driveline system stays still until any engagement process can take place. Theoretically, the rotation speed of the both surface should be equal as they are engaged to each other. Both of the surfaces should be adhere to each other thus results in no anticipation of angular velocity gradient. Observation from the Figure 4.1, the initial angular velocity of engine is 150./s, while the rest of the driveline system is in idle position. During the clutch engagement, the rotation speed of the first surface exceeds the other one due to the intermittent change between kinetic friction (μ_k) and static friction (μ_s) thus results in slipping motion which creates the difference in angular speed between two surfaces. Furthermore, the system experiencing a continuous decaying slip as angular velocity 2 approach angular velocity 1. The significant of the pure slip transient response reduces as the system try to achieve equilibrium state between two surfaces.

The aim of investigation on pure slip transient response is to determine the parameters that give a significant effect and control on clutch engagement rate. Certain parameters including the ratio of engine torque excitation composed of mean T_m to the saturation friction torque T_{sm} and engine torsional inertia is identified producing significant results on the clutch engagement rate. Those parameters are identified by conducting several simulations on each parameter involved in the governing equation of the system.

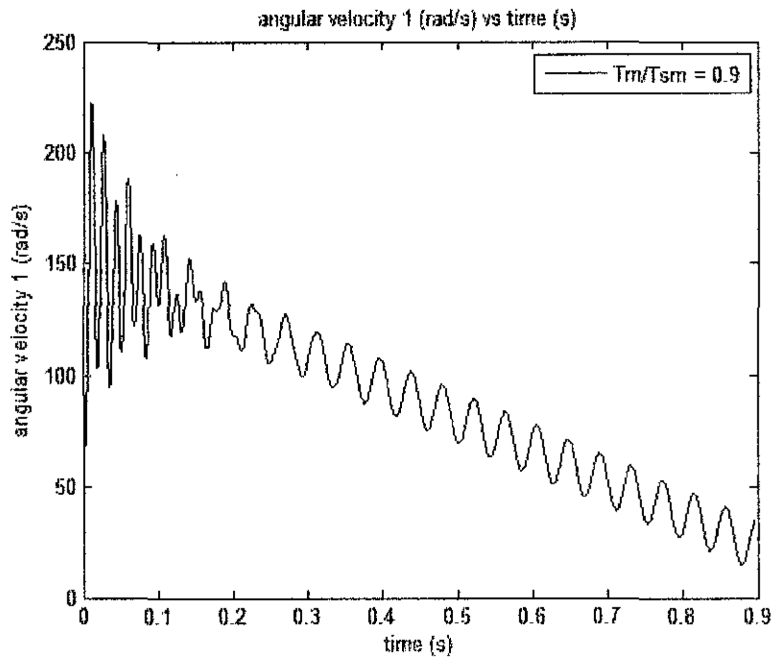
Various Graph Profile of Ratio of T_m / T_{sm}



(a)



(b)



(c)

Figure 4.2: Various graph profile of T_m / T_{sm} ratio.

The ratio T_m / T_{sm} provides a significant effect on the clutch engagement of the system in pure slip transient response or in other words the clutch engagement is controlled by the ratio of T_m / T_{sm}. Figure 4.2 verify that the clutch engagement of 2-DOF torsional system under pure slip transient response influence by the ratio of T_m / T_{sm}. For the sake of illustration, the results is distinguished by three different graph depicted in Figure 4.2. Figure 4.2(a) shows that when ratio of T_m / T_{sm} = 0.5, time taken to realize clutch engagement is approximately 0.17 second. An increase of ratio T_m / T_{sm} from 0.5 to 0.75 in Figure 4.2(b) indicate that time required to realize clutch engagement is approximately 0.28 second. Furthermore, in Figure 4.2(c) indicates that time required to realize clutch engagement is exceed 0.9 second at T_m / T_{sm} = 0.9. Comparison between three graphs in Figure 4.2 shows that an increase in ratio of T_m / T_{sm} results in increasing of time to realize final clutch engagement.

Effect of T_{sm}

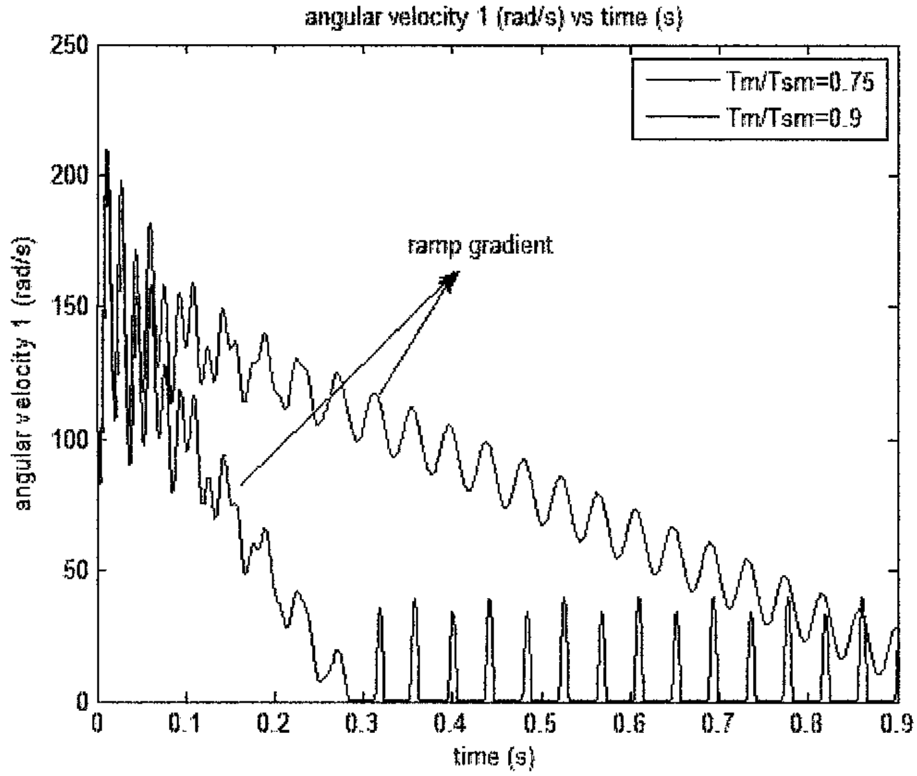


Figure 4.3: Effect of T_{sm} on pure slip transient

Under the situation when amplitude of the oscillatory part is small and when the decaying component dies out quickly, the clutch engagement rate is controlled by a ramp of gradient. As shown in the Figure 4.3, when T_m / T_{sm} is increased from 0.75 to 0.9, the ramp gradient decreases. This can be proofed by the calculation below.

Ramp of gradient a = $(T_m - T_{sm}) / I_1$

$T_m / T_{sm} = 0.75$,

a = (375 – 500) / 0.2

a = -625

$T_m / T_{sm} = 0.9$,

a = (450 – 500) / 0.2

a = -250

Effect of Inertia

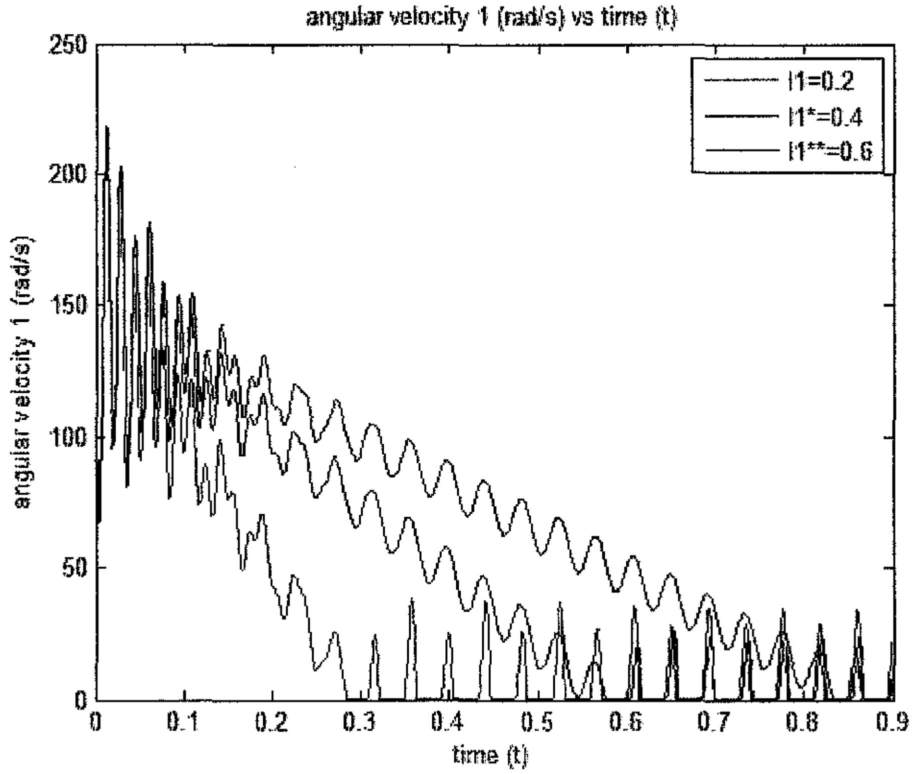


Figure 4.4: Effect of inertia on pure slip transient

As shown in Figure 4.4, an increase in I_1=0.2 to I_1*=0.4 and I_1*=0.4 to I_1**=0.6 indicates more kinetic energy involves thus lead the dissipation process to take longer time.

Effect of T_{sm} on the Clutch Engagement

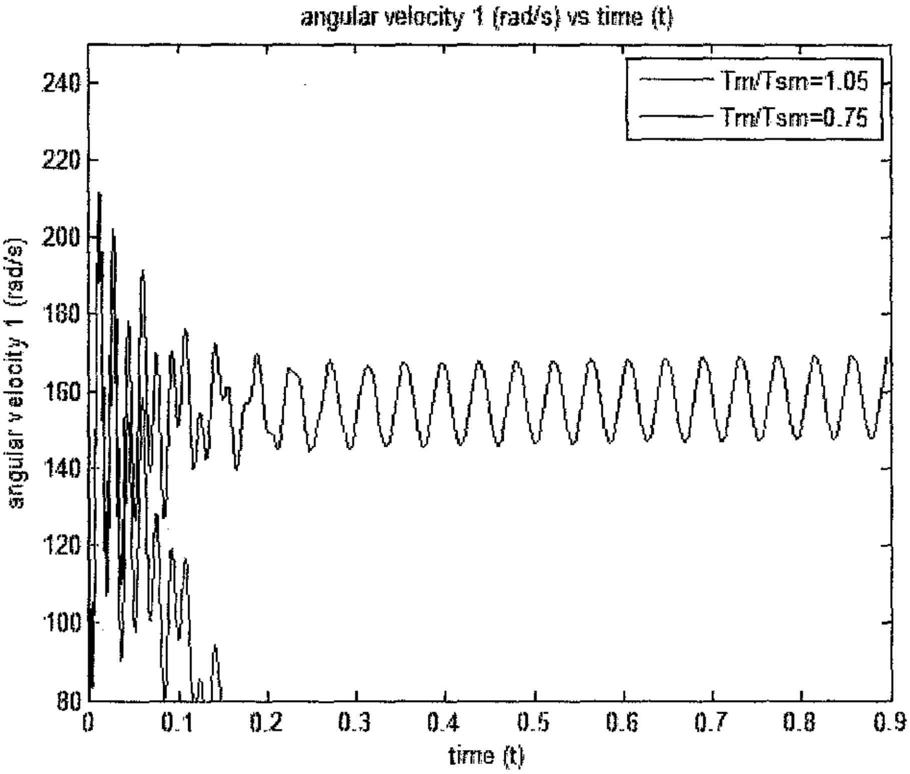


Figure 4.5: Effect of T_{sm} on clutch engagement

As shown in Figure 4.5, when T_m is higher than T_{sm}, no final engagement would take place. This because of the ramp of gradient rate is monotonically grown with time and dominates the response. However, when T_m is lower than T_{sm}, final engagement can be realized as the system trying to archived stick slip transient response in a steady state phase.

4.1.2 Stick-slip Transient Response (Judder)

In stick-slip transient response, investigation is focused on the parameter that produces a significant effect on the stick-slip response characteristic of the 2-DOF torsional system. The phenomenon of stick-slip torsional motion is usually referred to as the clutch judder problem that typically occurs at low frequencies. The low frequencies are cause by the friction surfaces that rub concentrically. In addition to objectionable noise problem, significant stick slip torsional motion could be transferred by the differential to the in the form fore-after jerk. Sometimes this problem is called negative damping introduced by the negative slope of μ_k ($\dot{\delta}_1$). Stick-slip response occurs when $\dot{\theta}_2$ approach $\dot{\theta}_1$, subsequent to the pure slip motion as discussed above. This phenomenon is caused by the external torsional load which is higher and results in higher value of kinetic friction (μ_k). Thus it allows the slip motion in stick condition. Based on the experimental results, there are three parameters identified that pronounced the stick-slip response namely kinetic friction (μ_k), saturation friction torque (T_{sm}) and phase lag (ψ). As stick slip investigation only concern on the angular velocity 1 ($\dot{\delta}_1$) with respect to time, the stick slip influence is determine by the time required for the clutch to slip simultaneously in stick condition which represented by the width of protrusion emerge in the graph profile.

Effect of Kinetic Friction (μ_k) on the 2-DOF torsional system

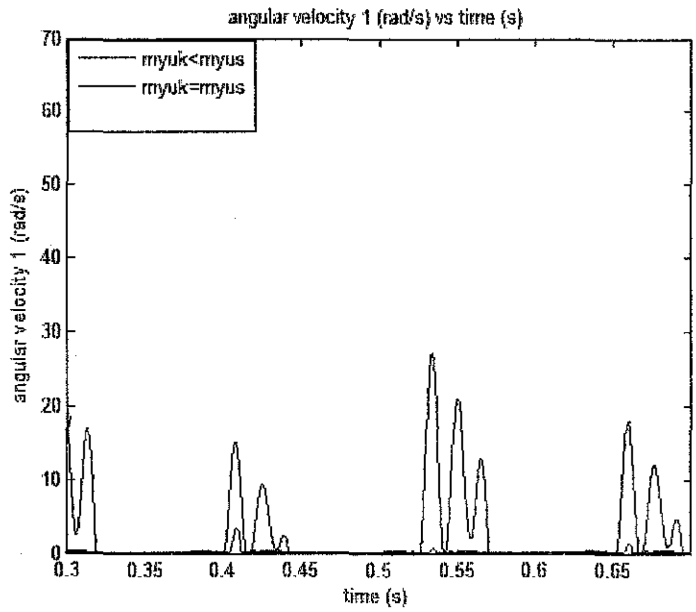


Figure 4.6: Effect of kinetic friction (μ_k) on the 2-DOF torsional system at ω = 50./s and T_{sm} = 500Nm.

First, the effect of kinetic friction on the stick slip transient response is investigated by setting constant saturation friction torque T_{sm} to the value of 500Nm with both kinetic and static friction is setup to be variable as shown in Figure 4.6. From the friction formulation of equation (2-13), a decrease in kinetic friction (μ_k) with angular velocity 1 ($\dot{\delta}_1$) affect the system in two ways. First, a negative slope regime is formed. Second, the saturation friction torque $T_{sm}= \mu_k ARP_m$ is reduced thus attenuate slip motion. Figure 4.6 shows results for two values of μ_k. As expected, the stick slip motion become more pronounced when $\mu_k=\mu_s$ because the negative damping enhances the slip motion. Conversely, when $\mu_k<\mu_s$ the stick slip motion are attenuated as a result of a positive damping as well as a lower value of T_{sm}. The summary of μ_k and T_{sm} that influence the characteristic of stick slip motion is tabulated in Table 4.2.

Table 4.2: Influence of μ_k and T_{sm} on stick slip motion.

condition	T_{sm}	slip motion	explanation
$\mu_k<\mu_s$	low	reduced	Reduced T_{sm} attenuate the slip motion.
$\mu_k=\mu_s$	high	increase	Negative slope introduced and pronounced the slip motion

Effect of Reduced T_{sm} and Negative Slope

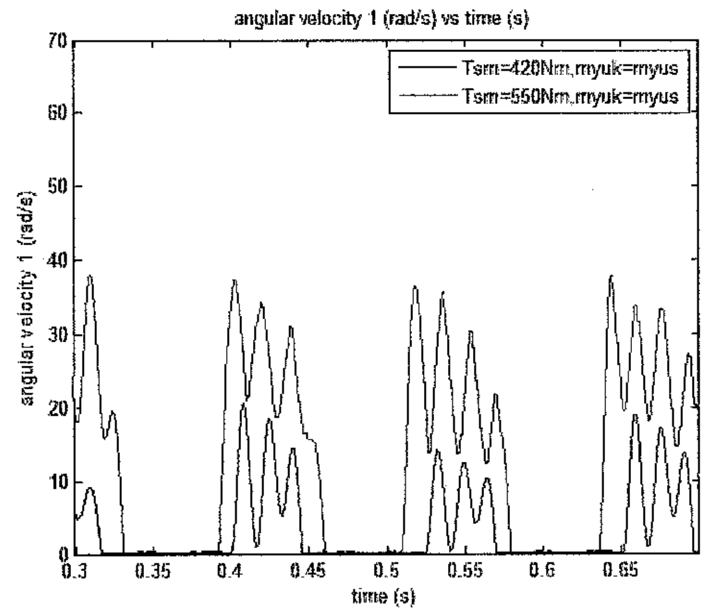
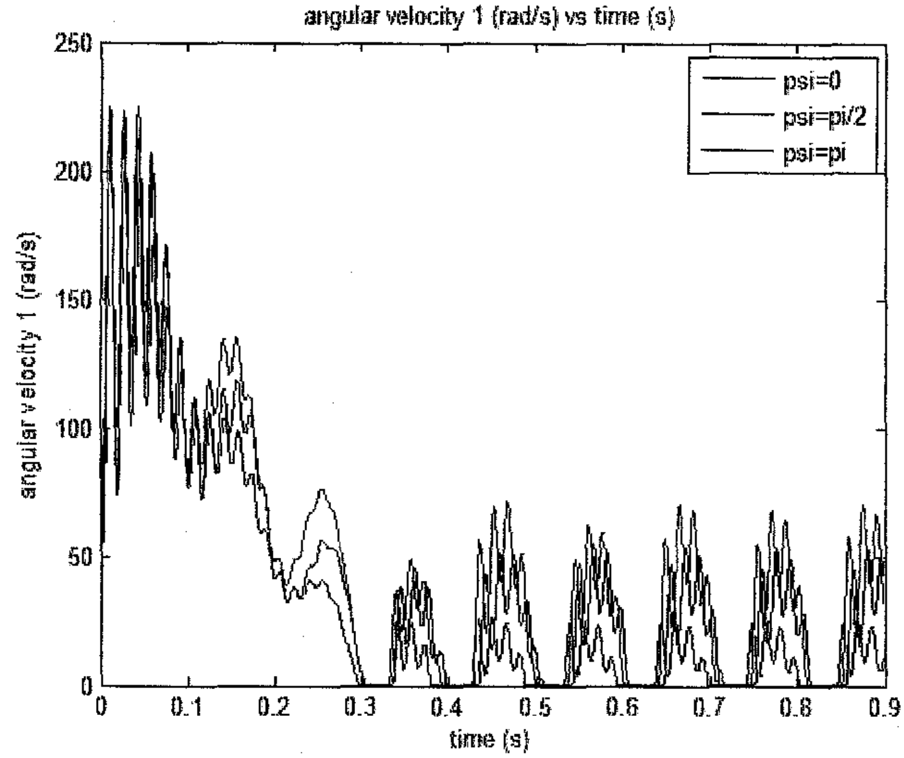


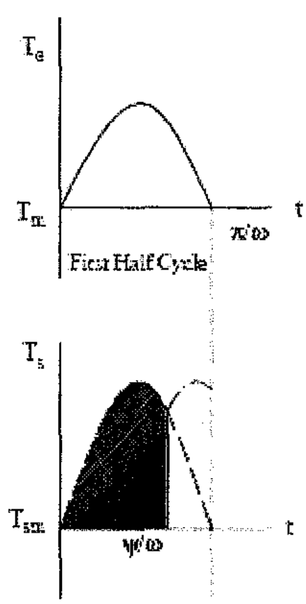
Figure 4.7: Effect of reduced T_{sm} and negative slope in friction on 2-DOF torsional system when excited at $\omega = 50./s$

To illustrate the effect of the negative slope more clearly, Figure 4.7 compares the result for two cases of T_{sm} and μ_k. Here, the static friction and kinetic friction is held constant, $\mu_k=\mu_s$ for both cases. At the same time, the value for T_{sm} is increase from 420Nm to 550Nm to illustrate the significant effect of negative slope. Note that although reduce in T_{sm} would attenuate the slip motion as discussed before, the introduction of negative slope characteristic could further amplify the slip motion which results in stick slip motion in a condition where T_{sm} is relatively lower.

Effect of Phase Lag



(a)



(b)

Figure 4.8: (a) Effect of phase lag given ω=60. (b) Quasi-static explanation of the effect of harmonically varying friction torque. (ω= ω_f): blue solid line $T_e(t)$;black dotted line $T_s(t)$ in phase; red dotted line $T_s(t)$ not in phase.

The result produced from the MATLAB programming based on the governing equation under following initial given condition based on previous study in steady state response [3].

$$\delta_1 = 0, \quad \dot{\delta}_1 = \Omega_e = 150./s, \qquad \delta_2 = 0, \quad \dot{\delta}_2 = 0,$$

The parameters used for the reduced driveline system of Figure 4.1 based on the Table 1.The typical transient response for 2-DOF torsional system given a condition at $\omega = \omega_f$ where ω=150./s, T_{sm}=500Nm, and T_{sp}=0.2T_{sm} is shown in Figure 4.1 obtained from the MATLAB programming.

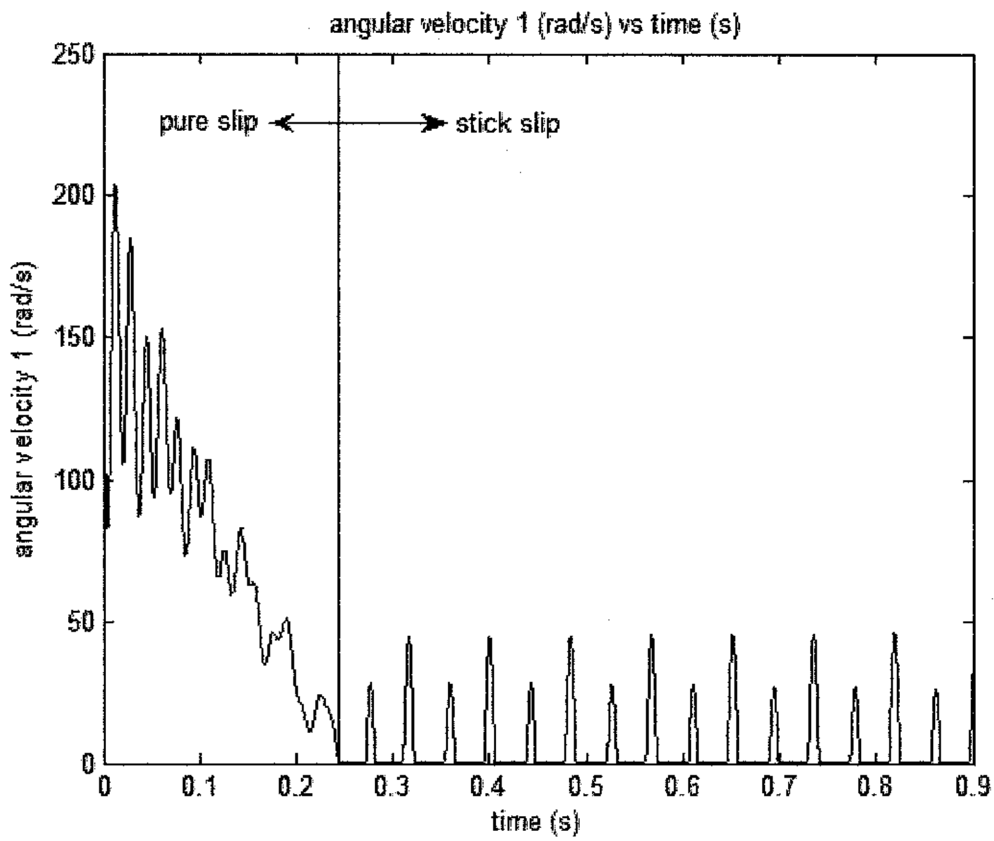


Figure 4.1: Transient response (pure slip and stick slip) of 2-DOF system.

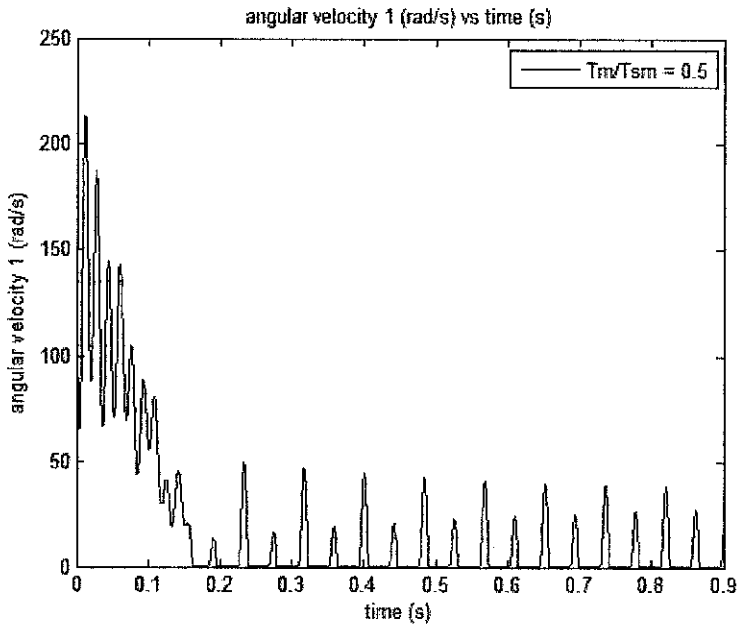
From Figure 4.1 observations, the transient response of the system can be categorized into two types of response which is time dependent during initial engagement. Those two responses are pure slip response and stick-slip response. Both of this response will discuss further in next sub-chapter. From the results shown in Figure 4.1, the pure slip transient response graph profile is not stable, inconsistence and fluctuates with time as angular velocity 1 continuously decaying with time from its initial condition at 150./s until its approach zero at approximately t=0.25. As the system approach stick slip transient response, the graph profile shown that the system experience more stable and consistence condition though in a fluctuate motion.

4.1.1 Pure Slip Transient Response

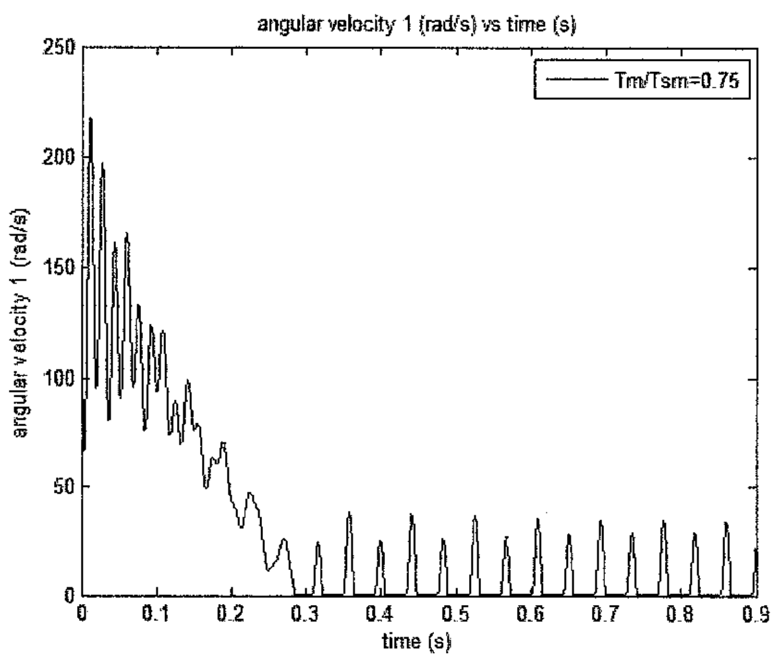
In pure slip transient response, the system is trying to achieve engagement of two surfaces with different angular velocity between them. Pure slip transient response results due to a speed different of I_1 and I_2 during initial engagement. During initial engagement, I_1 rotates at the same speed as the engine while I_2 and the rest of driveline system stays still until any engagement process can take place. Theoretically, the rotation speed of the both surface should be equal as they are engaged to each other. Both of the surfaces should be adhere to each other thus results in no anticipation of angular velocity gradient. Observation from the Figure 4.1, the initial angular velocity of engine is 150./s, while the rest of the driveline system is in idle position. During the clutch engagement, the rotation speed of the first surface exceeds the other one due to the intermittent change between kinetic friction (μ_k) and static friction (μ_s) thus results in slipping motion which creates the difference in angular speed between two surfaces. Furthermore, the system experiencing a continuous decaying slip as angular velocity 2 approach angular velocity 1. The significant of the pure slip transient response reduces as the system try to achieve equilibrium state between two surfaces.

The aim of investigation on pure slip transient response is to determine the parameters that give a significant effect and control on clutch engagement rate. Certain parameters including the ratio of engine torque excitation composed of mean T_m to the saturation friction torque T_{sm} and engine torsional inertia is identified producing significant results on the clutch engagement rate. Those parameters are identified by conducting several simulations on each parameter involved in the governing equation of the system.

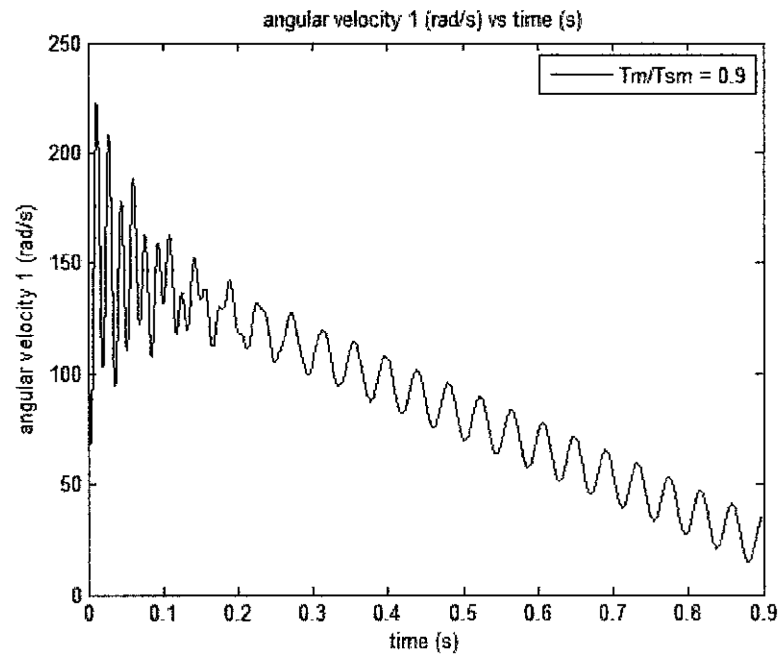
Various Graph Profile of Ratio of T_m / T_{sm}



(a)



(b)



(c)

Figure 4.2: Various graph profile of T_m / T_{sm} ratio.

The ratio T_m / T_{sm} provides a significant effect on the clutch engagement of the system in pure slip transient response or in other words the clutch engagement is controlled by the ratio of T_m / T_{sm}. Figure 4.2 verify that the clutch engagement of 2-DOF torsional system under pure slip transient response influence by the ratio of T_m / T_{sm}. For the sake of illustration, the results is distinguished by three different graph depicted in Figure 4.2. Figure 4.2(a) shows that when ratio of T_m / T_{sm} = 0.5, time taken to realize clutch engagement is approximately 0.17 second. An increase of ratio T_m / T_{sm} from 0.5 to 0.75 in Figure 4.2(b) indicate that time required to realize clutch engagement is approximately 0.28 second. Furthermore, in Figure 4.2(c) indicates that time required to realize clutch engagement is exceed 0.9 second at T_m / T_{sm} = 0.9. Comparison between three graphs in Figure 4.2 shows that an increase in ratio of T_m / T_{sm} results in increasing of time to realize final clutch engagement.

Effect of T_{sm}

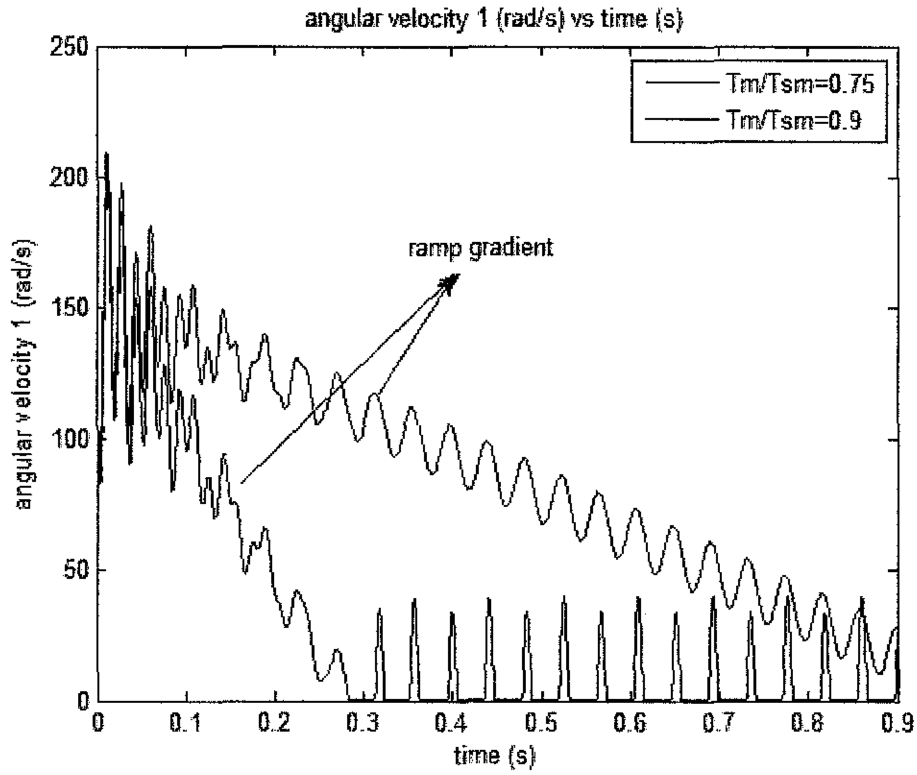


Figure 4.3: Effect of T_{sm} on pure slip transient

Under the situation when amplitude of the oscillatory part is small and when the decaying component dies out quickly, the clutch engagement rate is controlled by a ramp of gradient. As shown in the Figure 4.3, when T_m / T_{sm} is increased from 0.75 to 0.9, the ramp gradient decreases. This can be proofed by the calculation below.

Ramp of gradient a = $(T_m - T_{sm}) / I_1$

$T_m / T_{sm} = 0.75$,

a = (375 – 500) / 0.2

a = -625

$T_m / T_{sm} = 0.9$,

a = (450 – 500) / 0.2

a = -250

Effect of Inertia

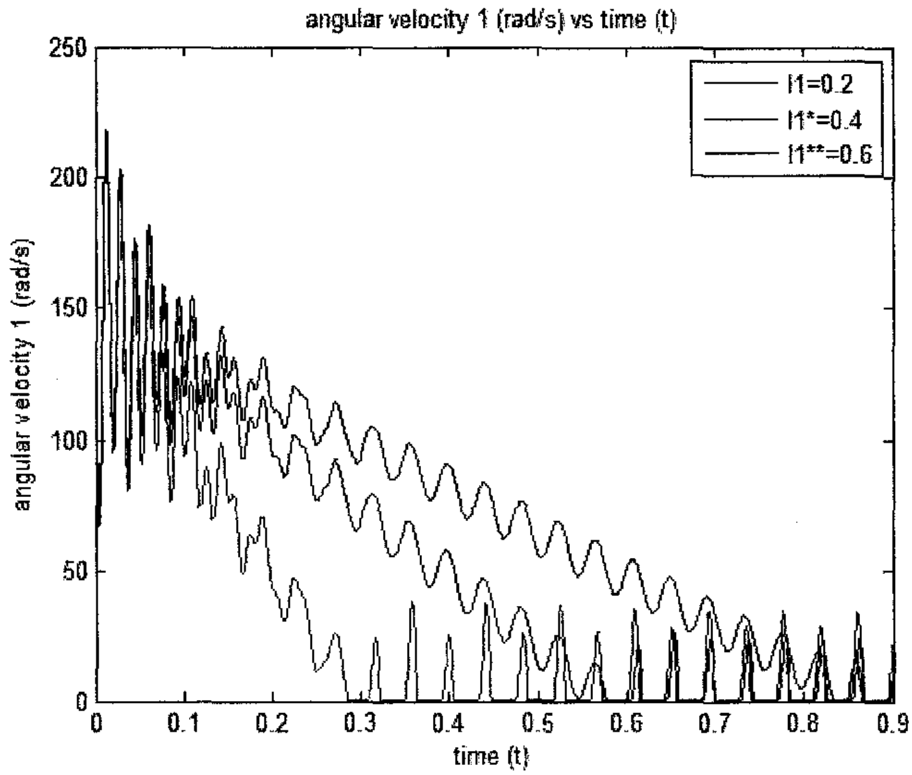


Figure 4.4: Effect of inertia on pure slip transient

As shown in Figure 4.4, an increase in I_1=0.2 to I_1*=0.4 and I_1*=0.4 to I_1**=0.6 indicates more kinetic energy involves thus lead the dissipation process to take longer time.

Effect of T_{sm} on the Clutch Engagement

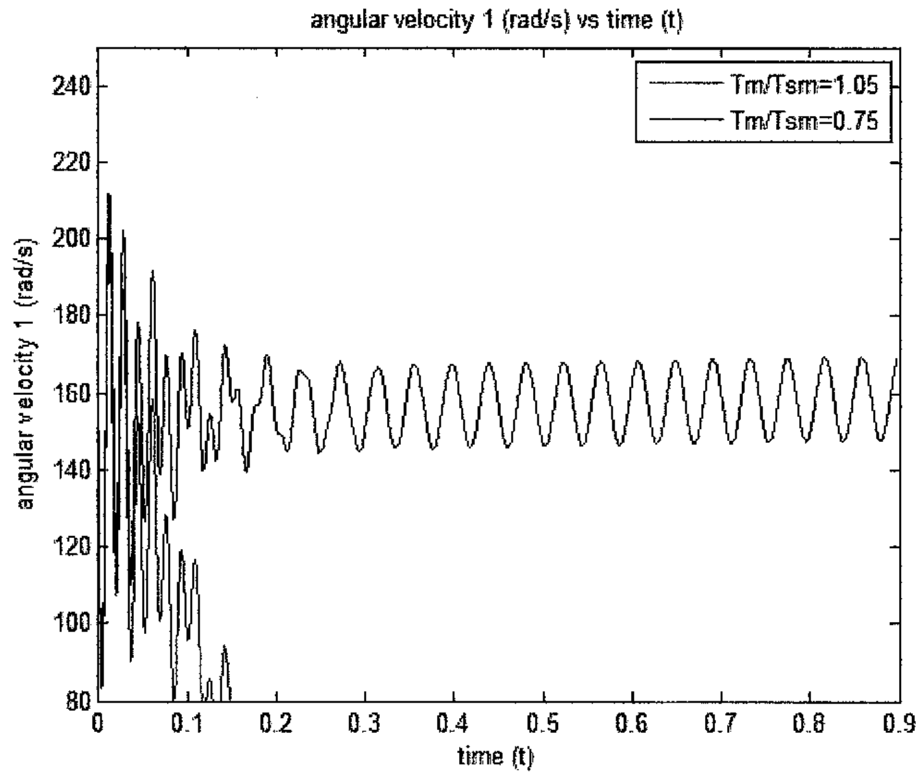


Figure 4.5: Effect of T_{sm} on clutch engagement

As shown in Figure 4.5, when T_m is higher than T_{sm}, no final engagement would take place. This because of the ramp of gradient rate is monotonically grown with time and dominates the response. However, when T_m is lower than T_{sm}, final engagement can be realized as the system trying to archived stick slip transient response in a steady state phase.

4.1.2 Stick-slip Transient Response (Judder)

In stick-slip transient response, investigation is focused on the parameter that produces a significant effect on the stick-slip response characteristic of the 2-DOF torsional system. The phenomenon of stick-slip torsional motion is usually referred to as the clutch judder problem that typically occurs at low frequencies. The low frequencies are cause by the friction surfaces that rub concentrically. In addition to objectionable noise problem, significant stick slip torsional motion could be transferred by the differential to the in the form fore-after jerk. Sometimes this problem is called negative damping introduced by the negative slope of μ_k ($\dot{\delta}_1$). Stick-slip response occurs when $\dot{\theta}_2$ approach $\dot{\theta}_1$, subsequent to the pure slip motion as discussed above. This phenomenon is caused by the external torsional load which is higher and results in higher value of kinetic friction (μ_k). Thus it allows the slip motion in stick condition. Based on the experimental results, there are three parameters identified that pronounced the stick-slip response namely kinetic friction (μ_k), saturation friction torque (T_{sm}) and phase lag (ψ). As stick slip investigation only concern on the angular velocity 1 ($\dot{\delta}_1$) with respect to time, the stick slip influence is determine by the time required for the clutch to slip simultaneously in stick condition which represented by the width of protrusion emerge in the graph profile.

Effect of Kinetic Friction (μ_k) on the 2-DOF torsional system

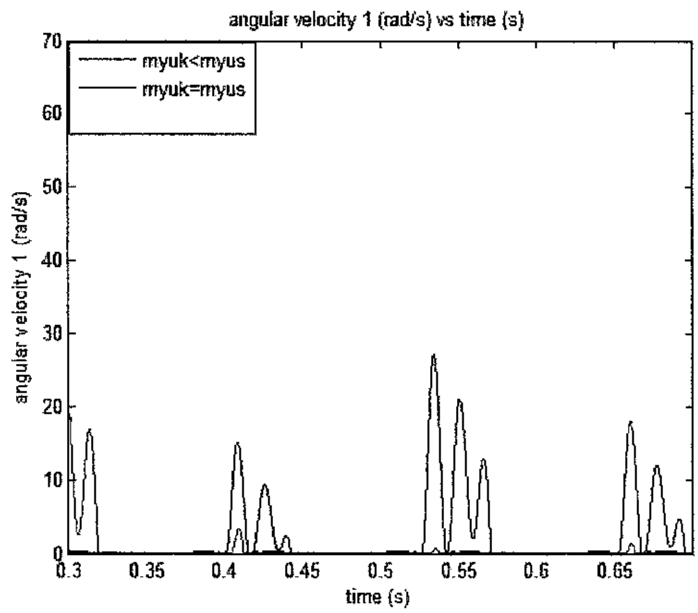


Figure 4.6: Effect of kinetic friction (μ_k) on the 2-DOF torsional system at ω = 50./s and T_{sm} = 500Nm.

First, the effect of kinetic friction on the stick slip transient response is investigated by setting constant saturation friction torque T_{sm} to the value of 500Nm with both kinetic and static friction is setup to be variable as shown in Figure 4.6. From the friction formulation of equation (2-13), a decrease in kinetic friction (μ_k) with angular velocity 1 ($\dot{\delta}_1$) affect the system in two ways. First, a negative slope regime is formed. Second, the saturation friction torque $T_{sm}= \mu_k ARP_m$ is reduced thus attenuate slip motion. Figure 4.6 shows results for two values of μ_k. As expected, the stick slip motion become more pronounced when $\mu_k=\mu_s$ because the negative damping enhances the slip motion. Conversely, when $\mu_k<\mu_s$ the stick slip motion are attenuated as a result of a positive damping as well as a lower value of T_{sm}. The summary of μ_k and T_{sm} that influence the characteristic of stick slip motion is tabulated in Table 4.2.

Table 4.2: Influence of μ_k and T_{sm} on stick slip motion.

condition	T_{sm}	slip motion	explanation
$\mu_k<\mu_s$	low	reduced	Reduced T_{sm} attenuate the slip motion.
$\mu_k=\mu_s$	high	increase	Negative slope introduced and pronounced the slip motion

Effect of Reduced T_{sm} and Negative Slope

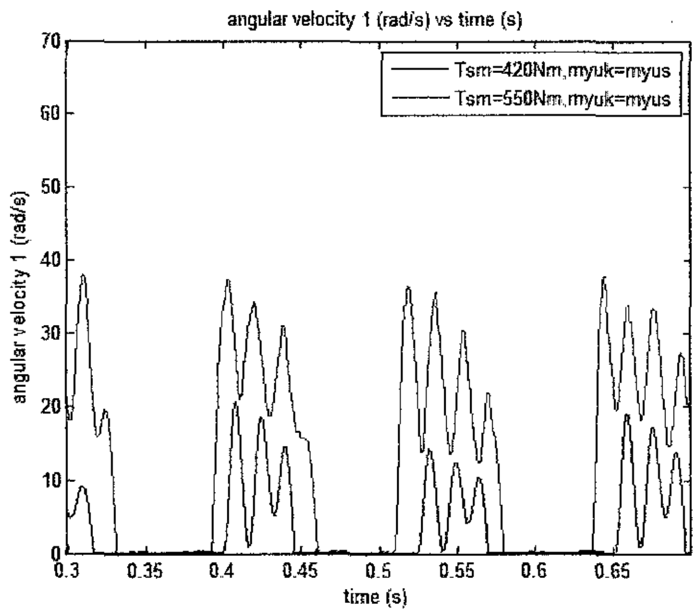
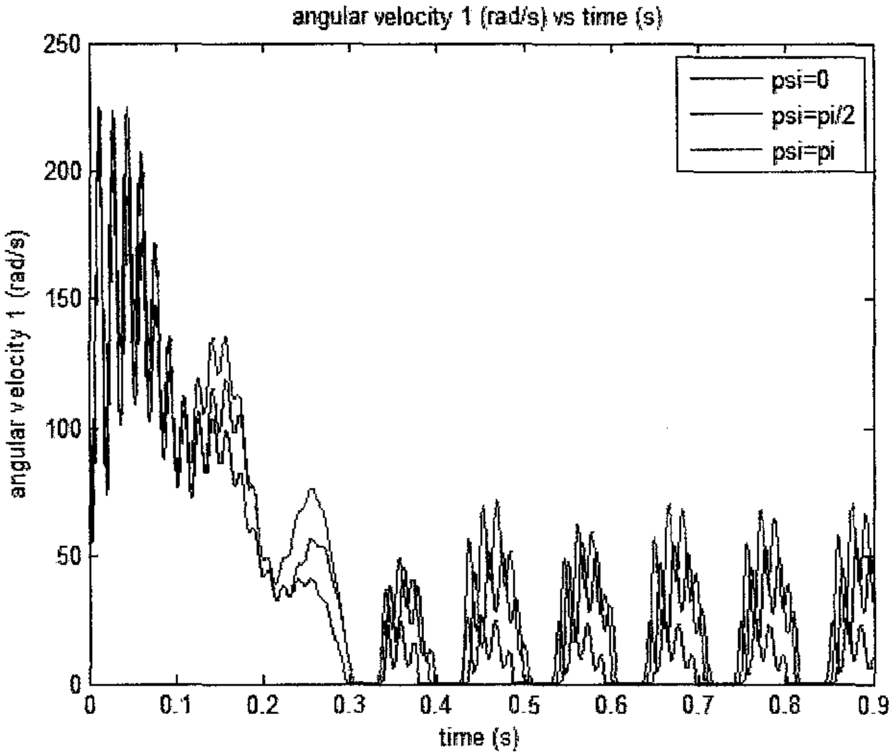


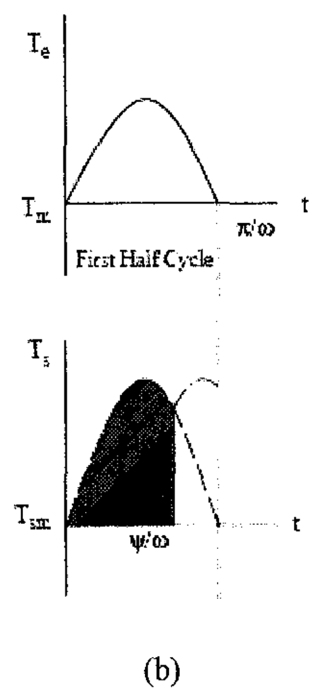
Figure 4.7: Effect of reduced T_{sm} and negative slope in friction on 2-DOF torsional system when excited at ω = 50./s

To illustrate the effect of the negative slope more clearly, Figure 4.7 compares the result for two cases of T_{sm} and μ_k. Here, the static friction and kinetic friction is held constant, $\mu_k=\mu_s$ for both cases. At the same time, the value for T_{sm} is increase from 420Nm to 550Nm to illustrate the significant effect of negative slope. Note that although reduce in T_{sm} would attenuate the slip motion as discussed before, the introduction of negative slope characteristic could further amplify the slip motion which results in stick slip motion in a condition where T_{sm} is relatively lower.

Effect of Phase Lag



(a)



(b)

Figure 4.8: (a) Effect of phase lag given ω=60. (b) Quasi-static explanation of the effect of harmonically varying friction torque. ($\omega = \omega_f$): blue solid line $T_e(t)$;black dotted line $T_s(t)$ in phase; red dotted line $T_s(t)$ not in phase.

Results from Figure 4.8(a) shows that phase lag (ψ) tends to affect the stick slip motion of the system. To examine the effect of harmonically varying friction torque on judder, first apply $T_s(t)$ at $\omega=\omega_f$, where ω is the frequency of engine torque $T_e(t)$, but with phase lag of ψ. Results from Figure 4.8(a) show that in phase $T_s(t)$ tends to attenuate the stick slip motion. According to journal of sound and vibration [3], a physical explanation can be found by analyzing the relationship between $T_e(t)$ and $T_s(t)$ in a quasi-static manner as shown in Figure 4.8(b). Shaded area represents the case of reduced slip motion. Note that only positive stick slip motion ($\dot{\delta}_1>0$) are excited under a significantly high positive mean torque T_m. When the engine torque is higher, the friction interface tends to initiate positive slipping motion. Quasi-statically, a higher value of T_s should suppress this tendency. But when there is phase lag between T_s and T_e, such suppression should be reduced. When positive slipping tends to occur in the first half engine cycle, $\psi=0$ provides the maximum decrease of the effective excitation.

CHAPTER 5
CONCLUSION AND RECOMMENDATION

5.1 Conclusion

This study on the automotive driveline system with torque converter clutch under varying load is significant in automotive research and development project. The development of the mathematical modelling of a system with governing equation lead to further study the transient response of the system with computer programming. The significant transient pure slip response and stick-slip response is identified with the introduction of different rotational speed of dry friction element. Specific effect of a significant dry friction path on the transient responses of torsional system is identified. The analysis shows that parameters identified as T_m, T_{sm} and I_1 control the clutch engagement rate within the friction interface. The friction characteristic with negative slope damping is found significantly to affect the dynamic of the system in two ways namely as negative damping effect and the reduction in the saturation friction torque. Although the reduction of saturation torque T_{sm} could further attenuate stick slip response, the negative damping effect under $\mu_k=\mu_s$ condition could further amplify the stick slip response. Furthermore, the phase lag between engine torque T_e and saturation torque T_s could amplify the stick slip motion in harmonically varying friction torque on judder. The judder problem results in unwanted noise produced by slipping clutch and wear of the clutch. Further enhancement and improvisation in this study would lead the solution to improve on clutch engagement rate and attenuate stick slip motion in torque converter clutch thus increase the engine efficiency and eliminate unwanted noise resulted from the judder motion.

5.2 Recommendation

To improve the results accuracy in this study, the comparison between numerical programming simulation and actual experimental measurement have to been performed to ensure results reliability and consistency.

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APPENDICES

APPENDIX A

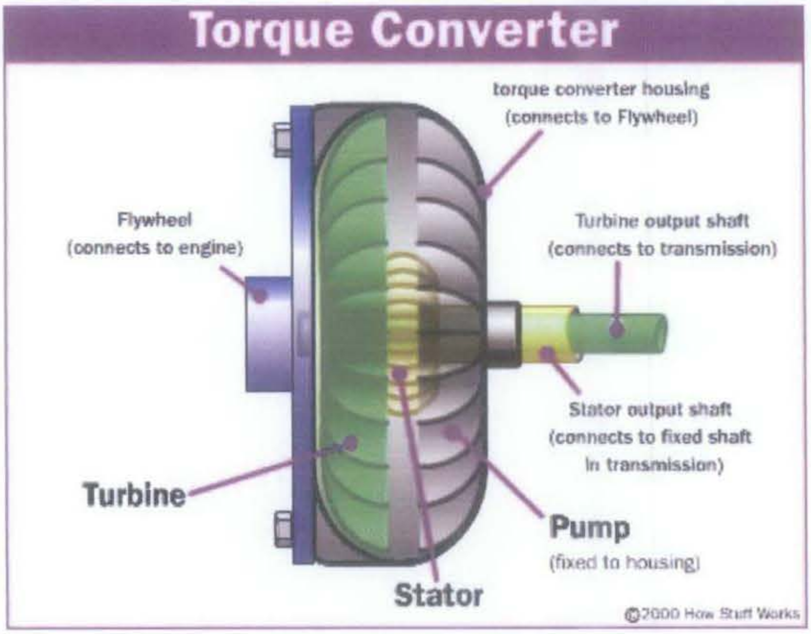

Figure A1: Torque converter component

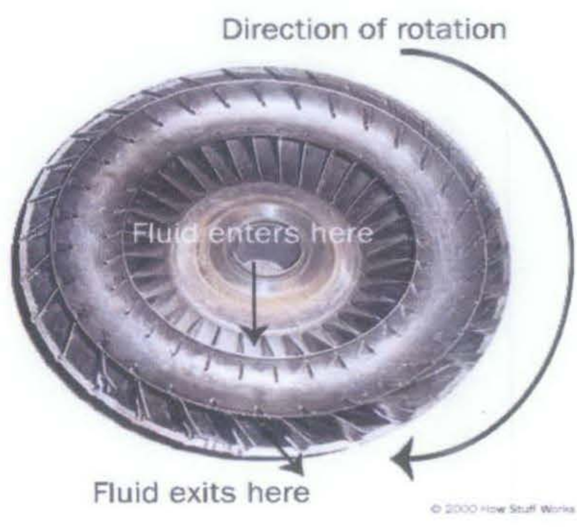

Figure A2: Fluid motion and direction in pump

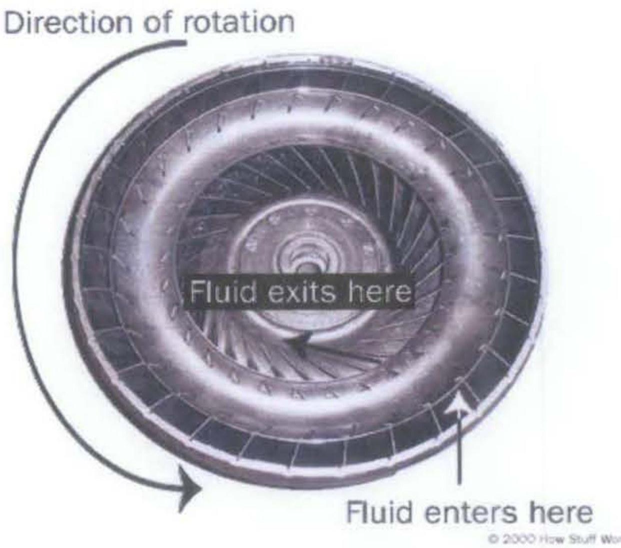

Figure A3: Fluid motion and direction in turbine

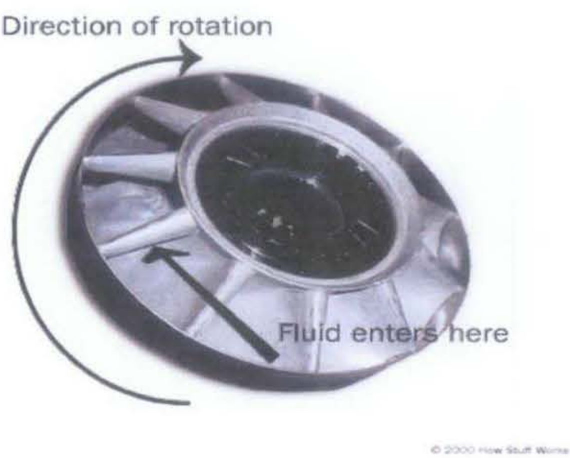

Figure A4: Fluid motion and direction in stator