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JUDUL: PERFORMANCE OF PID CONTROLLER FOR ENGINE SPEED CONTROLLER USING CVT

SESI PENGAJIAN: 2010/2011

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PERFORMANCE OF PID CONTROLLER FOR ENGINE SPEED CONTROLLER USING CVT

WAN HAZMIN BIN WAN OSMAN

BACHELOR OF ENGINEERING

UNIVERSITI MALAYSIA PAHANG

2010

PERFORMANCE OF PID CONTROLLER FOR ENGINE SPEED CONTROLLER USING CVT

WAN HAZMIN BIN WAN OSMAN

Report submitted in fulfilment of the requirements
for the award of the degree of
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DECEMBER 2010

UNIVERSITI MALAYSIA PAHANG

FACULTY OF MECHANICAL ENGINEERING

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DEDICATION

To my beloved father and mother

Wan Osman Bin Wan Ibrahim
Tengku Nariman Bt Tengku Jaafar

My Supervisor
Dr. Sugeng Ariyono

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ABSTRACT

This project is a simulation and experimental investigation into the development of PID controller using MATLAB SIMULINK software. The simulation development of the PID controller with the mathematical model of DC motor is done using Ziegler-Nichols method and trial and error method. The PID parameters are to be tested with an actual motor also with the PID controller in MATLAB/SIMULINK software. In order to implement the PID controller from the software to the actual DC motor data acquisition is used. From the simulation and the experiment, the result performance of the PID controller is compared in term of response and the assessment is presented. CVT transmission firstly designed using the Solid Work software to know the mechanical system and the dimension of this transmission. Then, the mathematical calculation is done to know the criteria DC Motor to be selected and the exact rotation of the motor as primary source to the transmission. The motor must be capable to produce 4.14N.m or more of torque in order to mechanism to function perfectly until it reach it limit.the DC motor should rotate only in 16.12rad or 2.57 rotation.Therefore PID controller are used to achieve this value in the simulation. After using the trial and error metod in order to decided the exact value for costant value of Kp=15, Ki=25 and Kd=0.15 in PID controller system to ensure the DC motor is rotated in 16.12rad.

ABSTRAK

Projek ini adalah simulasi dan eksperimen penyiasatan ke dalam pembangunan kawalan PID menggunakan perisian MATLAB SIMULINK. Pembangunan simulasi kontroler PID dengan model matematik motor DC dilakukan dengan menggunakan kaedah Ziegler-Nichols dan kaedah cuba and jaya. Parameter PID akan diuji dengan parameter motor sebenar dengan menggunakan pengawal PID pada perisisan MATLAB/SIMULINK. Bagi mengaplasikan pengawal PID dari perisian kepada motor DC sebenar, data 'acquisition card' digunakan. Dari simulasi dan eksperimen, keputusan kecekapan dari pengawal PID dibandingkan dari segi respon dan analisis dilakukan dan dibentangkan. Pada awalnya CVT direka menggunakan perisian Solid Work untuk mengetahui sistem mekanikal dan dimensi penghantaran ini. Kemudian, pengiraan matematik ini dilakukan untuk mengetahui kriteria DC Motor untuk dipilih dan putaran tepat dari motor sebagai sumber utama penghantaran. Motor harus mampu untuk menghasilkan 4.14Nm atau lebih dari torsi dalam rangka mekanisme berfungsi dengan sempurna hingga mencapai limitnya. Motor DC harus memainkan hanya pada 16.12rad atau 2.57 putaran. Oleh sebab itu kawalan PID digunakan untuk mencapai nilai ini dalam simulasi. Selepas menggunakan kaedah cuba dan jaya untuk memutuskan nilai yang tepat untuk nilai malar Kp = 15, Ki = 25 dan Kd = 0,15 dalam sistem kawalan PID untuk memastikan motor DC diputar pada 16.12rad.

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LIST OF SYMBOLS

e	The error between the reference and the output signal system
T_i	The integral time
T_d	The derivative time
K_p	Proportional gain
K_i	Integral gain
K_d	Derivative gain
α	Wrap angles
τ_{in}	Input torques
τ_l	Load torques
J	Moment of inertia of the rotor
b	Damping ratio of the mechanical system
K, K_e, K_t	Electromotive force constant
R	Electric resistance
L	Electric inductance
V	Source Voltage
θ	Position of shaft
h	Increment radius
L_{belt}	Belt length

LIST OF ABBREVIATIONS

CVT	Continuously Variable Transmission
EMDAP-CVT	Electromechanical Dual Acting Pulley Continuously Variable Transmission
PD	Proportional Derivative
PI	Proportional Integral
PID	Proportional Integral Derivative
DC	Direct current

CHAPTER 1

INTRODUCTION

1.1 INTRODUCTION

The internal combustion engine or (ICE) is an engine in which the combustion of a fuel occurs with an oxidizer using air in a combustion chamber. ICE is operates with optimum fuel consumption within a smaller range of its overall operational speed, so transmission gear ratios are designed to keep the engine operating conditions (Tawi, 1997).

Then, continuously variable transmissions (CVT) are designed to overcome this situation for controlling the transmission gear ratio such that the engine is kept operating within this optimum speed range most of the time and at the same time satisfy the driver's demand for more torque during vehicle acceleration. Lately CVT have become great deals of interest in the automotive sector due to the potential of lower emissions and better performance. A CVT is an emerging automotive transmission technology that offers a continuum of gear ratios between high and low extremes with fewer moving parts. This consequently enhances the fuel economy and acceleration performance of a vehicle by allowing better matching of the engine operating conditions to the variable driving scenarios (Sugeng, 2009) .

There are also many types of controller used in the industry, such controller is PID controller.PID controller or proportional-integral-derivative controller is a generic control loop feedback mechanism widely used in industrial control systems. A PID controller attempts to correct the error between a measured process variable and desired set point by calculating and then outputting a corrective action that can adjust the process accordingly. So by integrating the PID controller to the DC motor were able to correct made by the DC motor and control speed or the position of the motor to the desired point or speed.

1.2 PROBLEM STATEMENT

i. ICE commonly used for mobile propulsion in automobile, equipment, and other portable machineries. It is almost impossible to run an ICE in optimum control line or maximum power using conventional gearbox.
ii. Infinite transmission ratio can be control using CVT that allowing the engine to operate at optimum efficiency or fuel efficiency.
iii. Design a controller is the major challenging to all manufacturers that can match the torque capacity, efficiency, size, weight, and manufacturing cost of step-ratio transmission.

1.3 PROJECT BACKGROUND

Matlab software is used in this project. PID controller is a method of controller to controlling DC motor which is important component to determined gear ratio in this CVT. DC motor provides power to rotate the pinion of EMDAP-CVT, which in turn rotates the gear using a CAM hence creating linear movement of each pulley sheave. A DC motor model is needed to determine the amount of force to move the cam that changes the belt diameter of each pulley. As a result of this project, to design an effective PID controller, three gain parameters, namely proportional, integral and derivative gains need to be specified. The conventional approach to determine the PID parameters is to study the mathematical model of the process and try to use simple tuning parameters that provide a fixed set of gain parameters (Sugeng, 2009).

1.4 PROJECT OBJECTIVE

i. To design a CVT transmission model using a solid work software.
ii. To control the position of DC motor with PID controller using MATLAB/SIMULINK application.
iii. To design the PID controller and tune it using MATLAB/SIMULINK.

1.5 PROJECT SCOPES

The following are the scopes of the study:

i. Determine the effective rotation (position) of the DC motor that can move the pulley to desired gear ratio.
ii. Calculate the gear ratio that produced from the CVT design.
iii. The CVT design or type based on rubber belts.
iv. Design and produce the simulation of the PID controller.
v. Simulate the PID simulation with an actual DC motor.

CHAPTER 2

LITERATURE REVIEW

2.1 INTRODUCTION

CVT is a transmission device that is used to provide a set of discrete angular velocity outputs from a constant velocity source. A continuously variable transmission (CVT) serves the same function as a conventional power transmission device. The difference is that a non-discrete range of outputs is produced, and its speed ratio can be varied continuously. The CVT improves vehicle drive ability and passenger comfort as it removes the process of shifting gears in conventional transmission thus avoiding uneven vehicle acceleration.

2.2 BACKGROUND AND BRIEF HISTORY

In the year 1490, Leonardo de Vinci sketched his idea for CVT. CVT already begun in the early era of car development in the same period of conventional automatics. Due to cost concern, General Motors had developed a fully toroidal CVT and conducted extensive testing before eventually deciding to implement a conventional stepped-gear automatic. General Motor Research reworked on CVTs in the 1960s, but none ever saw their production. British manufacturer Austin used a CVT for several years in one of its smaller cars, but it was dropped due to its high cost, poor reliability, and inadequate torque transmission (Yamaguchi.J, 2000).Simple rubber band and cone system is the most material using in the early stage of CVT. It’s simply likes the one developed by a Dutch firm, DAF, in 1958. The problem is it could only handle 0.6 l engine, and severe problem with noise and rough starts eventually to hurt its reputation (Birch, 2000).

2.3 ADVANTAGE OF CONTINUOUS VARIABLE TRANSMISSION (CVT)

All drivers are familiar with the clunking sound of shifting transmission. Therefore, CVT is designed to overcome this problem with perfectly smooth and naturally changes its ratio discretely such that the driver or passenger feels only steady acceleration. In theory, as the harshness of shifts and discrete gears force the engine to run at a less than optimal speed, a CVT would cause less engine fatigue and would produce a more reliable transmission (Mori.H and Yamazki, 2001).

CVTs offer improved efficiency and performance. Table 2.1 shows the power efficiency of a typical five speeds automatic, which is the percentage of engine power transmitted through the transmission. This yields an average efficiency of 86%, compared with a typical manual transmission with 97% efficiency (Kluger and Fussner, 1997a).

Table 2.1: Efficiency versus gear ratio for automatic transmission.

Gear	Efficiency Range
1	60-85%
2	60-90%
3	85-95%
4	90-95%
5	85-94%

Source: (Kluger and Fussner, 1997b)

Table 2.2: Efficiency of various CVT design.

CVT Mechanism	Efficiency Range
Rubber belts	90-95%
Steel belts	90-97%
Toroidal traction	70-94%
Nutating traction	75-96%
Variable geometry	85-93%

Source: (Sugeng, 2009)

By comparison, Table 2.2 shows the efficiency range for several CVT designs. Their efficiency depends less on driving habit than manual transmission. Since CVT allows an engine to run at its most efficient point virtually independent of the vehicle speed, a CVT equipped vehicle yields fuel economy benefits when compared with a conventional transmission (Kluger and Fussner, 1997a) .

2.4 CHALLENGES AND LIMITATIONS

The progress of CVT development has been slow due to unsuccessful efforts to develop a CVT that can match the torque capacity, efficiency, size, weight, and manufacturing cost of step-ratio transmission. In addition, the delay in CVT development can be attributed to the lack of demand as the conventional manual and automatic transmission have long offered sufficient performance and fuel economy (Broge, 1999).

One of the major complaints that related to previous CVTs is the slippage in drive belt or roller has been with. The complaints triggered due to the lack of discrete gear teeth, which form a rigid mechanical connection between two gears which friction drives are inherently prone to slip, especially at high torque. A simple solution to this problem which has been used for many years is by limiting the usage of CVTs only in cars with relatively low torque engine. Other than that, another solution for the problem is by employing a torque converter. However, it will eventually reduce the CVT's efficiency (Yamaguchi.J, 2000).

CVTs can be applied in cars with high torque engine with the improvements in manufacturing technique, technology material processing, metallurgy, advance electronic control and advance engineering. The selection of the ratio is essential as to operate CVT at the optimal transmission ratio at any speed. Manual transmissions have manual controls, where the desired gear ratio totally depends on the driver to shift it while automatic transmissions have relatively simple shifting algorithms. However, more complex algorithm is required for CVT to accommodate an infinite division of speed and transmission ratios.

2.5 VEHICLE MODEL

Transmission of gearbox and the final drive shaft are important component to transmit the engine torque produced by the engine to the wheel. The whole vehicle model including the engine, clutch, CVT and load, and dynamics model of CVT system was developed based on different stages of engaging clutch and studied through simulation, similar study has been carried out by other researchers. They found that a conventional proportional control strategy could not satisfy the control demand for engaging clutch; hence they designed a fuzzy controller for the clutch control and applied self-adjusting PD for the ratio control. The simulation results indicated that the speed ratio controller has good control effect and implements reasonable match between engine and CVT. It demonstrates that the simulation model established is acceptable and reasonable, which can offer theoretical help to devise and develop CVT system (Jun and Long, 2001).

2.6 TRANSMISSION MODEL

A power transmission device whose speed ratio can be varied in a continuous manner is known as CVT. Meanwhile, traditional fixed ratio transmission (FRT) can only vary speed ratio in certain discrete steps.

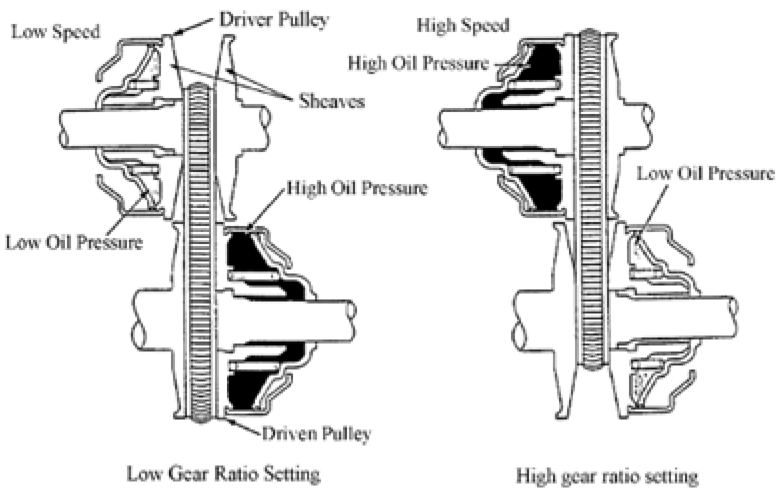


Figure 2.1: Overview of a Van Doome's Belt Driven CVT.

Source: (K.K.Ang et al., 2001)

Figure 2.1 illustrates two types model of transmission system of Van Doome's belt driven CVT.

2.7 CONTROLLER DESIGN

For both linear and non-linear systems, there are using PID controllers because of their simplicity. Adjusting the parameters is needed to controls satisfactory control performance. But the selection parameters for nonlinear systems are always a challenge for the control engineers involved PID. Therefore PID are widely use in simple linear control systems just show in figure 2.2.

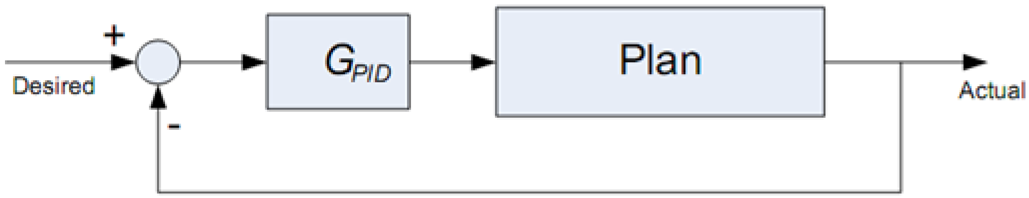


Figure 2.2: A general PID control scheme

The ideal continuous transfer function of a PID (G*PID*) controller is given by

$$Gpid = Kp(e + (\frac{1}{t}) \int_0^t e\,dt + Td\left(\frac{de}{dt}\right) \qquad (1)$$

Where,

Td - Kd/Kp

e - The error between the reference and the output signal system

Ti - The integral time

Td - The derivative time

Kp - Proportional gain

Ki - Integral gain

Kd - Derivative gain

In digital control and for small time sampling (Ts), the equation can be approximated by

$$Gpid = Kp(en) + \frac{Ts}{Ti}\sum_{j=1}^{n} ej + \frac{Td(en - en - 1)}{ts} \qquad (2)$$

2.8 DYNAMIC MODELLING OF BELT CVT

Steel V-belt or a rubber V-belt is commonly used as power-transmitting device in a belt-type CVT. Most of the existing models CVTs are based on the principles of quasi-static equilibrium, which are steady-state model with a few exceptions. In order to achieve the quasi-static equilibrium the analysis is used to develop a set of equations that capture the dynamic interactions between the belt and the pulley. Variable sliding angle approach was implemented to describe friction between the belt and the pulley as the belt is capable of moving both radials and tangentially.

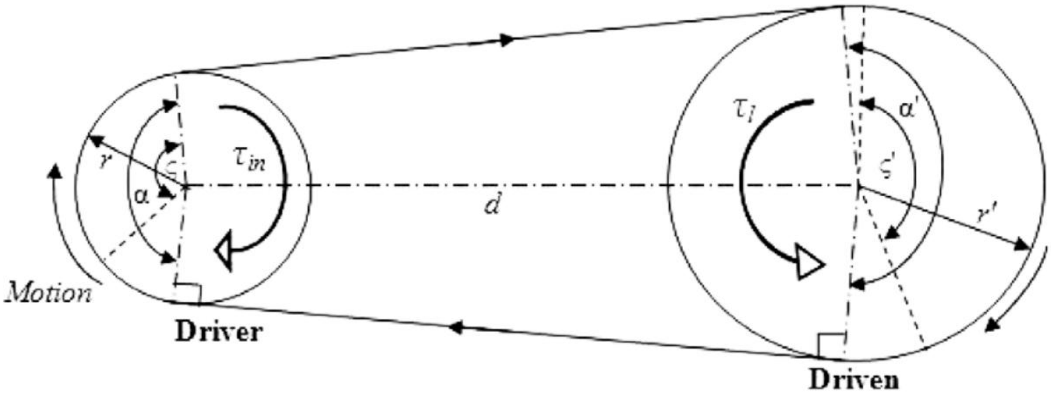


Figure 2.3: Geometry of the Belt-Driven CVT

Source: (K.K.Ang et al., 2001)

Many researchers assumed constant sliding angle over the pulley wrap to derive the equations of motion of belt. Prediction of dynamics of a belt-pulley system can be made from the variable sliding angle which required are the equilibrium, compatibility, and constitutive equations be solved simultaneously. Only centripetal effects were modelled to account for the influence of belt inertia on system dynamics.

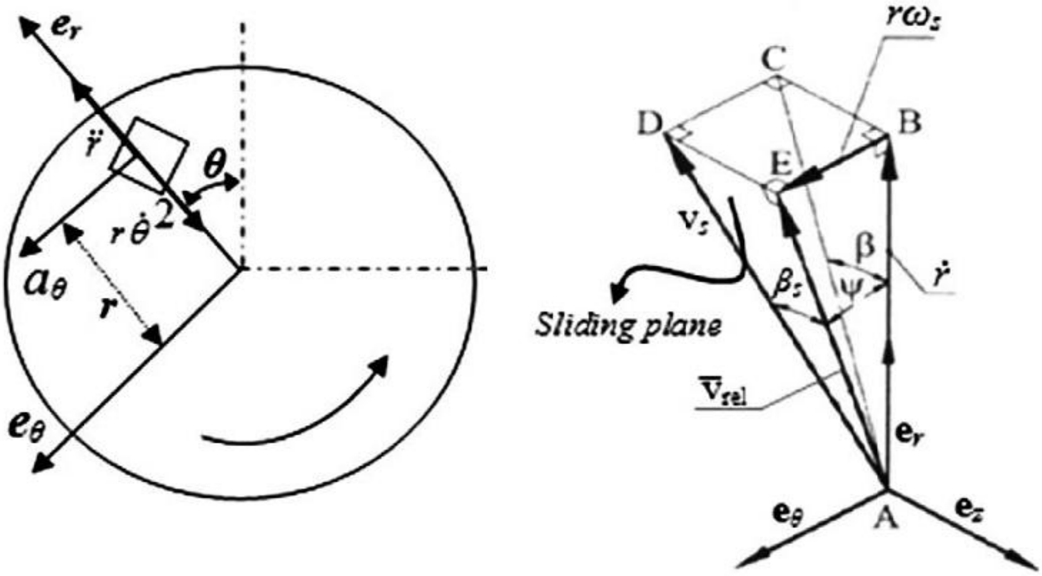


Figure 2.4: Kinematic Description of a Belt Element on the Pulley Wrap.

Source: (N.Srivastava and Imtiaz.Haque, 2009)

The geometric configuration, sliding plane, and simplified kinematics of a V-belt CVT drive with negligible belt flexural effects are illustrated in Figure 2.4. Only the sliding or active arcs, represented in Figure 2.3 contribute actively to torque transmission while (α, α') represent the wrap angles of the belt–pulley contacting arcs. In addition to that, (τ_{in} ,τ_l) represent the input and load torques on the driver and driven pulleys.

Deflection of pulley sheaves in the axial direction widens or shortens the groove width, thereby influencing the motion of belt in the pulley groove. The phenomena of local deflection, plate deflection, and pulley skewness are caused by the displacement of pulley sheaves. Figure 2.5 shows the variation in pulley groove width due to elastic deformation in the axial direction and due to skewness of the pulley halves. The local axial pressure is depending on the local deflection. Plate theory was used in order to obtain the global deflection of pulley halves. After that, as to obtain the dynamic performance indicators of the belt–pulley system plate equations and the belt equations were solved simultaneously.

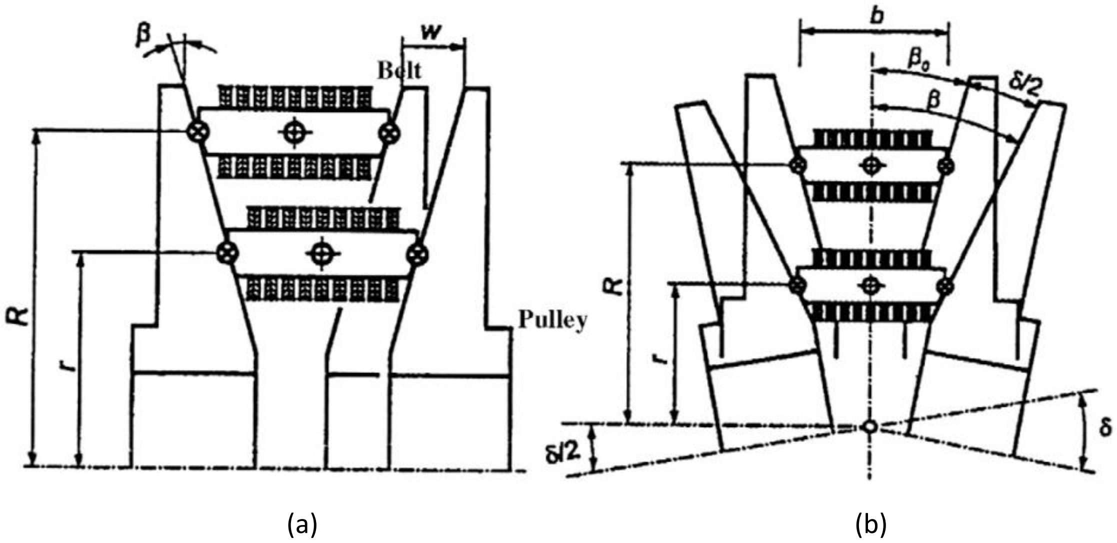


Figure 2.5: Pulley deformation model :(a) axial deformation; (b) pulley skewness.

Source: (N.Srivastava and Imtiaz.Haque, 2009)

Frictional forces are radially directed and all the derivatives in the differential equations vanish with the existence of singular solutions also known as orthogonal points. The character of a solution at an orthogonal point to a great extent determines the character of the solution at other locations and conditions. The axial forces for pulleys with small to medium skewness did not differ much from the ones in ideal case. On the other hand, the pulley axial forces from low to medium torque applications have increased by large skewness.

2.9 ELECTROMECHANICAL DUAL ACTING PULLEY CONTINUOUSLY VARIABLE TRANSMISSION (EMDAP-CVT)

EMDAP-CVT offers many potential advantages for good drivability. The power flows through the power train is not interrupted during acceleration as in conventional transmissions and this makes it possible to gain a smooth, rapid and step less response to drivers' demand.

This EMDAP-CVT uses electro-mechanical element to control the axial movement of the pulley sheaves and utilises two DC motors which are coupled with a uniquely designed power screw mechanisms. These are needed to provide high axial or clamping force to compress metal pushing V-belt (MPVB) via the CVT V-pulley sheaves. EMDAP-CVT is designed in such a way that its main components are symmetrical and dual acting, with plane of symmetry parallel to the transverse cross-section of both the input and output shafts.

The main sub-components of EMDAP-CVT are:

i. DC electric motor and its gear reducer mechanism
ii. Power screw mechanism
iii. V-pulley mechanism and
iv. Reverse forward mechanism.

The DC motor and its gear reducer mechanism are responsible to provide the required high clamping force to maintain belt tension and the appropriate ratio change during operation. The power screw mechanism is the media that transform and transmit the rotational motion of the DC motor and its gear reducer mechanism, into linear axial clamping force and provide the self locking mechanism to maintain the required constant ratio during operation. The V pulley mechanism will then transforms and transmits the linear axial clamping forces into circumferential rotational forces provided by the engine via the MPVB, which then, transmit it to the output shaft and finally, the reverse forward mechanism to enable the EMDAP-CVT to reverse its direction of rotation.

The general block diagram of EMDAP-CVT model is shown in Figure 2.9.1. The input voltage of each DC motor will rotate the power screw mechanism and create movement of each pulley sheave, hence creating different belt radii, Rp and Rs. The working principle of the block diagram will be detailed in the next section.

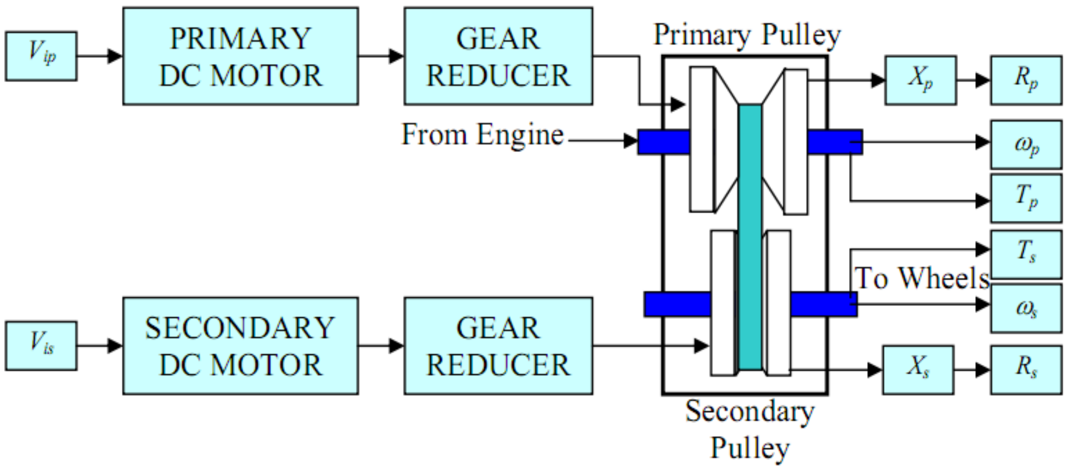


Figure 2.6: Block diagram of EMDAP-CVT

Source: (Sugeng, 2009)

2.10 POWER SCREW SYSTEM

The MPVB CVT force is quite large approximately about 15 to 20 kN for the compression force required to change the pulley ratios (Micklem.J.D, 1990, Tawi, 1997). Design of the CVT pulleys mechanisms in which the forces cancel out each other and are fully sustained by the pulleys' shaft is the main challenge for researchers. This probably explains why most of the present CVT pulley systems opt to hydraulic mechanism.

Electromechanical CVT with power screw mechanism requirements lead to DGR, UTM to propose the power screw design. The axial movements of both the primary and secondary pulley sheaves will determine the CVT ratio. During the CVT ratio change, the axial movement of the driving and the driven pulleys are in different directions; inward to the MPVB and outward from the MPVB. Figure 2.7 shows the proposed design of power screw to actuate the pulley axial movement.

In order to produce an opposite movement between the two sheaves pulley moving axially simultaneously, there are two pairs of power screws, which are fabricated with different thread turn direction (right/left hand turn). By utilizing two DC motors as actuators to turn the power screw, the ratio of the CVT can be obtained. The dual acting pulley sheave movements will enable the metal belt to remain at its centre line, hence avoiding belt misalignment. This CVT system is named as Electro-Mechanical Dual Acting Pulley Continuously Variable Transmission (EMDAP-CVT).

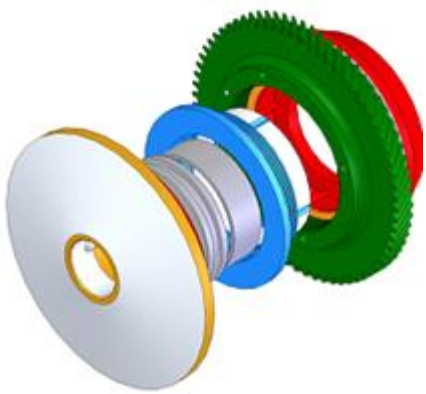


Figure 2.7: Power Screw Design

Source: (Sugeng, 2009)

2.11 PID CONTROLLER OF DC MOTOR

A proportional–integral–derivative controller (PID controller) is a generic control loop feedback mechanism (controller) widely used in industrial control systems a PID is the most commonly used feedback controller. A PID controller calculates an "error" value as the difference between a measured process variable and a desired set point. The controller attempts to minimize the error by adjusting the process control inputs. In the absence of knowledge of the underlying process, a PID controller is the best controller. However, for best performance, the PID parameters used in the calculation must be tuned according to the nature of the system – while the design is generic, the parameters depend on the specific system.

The PID controller calculation (algorithm) involves three separate parameters, and is accordingly sometimes called three-term control: the proportional, the integral and derivative values, denoted P, I, and D. The proportional value determines the reaction to the current error, the integral value determines the reaction based on the sum of recent errors, and the derivative value determines the reaction based on the rate at which the error has been changing. The weighted sum of these three actions is used to adjust the process via a control element such as the position of a control valve or the power supply of a heating element. Heuristically, these values can be interpreted in terms of time: P depends on the present error, I on the accumulation of past errors, and D is a prediction of future errors, based on current rate of change.

By tuning the three constants in the PID controller algorithm, the controller can provide control action designed for specific process requirements. The response of the controller can be described in terms of the responsiveness of the controller to an error, the degree to which the controller overshoots the set point and the degree of system oscillation. Note that the use of the PID algorithm for control does not guarantee optimal control of the system or system stability.

Some applications may require using only one or two modes to provide the appropriate system control. This is achieved by setting the gain of undesired control outputs to zero. A PID controller will be called a PI, PD, P or I controller in the absence of the respective control actions. PI controllers are fairly common, since derivative action is sensitive to measurement noise, whereas the absence of an integral value may prevent the system from reaching its target value due to the control action.

CHAPTER 3

METHODOLOGY

3.1 INTRODUCTION

This chapter explains project methodology where it specifically describes the method that will use in this project. It also as guideline to ensure the project flows that have planned at the beginning will follow accordingly. A CVT can be controlled by a mechanical or an electronic system, or it can be automatic by using a centrifugal CVT force. The most common and used of CVT in industrial nowadays consist two pulleys that can change diameter which is connected by a belt.

3.2 PID THEORY IN THIS DESIGN

The characteristics of the each proportional (P), the integral (I), and the derivative (D) controls, and how to use them to obtain a desired response. In this theory, we will consider the following unity feedback system:

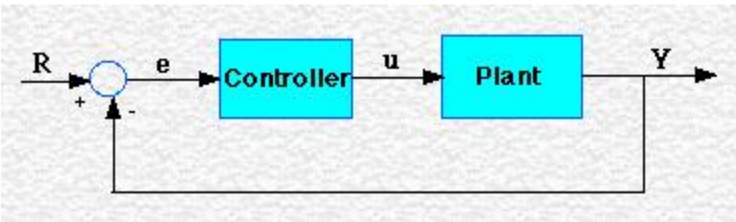


Figure 3.1: Concept of PID Controller

Plant : A system to be controlled

Controller : Provides the excitation for the plant or designed to control the overall system behaviour

3.3 THE THREE-TERM CONTROLLER

The transfer function of the PID controller is as followed:

$$Kp + \frac{Ki}{s} + Kds = \frac{Kds^2 + Kps + Ki}{s} \tag{3}$$

Kp = Proportional gain

KI = Integral gain

Kd = Derivative gain

First, let's take a look at how the PID controller works in a closed-loop system using the schematic shown above. The variable (e) represents the tracking error, the difference between the desired input value (R) and the actual output (Y). This error signal (e) will be sent to the PID controller, and the controller computes both the derivative and the integral of this error signal. The signal (u) just past the controller is now equal to the proportional gain (Kp) times the magnitude of the error plus the integral gain (Ki) times the integral of the error plus the derivative gain (Kd) times the derivative of the error.

$$u = Kpe + ki \int edt + Kd\frac{de}{dt} \tag{4}$$

This signal (u) will be sent to the plant, and the new output (Y) will be obtained. This new output (Y) will be sent back to the sensor again to find the new error signal (e). The controller takes this new error signal and computes its derivative and it’s integral again. This process goes on and on.

3.3.1 The Characteristics of P, I, and D Controllers

A proportional controller (Kp) will have the effect of reducing the rise time, but never eliminate, the steady-state error. An integral control (Ki) will have the effect of eliminating the steady-state error, but it may make the transient response worse. A derivative control (Kd) will have the effect of increasing the stability of the system, reducing the overshoot, and improving the transient response. Effects of each of controllers Kp, Kd, and Ki on a closed-loop system are summarized in the table shown below.

Table 3.1: Effect of Parameter Controller

Controller	Rise Time	Overshoot	Settling Time	S-S Error
Kp	Decrease	Increase	Small Change	Decrease
Ki	Decrease	Increase	Increase	Eliminate
Kd	Small Change	Decrease	Decrease	Small Change

Note that these correlations may not be exactly accurate, because Kp, Ki, and Kd are dependent of each other. In fact, changing one of these variables can change the effect of the other two. For this reason, the table should only be used as a reference when only determining the values for Ki, Kp and Kd.

3.4 DC MOTOR

Industrial applications use dc motors because the speed-torque relationship can be varied to almost any useful form for both dc motor and regeneration applications in either direction of rotation. Continuous operation is commonly available over a speed range of 8:1. Infinite range (smooth control down to zero speed) for short durations or reduced load is also common.

DC motors are often applied where they momentarily deliver three or more times their rated torque. In emergency situations they can supply over five times rated torque without stalling (power supply permitting).

Dynamic braking (dc motor-generated energy is fed to a resistor grid) or regenerative braking (dc motor-generated energy is fed back into the dc motor supply) can be obtained with dc motors on applications requiring quick stops, thus eliminating the need for, or reducing the size of, a mechanical brake.

DC motors feature a speed which can be controlled smoothly down to zero, immediately followed by acceleration in the opposite direction without power circuit switching. Then DC motors respond quickly to changes in control signals due to the high ratio of torque to inertia.

3.5 DC MOTOR MODELING

A common actuator in control systems is the DC motor. It directly provides rotary motion and, coupled with wheels or drums and cables, provide transitional motion. The electric circuit of the armature and the free body diagram of the rotor are shown in the following figure:

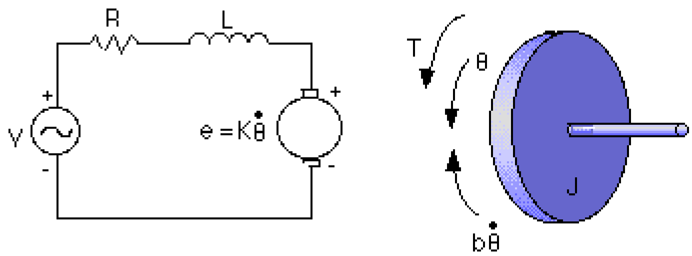


Figure 3.2: DC Motor Free Body Diagram

For the DC motor model, its need some values for the physical parameters. These values were derived by experiment from an actual motor.

Moment of inertia of the rotor = (J)
Damping ratio of the mechanical system = (b)
Electromotive force constant = (K=Ke=Kt)
Electric resistance = (R)
Electric inductance = (L)
Input (V): Source Voltage
Output (θ): position of shaft
The rotor and shaft are assumed to be rigid

The motor torque, T, is related to the armature current, i, by a constant factor Kt. The back emf, e, is related to the rotational velocity by the following equations:

$$T = K_t i \tag{5}$$

$$e = Ke\frac{d\theta}{dt} \tag{6}$$

In SI units (which we will use), Kt (armature constant) is equal to Ke (motor constant).

This system will be modeled by summing the torques acting on the rotor inertia and integrating the acceleration to give the velocity, and integrating velocity to get position. Also, Kirchoff's laws will be applied to the armature circuit. Open Simulink and open a new model window. First, we will model the integrals of the rotational acceleration and of the rate of change of armature current.

$$\iint \frac{d^2\theta}{dt^2} = \int \frac{d\theta}{dt} = \theta \tag{7}$$

$$\int \frac{di}{dt} = i \tag{8}$$

$$J\frac{d^2\theta}{dt^2} = T - b\frac{d\theta}{dt} \rightarrow \frac{d^2\theta}{dt^2} = \frac{1}{J}(K_t i - b\frac{d\theta}{dt}) \tag{9}$$

$$L\frac{di}{dt} = -Ri + V - e \rightarrow \frac{di}{dt} = \frac{1}{L}(-Ri + V - K_e\frac{d\theta}{dt}) \tag{10}$$

Using the mathematical equation as a reference to build the simulink block, the result of Dc motor position (theta) as a output from this simulink diagram showing in figure 3.3 based on the step input.

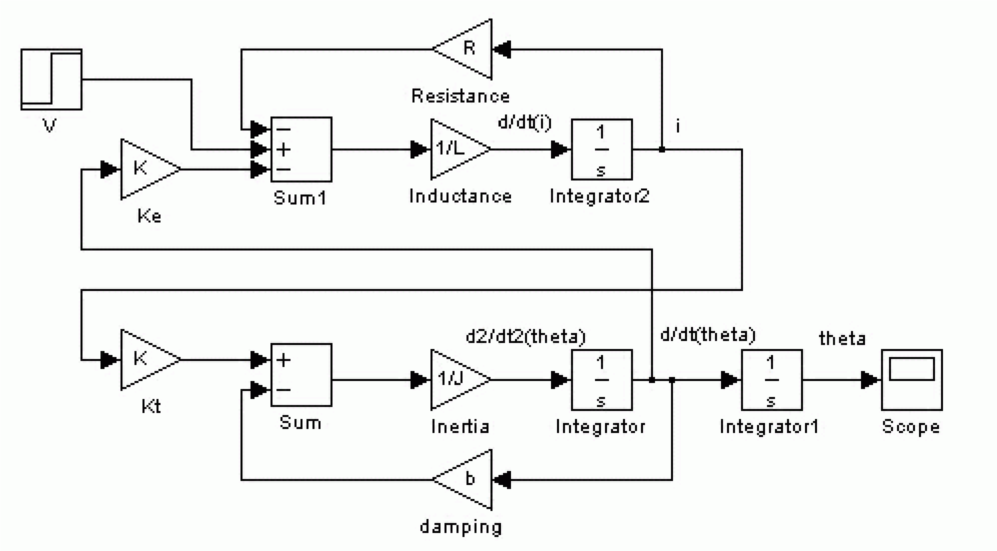


Figure 3.3: Simple Simulink Diagram of DC Motor

3.6 PID MODELING FOR THE DC MOTOR POSITION

DC motor mathematical model is completed and convert all the equation from time domain to frequency domain using Laplace Transform, then proceed with the DC motor modeling using the Matlab software.

The system schematic looks like:

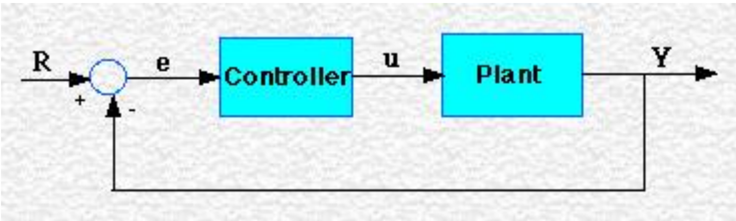


Figure 3.4: DC Motor with PID Controller

After modeling in Matlab :

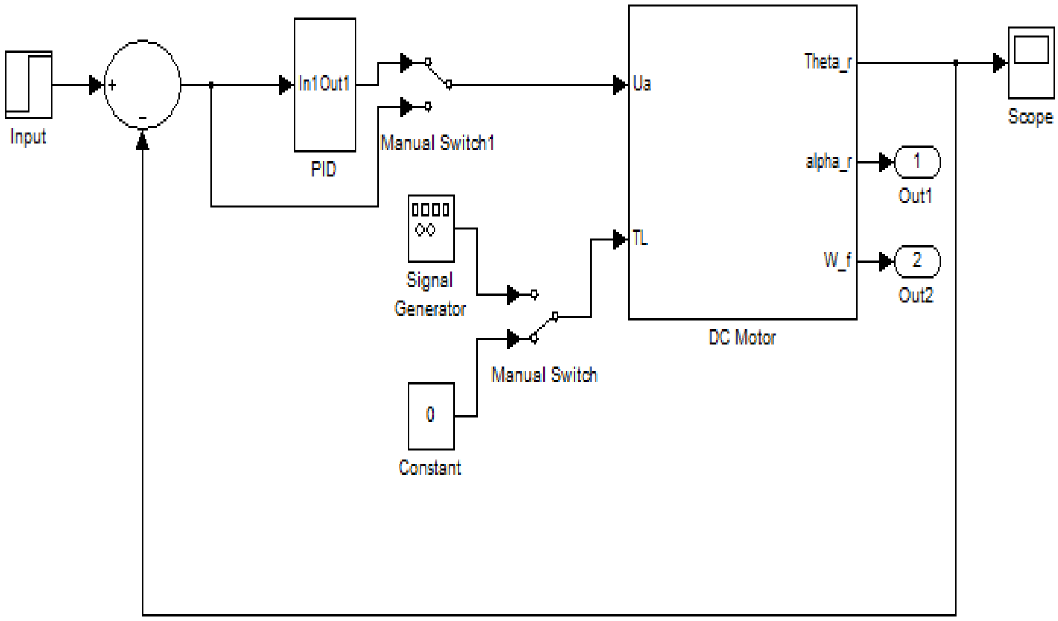


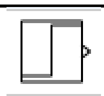
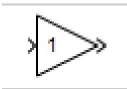
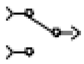
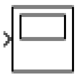
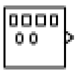
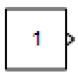
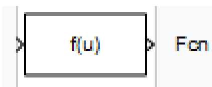
Figure 3.5: Complete Simulink Block Diagram for DC Motor with PID Controller

3.7 DESIGN OF SIMULINK BLOCK

3.7.1 Open SIMULINK Library

To get started in producing a model from DC motor design project, the properties or function of the block used must be list down first. To ensure the blocks are chose according to DC motor design criteria.

Table 3.2: List Block Used and Function

NAME	BLOCK	DESCRIPTION
Step		Output a step.
Sum		Add or subtract inputs. Must be specify the inputs signal(e.g+,-)
Gain		This block creates a bus signal from its input.
Manual switch		Output toggles between two inputs by double clicking on the block.
scope		To produce the output graph.
Signal generator		Output various wave forms.
Out1		Provide an output port for a subsystem or model.
Constant		Output the constant specified by the ‘constant value’ parameter.
Fcn		General expression block. Use “u” as the input variable name.

Source: Matlab

3.8 DC MOTOR MATHEMATICAL EQUATION

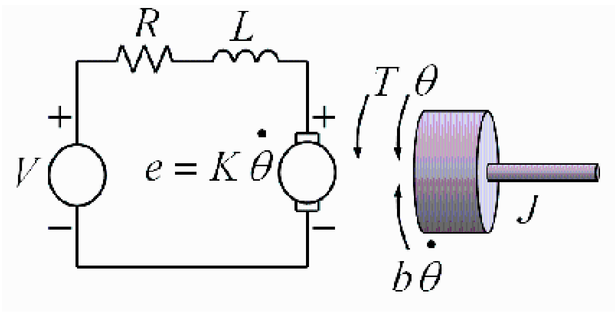


Figure 3.6: DC Motor System

Figure 3.6 show the dc motor system in terms of parameters such as current, inductance, rotor inertia, load, back emf, armature resistance, torque constant and moment of inertia. All of the parameter will be solved using the mathematical equation and then convert the time domain to frequency domain using Laplace Transform to be set in DC motor subsystem equation.

$$\frac{di}{dt} = -\frac{ra}{La}ia - \frac{K\omega}{La}ia + \frac{1}{La}ua \tag{11}$$

$$\frac{d\omega r}{dt} = \frac{Kt}{J}ia - \frac{Bm}{J}\omega r - \frac{1}{J}TL \tag{12}$$

$$\frac{d\theta r}{dt} = \omega r \tag{13}$$

$$\frac{d\omega r}{dt} = \alpha r \tag{14}$$

$$\theta = kgear\theta r \tag{15}$$

From (11),

$$ia(s) = \frac{Ua(s) - K\omega r(s)}{Las + ra} \tag{16}$$

From (12),

$$\omega r(s) = \frac{Ia(s)Kt - TL(s)}{Js + Bm} \tag{17}$$

Substitute (16) in (17),

$$\omega r(s) = \left(\frac{Ua(s)}{Las + ra} - \frac{K_{\omega}\,\omega r(s)}{Las + ra}\right)\frac{Kt}{Js + Bm} - \frac{TL(s)}{Js + Bm}$$

$$\omega r(s) = \frac{Ua(s)Kt - TL(s)(Las + ra)}{[JLa\,s^2 + (Jra + BmLa)s + Bmra + KtK\omega]} \tag{18}$$

$$\theta r(s) = \frac{\omega r(s)}{s} \tag{9}$$

$$\alpha r(s) = s\omega r(s) \tag{20}$$

$$\frac{kT}{J*La.s^2 + (J*ra + Bm*La)s + Bm*ra - kT*kw} \tag{21}$$

$$\frac{La.s + ra}{J*La.s^2 + (J*ra + Bm*ra - kT*kw} \tag{22}$$

Used above mathematical as a reference to build the dc motor model in matlab Open Simulink library browser-user defined function-Fcn. Use this Fcn block to defined our mathematical equation. In this model its need 2 input data to be verified in this Fcn block.

From (21) and (22),

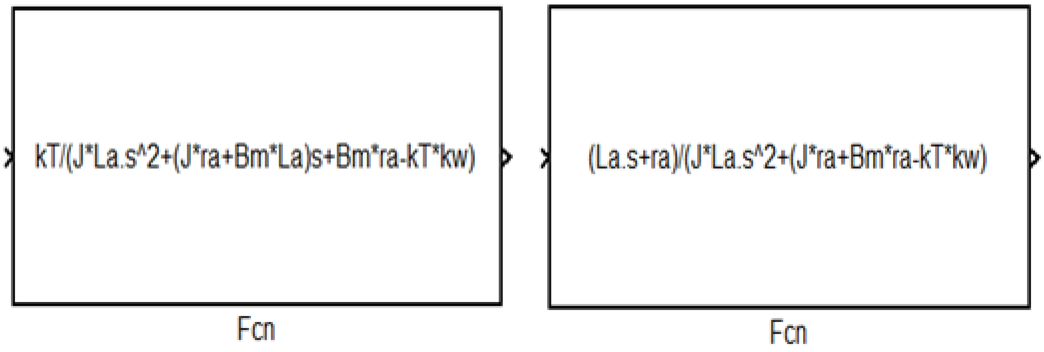


Figure 3.7: Simulink block of Equation 21 and 22.

Build the DC motor model using equation (21) and (22):

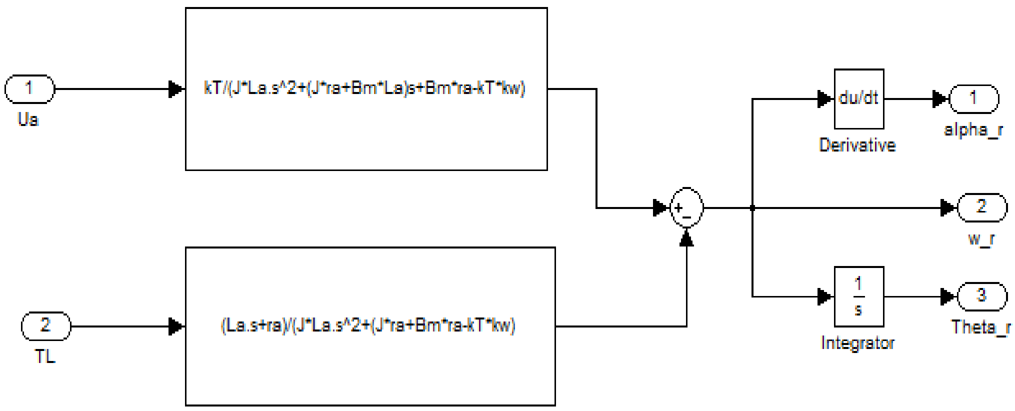


Figure 3.8: Simulink Block using fcn-block by User Defined Function

From the Figure 3.8 it's similarly when only using the block to build the model. So, the subsystem of DC motor block will be look like Figure 3.9:

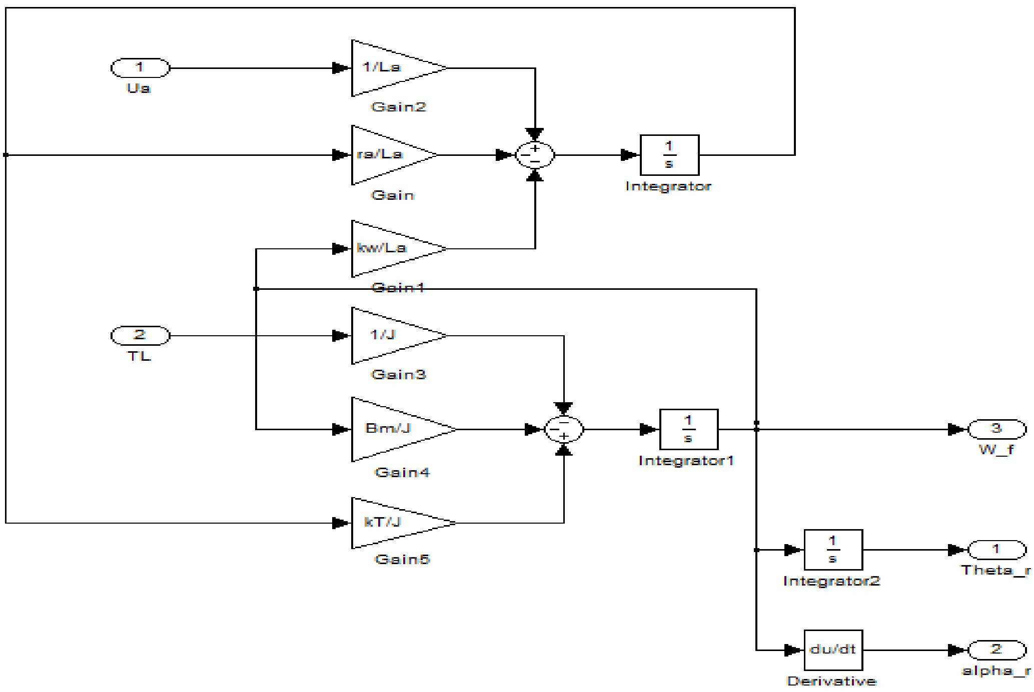


Figure 3.9: Structure of DC Motor Simulink Block

Then build the model of PID controller based on PID controller transfer function equation:

$$Kp + \frac{Ki}{s} Kds = \frac{Kds^2 + Kps + Ki}{s} \tag{23}$$

The model of PID controller its look likes:

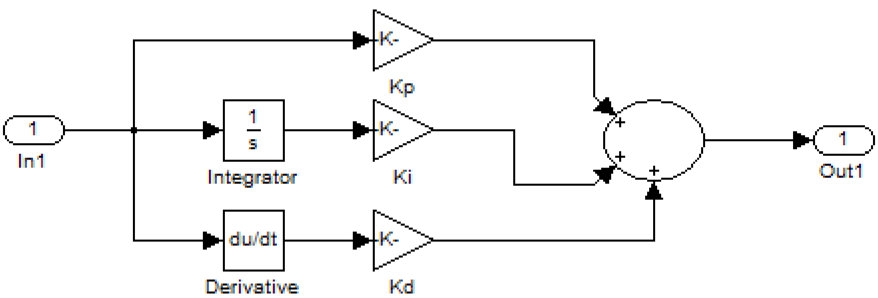


Figure 3.10: PID Controller Block Diagram

3.9 TRIAL AND ERROR METHOD

Heuristic or trial by error is a conventional method of problem solving for obtaining knowledge, both proportional knowledge and know-how. In the field of computer science, the method is called generate and test. In elementary algebra, when solving equations, it is "guess and check”. This approach can be seen as one of the two basic approaches to problem solving and is contrasted with an approach using insight and theory.

Due to the unsuccessful use of Ziegler Nichols method where when Kp increased the step pulse did not oscillate. Therefore to find the value of Kp, Ki and Kd, trial and error method was the resort. Through this method the value of Kp,Ki and Kd was obtain by increasing their value until the best result are obtained. The value of Kp,Ki and Kd are presented in table 3.3.

Table 3.3: Typical Values of Proportional, Integral, and Derivative Feedback Coefficient for PID-type Controller

Controller	Kp	Ki	Kd
P	0.5Kp max	-	-
PI	0.45Kp max	1.2 $Tosc$	-
PID	0.6Kp max	2 $Tosc$	$Tosc/8$

Kp= proportional gain

Ki=integral gain

Kd = derivative gain

CHAPTER 4

RESULT AND DISCUSSION

4.1 INTRODUCTION

In this experiment, PID controller was proposed to control the DC motor position (θ,rad). The purpose of this part is to show how result from controllers can be obtained from design selected and knowledge of the process CVT transmission to be controlled. The process with the analysis of the design to knows the requirement position of the CAM to move the pulley in linear motion in order to control the pulley radius using the mathematical equation. The choice of control loops and the controller setting may then be changed if it performance is not satisfactory. The detail drawing and design of the CVT using SolidWork software and the PID controller using MATLAB is shown in this chapter.

4.2 SELECTED DESIGN

The complete system of one rubber V belt where is applied to CVT transmission is shown in Figure 4.1 below. The design dimension is made to overcome the torque produce from Daihatsu Mira engine 660 cc. the design is considering the step to fabricated, cost of production and reliability of the conceptual design. The CAM works as a converter from the rotational motion from the DC motor to the linear motion in order to move the pulley.

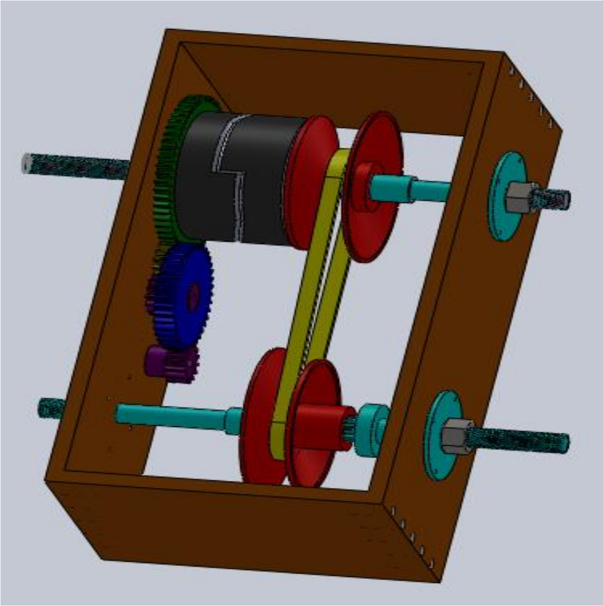
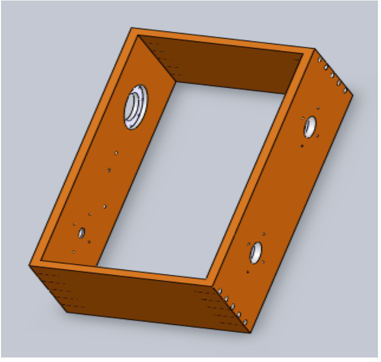


Figure 4.1: Design of CVT Transmission Using CAM

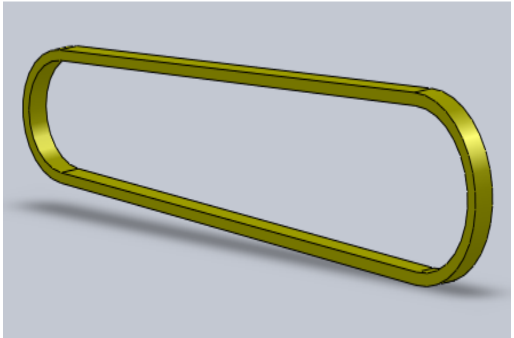
4.2.1 DETAIL OF PART AND DESCRIPTION

The selected design is choosing after consideration the function of each part of the design. The function of each part from the model is showing in Table 4.1 below.

Table 4.1: Detail of Part and Description by Solidworks Software

Part	Descriptions
1.**Casing** 	Place where CVT system running. Very hard to support heavy CAM and pulleys. Made from aluminum alloy.

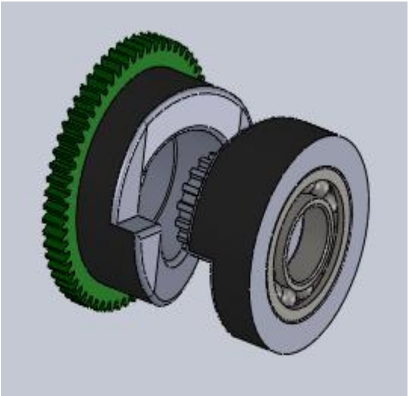
2. **Belt**



This v belt drive is to transmit rotational motion and torque from driving to driven pulley.

Made of fabric and cord, usually cotton, rayon, or nylon, and impregnated with rubber.

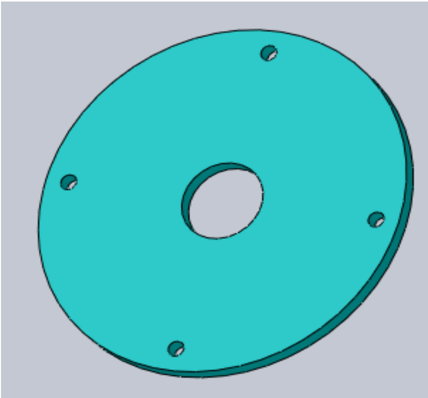
3.**Female and Male CAM**



Main mechanism to move the moveable sheave to change the diameter of belt at driver pulley.

Made from Carbon Alloy to prevent wear and has high hardness.

4.**Radial Bearing Cover**

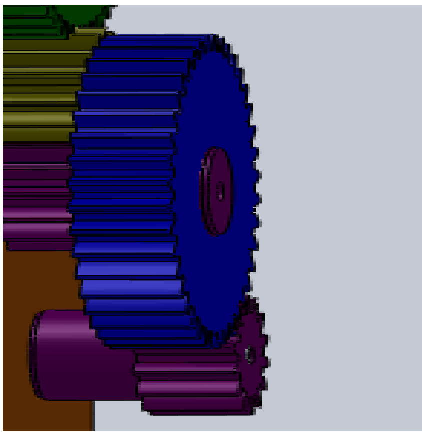


This part functions as cover for radial ball bearing that installed in the casing.

Other then also provide support for the bearing well rotating.

Made from steel.

5.**Gear Set**

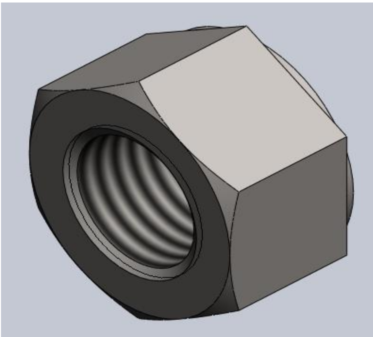


The set of gear is the connection medium between the DC motor and the CAM.

It's amplified the torque from the motor to CAM.

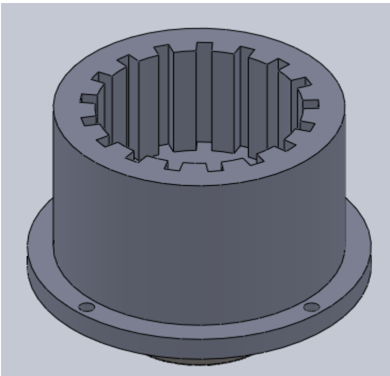
Made from Cobalt steel.

6.**Hex Bolt Nut**



This bolt is to tighten the shafts to the casing.

7.**CAM Sitter**



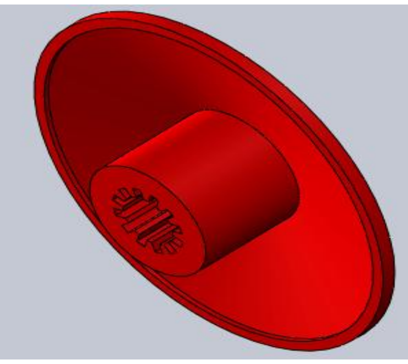
This part provides support for CAM and preventing it from rotate with the shaft.

Other than that this part also enable the Male CAM to move in X axis.

8.**Fix Sheave**

This is the side of the pulley where it only rotates with the shaft.

9. **Movable Sheave**



The sheave is the moveable sides for the pulley which change enable the belt to change diameter in both driver and driven pulleys.

4.3 ANALYSIS OF GEAR RATIO

One of the most important operations to be studied in this project is about the gear ratio. Gear ratio is the main problem to all manufacturers to control it. The best of CVT is determined from its gear ratio controller because the main issues gear ratio transfer power from engine to drive the pulley and the driven pulley drives the wheels. At any time, the most suitable ratio should be decided so that performance and energy efficiency are both optimized. Theoretically it looks very simple, but the implementation is very difficult.

4.3.1 CAM Parameter

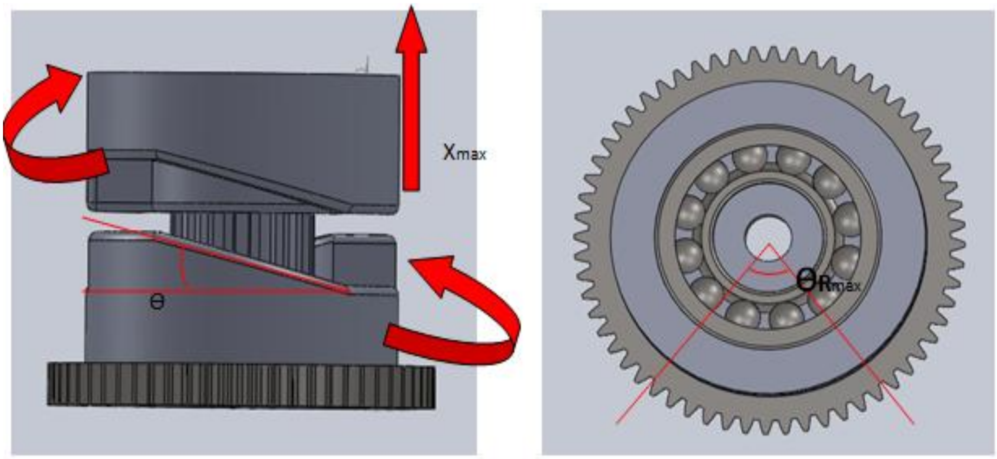


Figure4.2: Side and Top View of Female & Male CAM

$$\Theta = 14.01°$$

$$\Theta Rmax = 75.08°$$

$$Xmax = 1.5\ cm$$

The relation between the maximum rotation angle of the cam, ӨRmax and the distance of increasing, Xmax is when the cam start to rotate from 0° to 75.08° it also increase the distance of X from 0cm to 1.5cm.

So,

X= Xmax/ ӨRmax

=1.5cm/75.08°=0.2 mm/°

Thus, distance will be increase by 0.2mm per degree of rotaion.

4.3.2 Effect of CAM Movement

The relation between ϴ and the ϴRmax is also same with the above statement, that the rotating of cam from 0° to 75.08° will affect the X to increase from 0cm to 1.5cm by the constant angle of inclination 14.01°.

Other calculation to prove the increasing of x direction by degree of inclination 14.01° is:

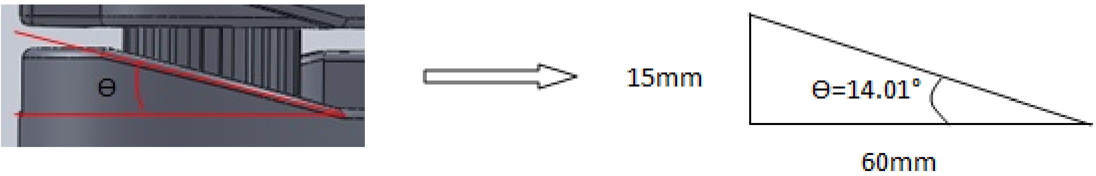


Figure4.3: Side View of CAM With Angle Inclination 14.01°

$$\frac{60}{75.08} = 0.8mm/1°$$

If the cam moving of ϴRmax by 1°,

$$tan14.01° = \frac{x}{0.8}$$
$$x = 0.2mm$$

Table 4.2: Relation Movement of CAM and X direction

ϴR(°)	ϴ (°)	x(cm)
0.00	14.01	0.00
10.00	14.01	0.1998
20.00	14.01	0.3996
30.00	14.01	0.5994
40.00	14.01	0.7992
50.00	14.01	0.999
60.00	14.01	1.1988
70.00	14.01	1.3986

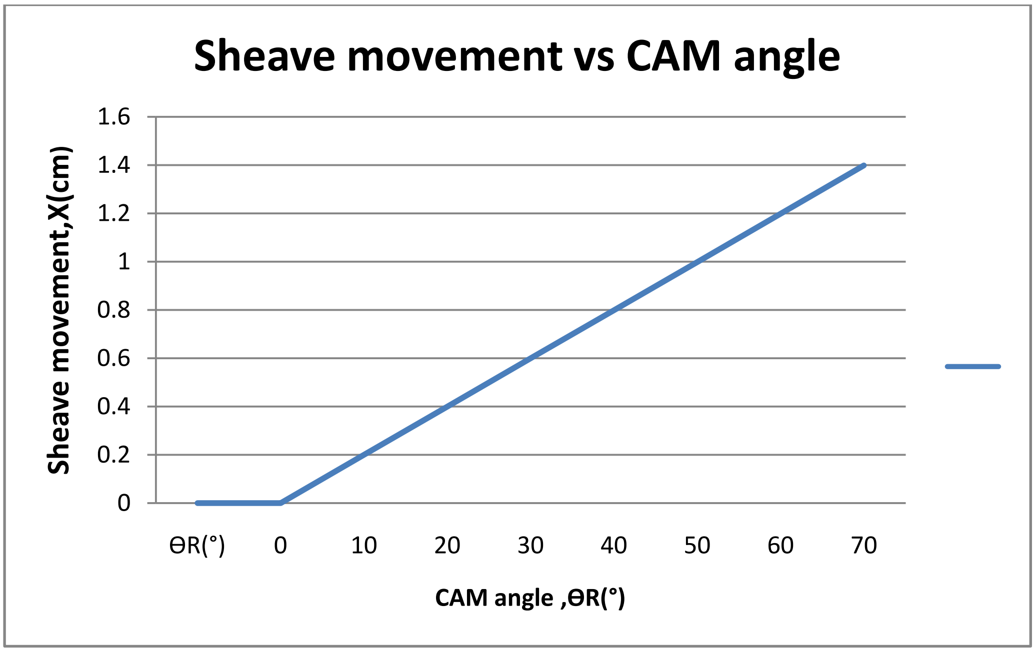


Figure 4.4: Sheave Movement vs CAM Angle Graph

In every 1° of CAM movement the CAM will expand 0.2mm. The expansion of CAM is directly proportional to the distance travel by moveable sheave at driver pulley. The relation is shown in Figure 4.4.

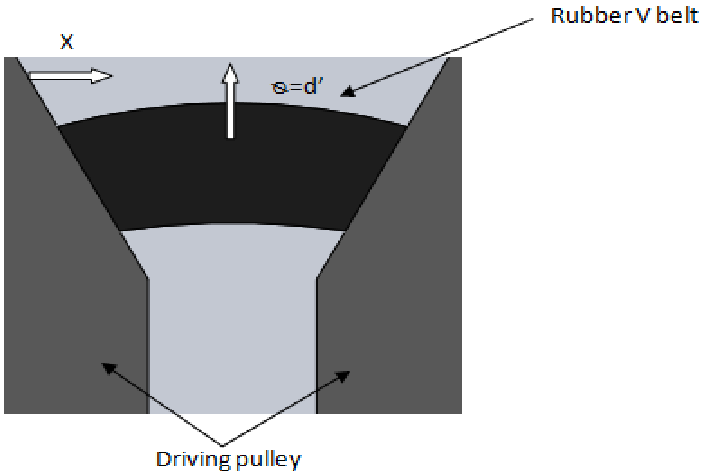


Figure 4.5: Belt and Pulley Movement

Initial d=40mm

If the ƟRmax rotated 1°, the cam expand 0.2mm due to inclined surface and push the moveable pulley in X direction increasing the diameter of rubber belt or in other word increase the diameter of driving pulley.

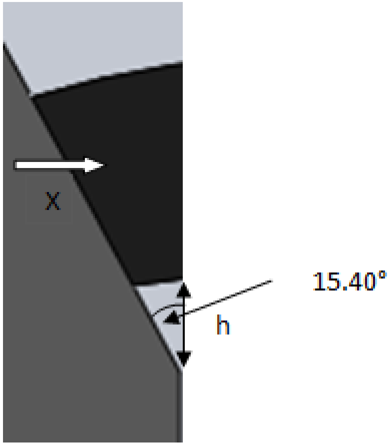


Figure4.6: Detail View of Belt and Pulley Movement

$$x = 0.2mm,$$

$$tan15.40° = \frac{x}{h}$$

$$tan15.40° = \frac{0.2mm}{h}$$

$$h = 0.726mm$$

h is the increment radius of the v belt at driver pulley and the increment influence the ratio of the driver and driven pulley diameter.

$$d' = 40mm$$

$$dnew = 40 + (0.726 \times 2)$$

$$= 41.452mm$$

4.4 Belt and Pulley Geometry

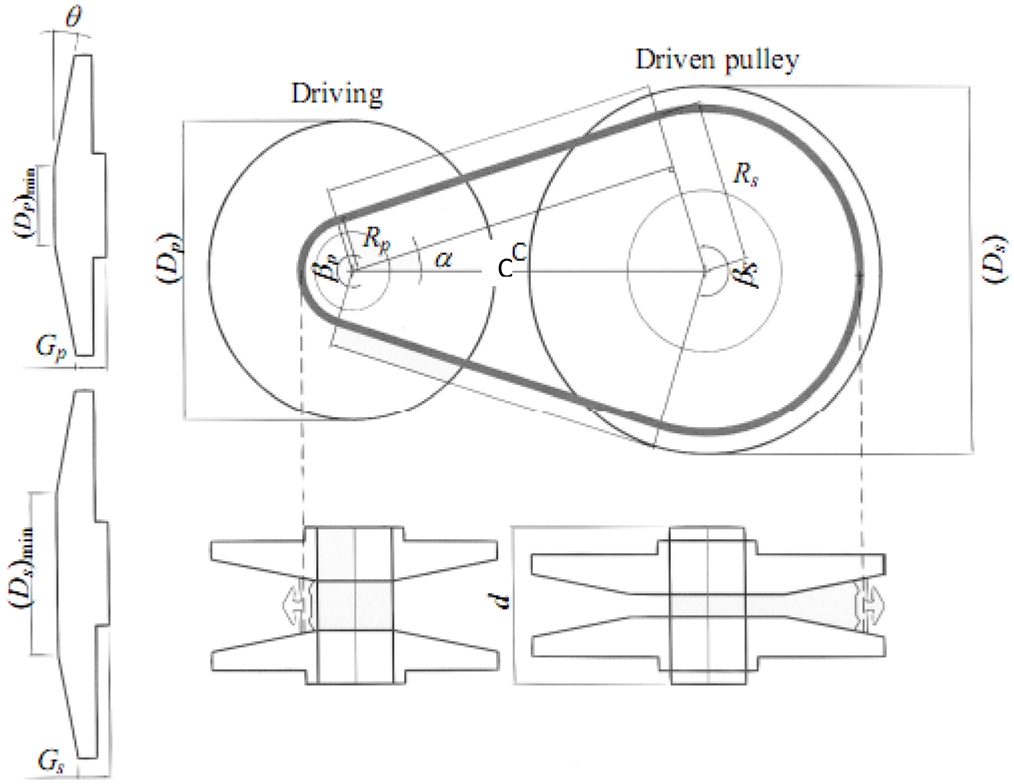


Figure 4.7: Basic Belt Drive Geometry

$$c = 208mm$$

$L\ belt = 650mm$ (from manufacturer)

For the first gear, realistic diameter for Dp in this design is 40mm. So, diameter for Ds can be calculate using this equation:

$$L_p = \frac{\pi(d + D)}{2} + 2C + \frac{(D - d)^2}{4C}$$

$$650 = \frac{\pi(40 + D)}{2} + 2(208) + \frac{(D - 40)^2}{4(208)} =$$

$$D = 105.67mm$$

The inclination angle,α for this 1st gear can be calculated using this equation:

$$L = Rp(\pi - 2\alpha) + Rs(\pi + 2\alpha) + 2\sqrt{c^2 - (Rs - Rp)^2}$$

$$650 = 20(\pi - 2\alpha) + 52.835(\pi + 2\alpha) + 2\sqrt{208^2 - (52.835 - 20)^2}$$
$$\alpha = 9°$$

$$\beta p = \pi - 2a = \pi - 2(0.157) = 2.83\, rad$$

$$\beta s = \pi + 2a = \pi + 2(0.157) = 3.46\, rad$$

Table 4.3: Parameter of Pulley and CAM

Rp (mm)	**Rs (mm)**	**Belt Length (mm)**	**Cam Rot. (deg.)**	**Gear Ratio**
24.40	51.97	649.68	0.0	2.13
25.25	51.25	649.68	2.6	2.03
26.10	50.53	649.68	5.1	1.94
26.94	49.80	649.68	7.7	1.85
27.78	49.07	649.68	10.3	1.77
28.62	48.34	649.68	12.9	1.69
29.45	47.60	649.68	15.4	1.62
30.28	46.85	649.68	18.0	1.55
31.11	46.10	649.68	20.6	1.48
31.93	45.35	649.68	23.1	1.42
32.75	44.59	649.68	25.7	1.36
33.57	43.83	649.68	28.3	1.31
34.38	43.07	649.68	30.9	1.25
35.19	42.30	649.68	33.4	1.20
35.99	41.52	649.68	36.0	1.15
36.80	40.75	649.68	38.6	1.11
37.59	39.96	649.68	41.1	1.06

38.39	39.18	649.68	43.7	1.02
39.18	38.39	649.68	46.3	0.98
39.96	37.59	649.68	48.9	0.94
40.75	36.80	649.68	51.4	0.90
41.52	35.99	649.68	54.0	0.87
42.30	35.19	649.68	56.6	0.83
43.07	34.38	649.68	59.1	0.80
43.83	33.57	649.68	61.7	0.77
44.59	32.75	649.68	64.3	0.73
45.35	31.93	649.68	66.9	0.70
46.10	31.11	649.68	69.4	0.67
46.85	30.28	649.68	72.0	0.65
47.60	29.45	649.68	74.6	0.62
48.34	28.62	649.68	77.1	0.59
49.07	27.78	649.68	79.7	0.57
49.80	26.94	649.68	82.3	0.54
50.53	26.10	649.68	84.9	0.52
51.25	25.25	649.68	87.4	0.49
51.97	24.40	649.68	90.0	0.47

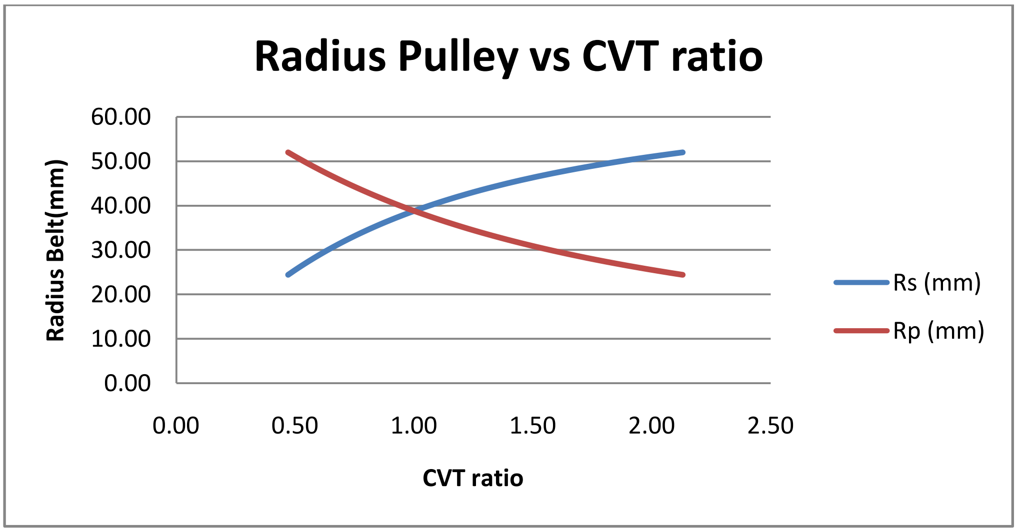


Figure 4.8: Radius Pulley vs CVT Ratio

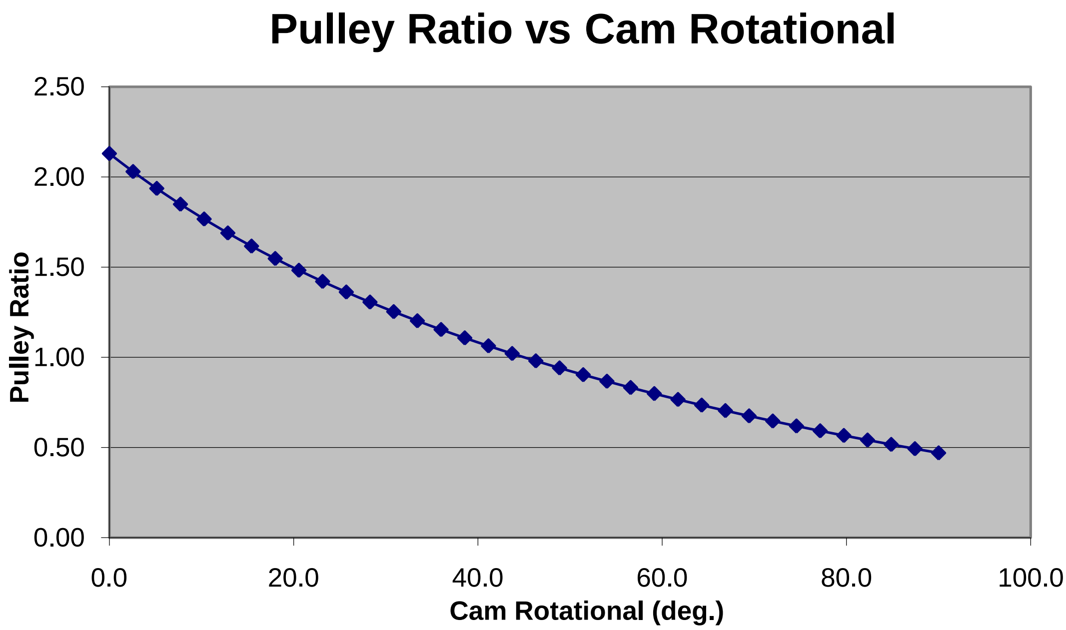


Figure 4.9: Pulley Ratio vs CAM Rotational

4.5 Force analysis of CAM relative to RPM of the driver pulley

The force acting on the pulley relative to rpm of the shaft is basically the same as CVT that used centrifugal force. Therefore the analysis of force can be done by referring to the centrifugal force concept and relate it to the new design. The analysis begin by determining the maximum centrifugal force acting on the driver pulley at the maximum RPM. The maximum centrifugal force at maximum RPM will be used as a reference for new mechanism design.

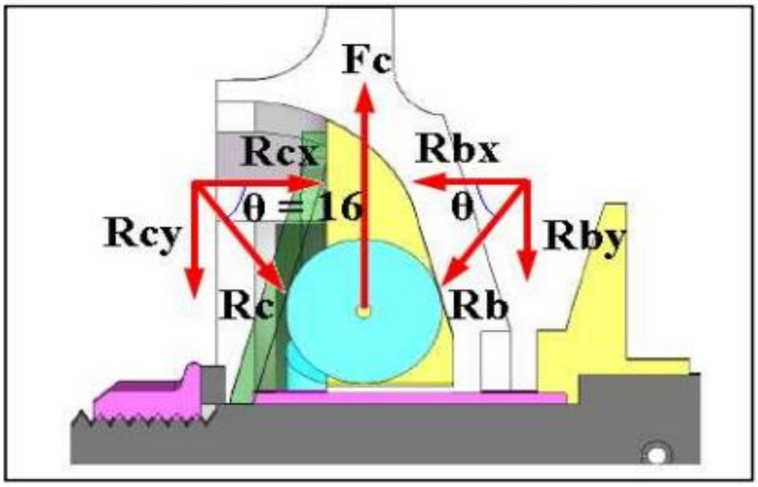


Figure 4.10: free Body Diagram of Roller in The Centrifugal force CVT

4.5.1 Engine RPM and Centrifugal forces, Fc of shaft.

According to Newton's second law of motion,

Force= mass × centrifugal acceleration

$$F = m \times \frac{v^2}{r}$$

$$= \frac{mv^2}{r}$$

The linear velocity and angular velocity is a follow this equation,

Linear velocity =angular velocity × radius of the path

$$v = r \times \omega$$

Hence, sub v into F= $m \times \frac{v^2}{r}$

$$= \frac{m(\omega r)^2}{r}$$

$$\text{Fc}=mr\omega^2$$

Let the minimum RPM for an engine to be 1000 RPM

$$\omega = \frac{2\pi/RPM}{60}$$

$$m = 0.06kg$$

$$r = 0.035m$$

So,

$$Fc = (0.06)(0.035)(\frac{2\pi \times 1000}{60})^2$$

$$=23N$$

Table 4.4: Relation Fc with RPM

Fc,*N*	RPM
23	1000
92	2000
207	3000
368.5	4000
575.7	5000
829.0	6000
1128.4	7000
1473.8	8000

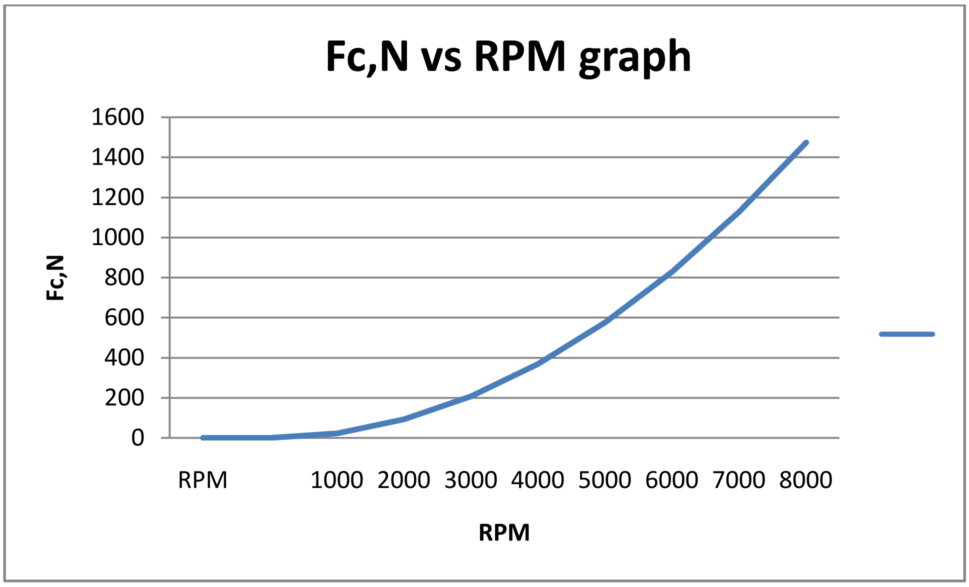


Figure 4.11: Fc,N vs RPM Graph

Therefore at RPM 6000, the centrifugal force is 829.04*N*. So, in this design by using CAM it is need requirement torque to move the distance between two pulleys. The pitch radius of the gear attach at the Female CAM, r=62mm=0.062m.

$$\tau = Fc\max \times r$$
$$\tau = 892.04 \times 0.062$$
$$\tau = 51.04N$$

The CAM needed 51.04 N forces to push the moveable sheave at 6000 RPM. Therefore the DC motor used must at that capability to amount of torque. However the torque from CAM is step down by the train gears connecting CAM and DC motor. With that the torque produce by the DC motor can be less than the 51.04N.

4.6 Gear ratio analysis

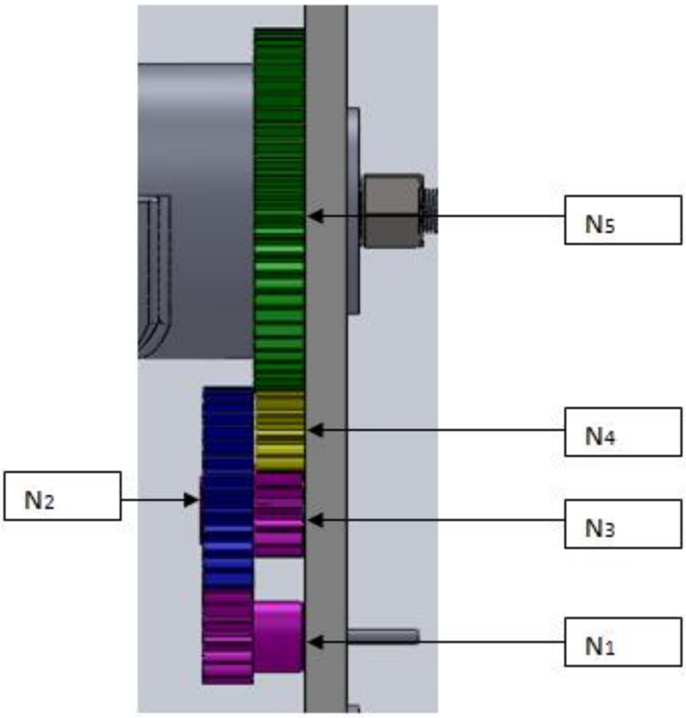


Figure 4.12: Train Gears

N5=60t

N4=40t

N3=N2=N1=14

Assume N5 has the v=1rev/min.

So,

$$v4 = \frac{n5}{n4} v4$$

$$v4 = 4.3 rev/min$$

$$v2 = v3 = v4,$$

$$v1 = \frac{n2}{n1}(4.3)$$

$$= \frac{40}{14}(4.3)$$

$$v1 = 12.3\ rev/min$$

Therefore the motor have to 12.3 rev in order to turn 1 rev of the cam means by ratio of 1:12.3

$$1rev = 360°$$

In order to turn the cam at maximum angle ϴRmax =75.08° to increase the distance of X from 0 to 15mm the motor have to turn:

$$\frac{75.08}{360°} = 0.2rev$$

$$motor\ rev = \frac{0.2}{1} 12.3 = 2.46\ rev$$

4.7 DC motor analysis

After knowing the maximum torque and also the speed gear ratio attach between the DC motor and the CAM, the choice of motor have to be suitable and fulfill the minimum specification in order to provide perfect control for driver to control the transmission ratio.

The ratio of the gear is 1:12.3, that means the torque that need to supply by the DC motor is 12.3 times less than the amount it need to turn the CAM at 6000 RPM. Therefore amount of the motor is:

$$\tau' = \frac{51.04N.m}{12.3}$$

$$\tau' = 4.14N.m$$

Therefore the motor must be capable to produce 4.14N.m or more of torque in order to mechanism to function perfectly until it reach it limit. The choice of motor is as follow.

Figure 4.13: BY88BL DC brushless motor (BY88BL120)

Table 4.5: DC Motor Specification

Parameter Name	Parameter Value	Units
Rated Voltage	24	V
Electric Resistance (R)	4	Ohm(Ω)
Rated Speed	1480	rpm
No Load Speed	1950	rpm
Moment of Inertia of the Rotor (J)	3.2284E-6	$\mathrm{kg.m^2/s^2}$
Rated current	7	A
No Load Current	0.5	A
Rated Torque	8	N.m
Winding Type	Y	
Damping Ratio of the Mechanical System (b)	3.5077E-6	N.m.s

Power	120	watts
Electromotive Force Constant (K=Ke=Kt)	0.0274	N.m/Amp
Electric Inductance (L)	2.75E-6	H

Source: http://www.brushlessmotor.cn/BLDC/BY88BL110.pdf

Based on the table provided by ‘ BOYANG Brushless DC motor’ model (BY88BL120) is the best selected motor based on rated torque 8N.m that just over the requirement of the CVT design that need 4.14N. The no load speed for this model is 1950 rpm. So, the speed for the CAM to finish the 75° rotation is as follow:

Motor speed is 1480 RPM=24.666 rev/s

Gear ratio 1:12.3

CAM speed,v’,

$$v' = \frac{24.666}{12.3}$$

$$v' = 2.0\, rev/s$$

Time to complete 75.08° rotation,

$$2\, rev = 720°$$

$$v' = 720°/sec$$

$$t = \frac{75.08°}{720°/sec}$$

$$t = 0.104\, s$$

The time to complete the rotation of the CAM is 0.104 sec, and it consider too fast to control. Therefore, the equip DC driver is important to lower the speed and also the torque.

4.7 Analysis Using PID Controller

After complete the mathematical analysis and getting all the requirement that need to be control using the PID controller such as the position of DC motor to be $\theta(t)$ =923.48° deg of rotation or 16.12rad because the unit of theta will be discusses in radian. The analysis continued with the PID controller using MATLAB software in order to control DC motor to suicide desired position. From the mathematical analysis also state that settling time must be done at t=0.104sec.

4.8 Simulation without PID Controller

The detailed and explicit block for DC motor without PID controller is shown in figure.

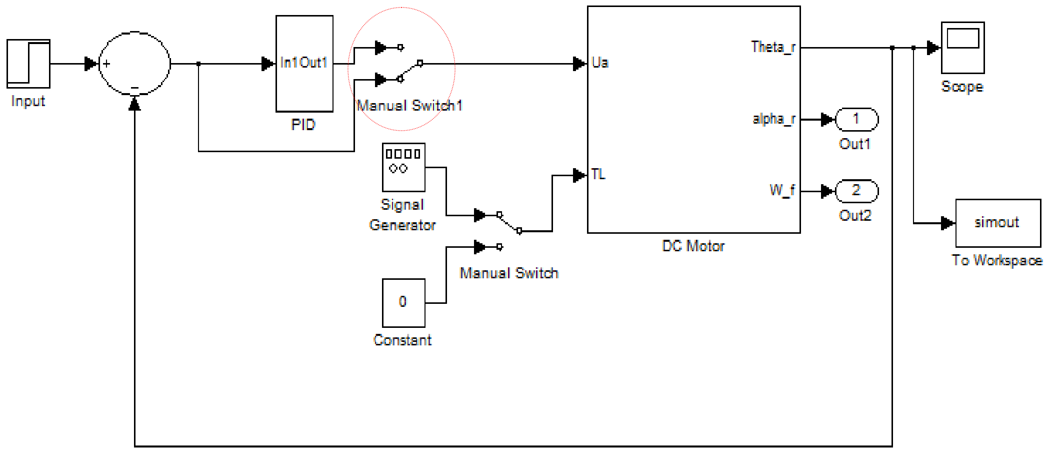


Figure 4.14: Simulation without PID Controller

The DC motor parameter ra, La, Bm, ka and J is entered in the simulation block. Assigning the desired reference position of the DC motor to be $\theta(t)$ =923.48° deg of rotation or 16.12rad because the unit of theta will be discusses in radian. Other desired output to be discusses is the time complete. After the simulation start, the graph will shown the position (rad) vs. time(secs).

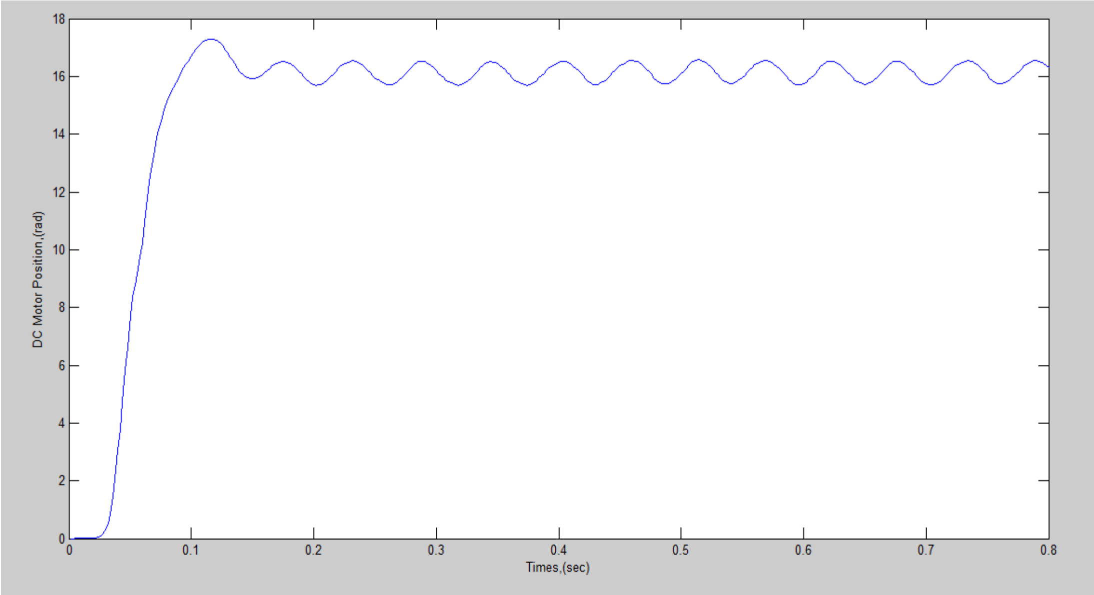


Figure 4.15: Motor position vs. Times Graph without PID Controller

From the plots above we can see that steady-state error is too large and the settling time is too long. Recall from PID controller theory that tuning an integral term will eliminate the steady-state error and a derivative term will reduce the overshoot. Let's first try a PI controller in order to get rid of the disturbance steady-state error. Change the PI parameter, so it looks like:

Kp =15and Ki=25

4.9 Simulation with PI Controller

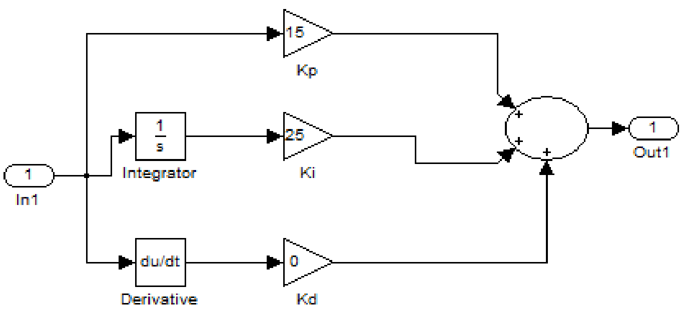


Figure 4.16: PI Controller Parameter

The integral term in a PI controller causes the steady-state error to reduce to zero, which is not the case for proportional-only control in general. The lack of derivative action may make the system steadier in the steady state in the case of noisy data. This is because derivative action is more sensitive to higher-frequency terms in the inputs. Without derivative action, a PI-controlled system is less responsive to real (non-noise) and relatively fast alterations in state and so the system will be slower to reach set point and slower to respond to perturbations.

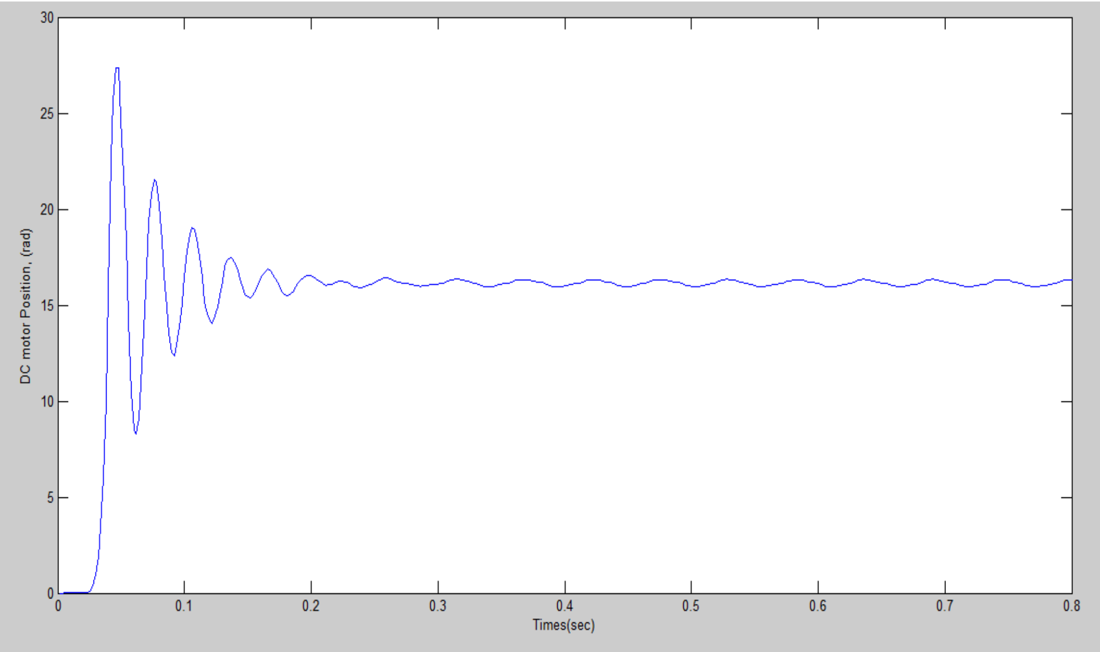


Figure 4.17: Motor position vs. Times Graph with PI Controller

After response to the disturbance of step input Kp =15 and Ki=25, we can see the steady-error is been reduce. But increase the integral will also increase overshoot.

Proportional Control	PI Control	PID Control
Stable	Maintain Stability	Derivative term reduces overshoot, settling time
Faster Response = Bigger Overshoot	Decrease Steady State Error = Bigger Overshoot	Feed Forward
Steady State Error		Overcome damping

For the last one, tuning the PID controller to reduces overshoot, settling time. So the term of Kp, Ki and Kd it looks like:

Kp =15

Ki =25

Kd =0.15

4.10 Simulation with PID controller

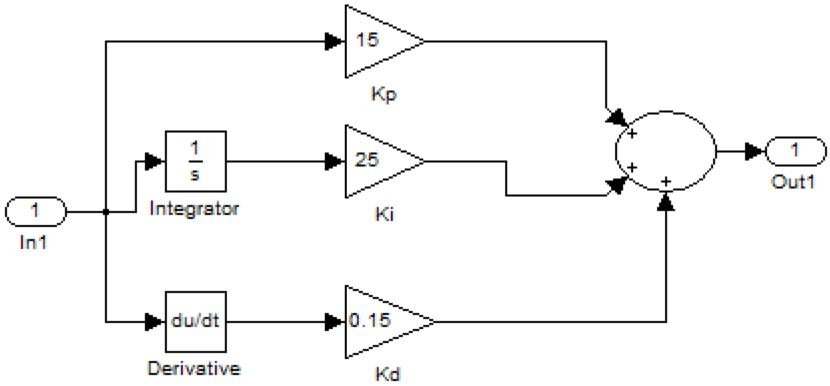


Figure 4.18: PID Controller Parameter

The PID controller simulation involves three separate parameters, and is accordingly sometimes called three-term control: the proportional, the integral and derivative values, denoted P, I, and D. The proportional value determines the reaction to the current error, the integral value determines the reaction based on the sum of recent errors, and the derivative value determines the reaction based on the rate at which the error has been changing.

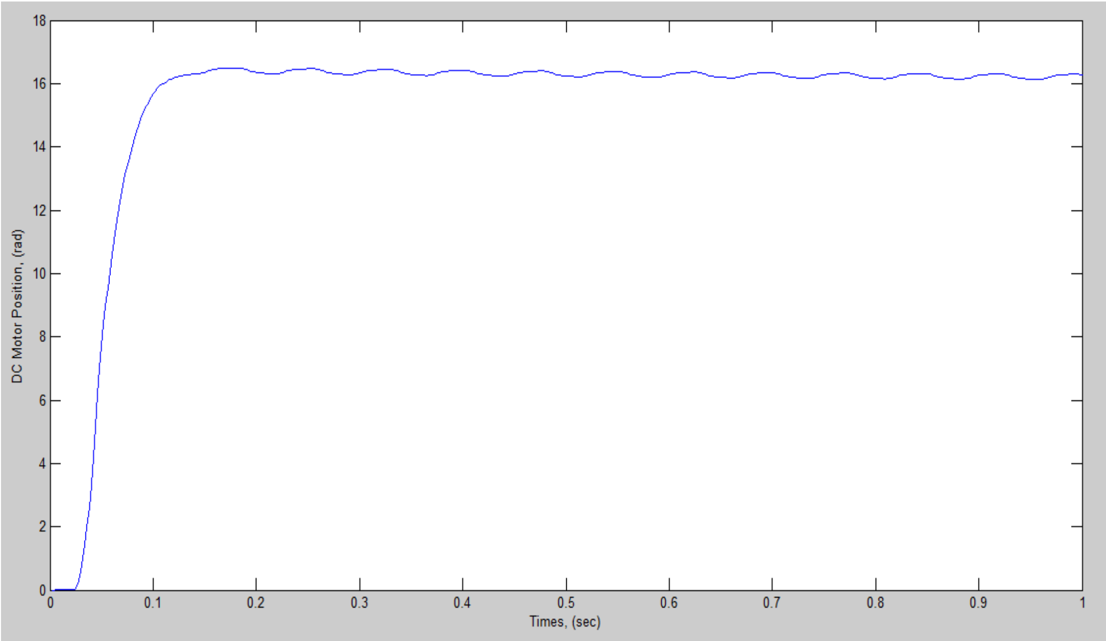


Figure 4.19: Motor position vs. Times Graph with PID Controller

From the above plot, we can see that the settling is roughly ≅ 0.14s, it has less than 0.05 overshoot, and it has small steady state error. So now, if the tuning using the PID controller with

Kp =15

Ki =25

Kd =0.15

All of my design requirement will be satisfied.

CHAPTER 5

CONCLUSION AND RECOMMENDATION

5.1 CONCLUSION

MATLAB and Simulink is very user-friendly software, through which control system is design using various block provided. Simulink save a lot of time by avoiding hundred lines of coding. MATLAB and simulink are used for simulation and for designing real-time model. There is an inherit advantages in using Simulink to model the control system. It saves time and effort, allowing the engineer to design the system in a straight forward manner, rather than wasting time writing source code from scratch. Only recently has simulink had the capability to directly target hardware. It was now possible to create Simulink models for motor testing, open loop system design, as well as closed-loop system design without writing any lines of code.

The basic objective of this thesis is to control the position of DC motor using the PID controller software because the requirement of the CVT-design to control the maximum angle of the can be only <75.08° instead of 2.57 of rotation. So, the aim of this thesis to control the motor to make the position of rotation to be 16.12rad in order to achieve the maximum angle of the cam. The objective is satisfied from the result, its mean the degree of rotation Cam is under control.

5.2 RECOMMENDATION

Lots of future work can be done to exploit the advantages of MATLAB and Simulink and their hardware targeting capabilities. This thesis used a simple structure of PID controller, a complicated structure can be chosen to obtain better output. There are various advances tuning method such as Robust Adaptive PID (RaPID) could be used in designing PID controller other than Ziegler Nichols methods and trial and error methods. An intelligent controller could also be used for position control of DC drives, so the combination of intelligent controller and PID controller can be used for better control of motor position. Analytical of other component is also further study for improvement of selected design. The inclination of the pulley needs more analytical research to get the optimum result.

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Appendix A

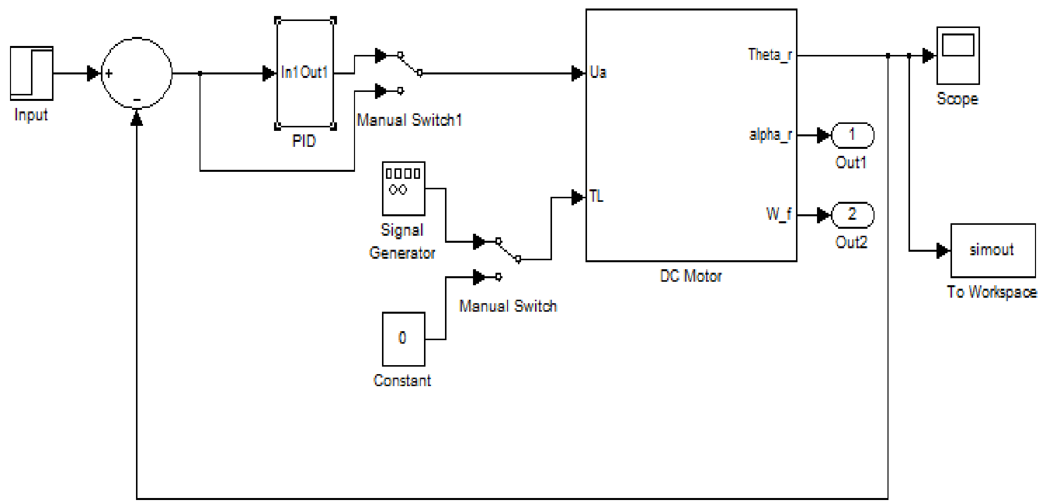

Figure A1 : Simulink Block of PID Control DC Motor (Simulation)

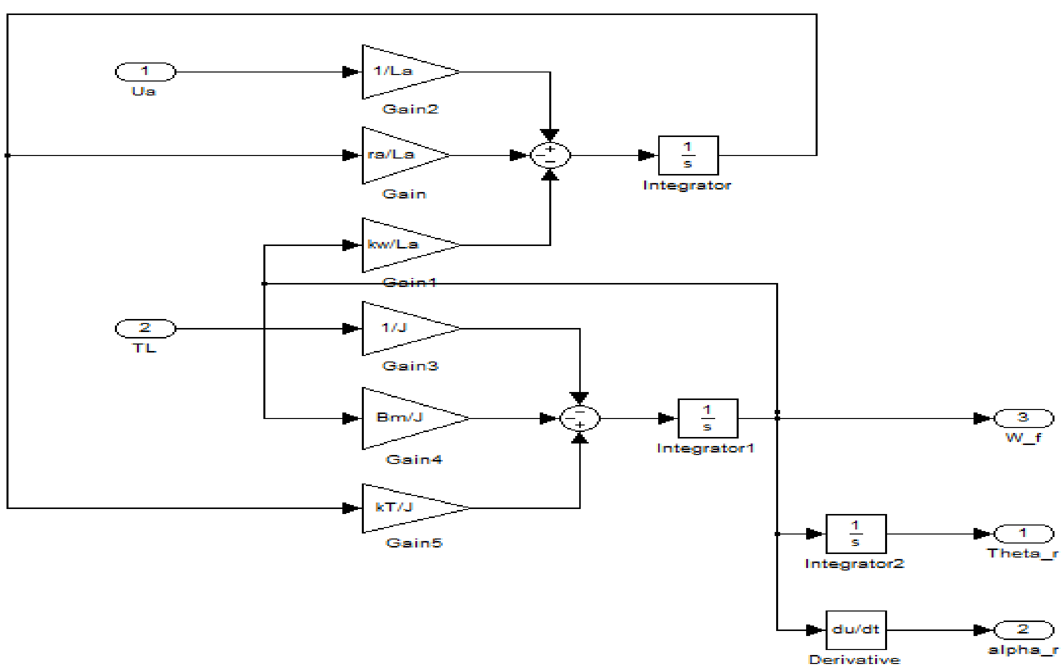

Figure A2 : Simulink Block of DC Motor

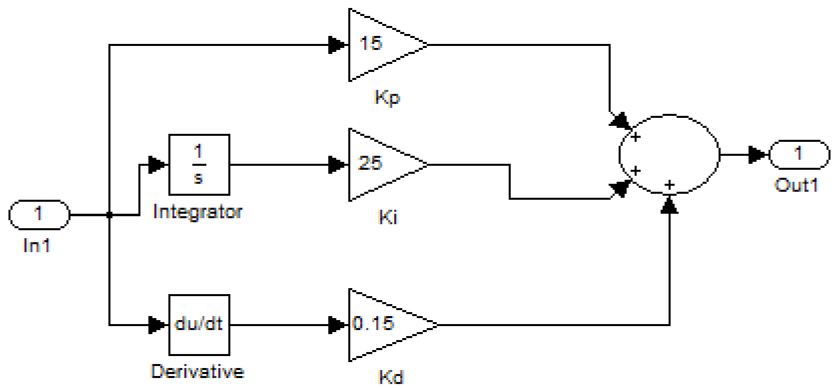

Figure A3 : Simulink Block of PID controller

Num.	alpha (deg.)	Bp (rad)	Bs (rad)	c (mm)	m (mm)	n (mm)	Rp (mm)	Rs (mm)	Belt Length (mm)	Gear Ratio	Yp (mm)	Ys (mm)	Xp max (mm)	Xp (mm)	Xs (mm)	Cam Rot. (deg.)
1	7.80	2.869	3.414	201.12	27.56	24.40	24.40	51.97	649.68	2.13	27.60	0.03	7.458	0.000	7.458	0.0
2	7.36	2.885	3.398	201.33	26.00	25.25	25.25	51.25	649.68	2.03	26.75	0.75		0.230	7.264	2.6
3	6.91	2.900	3.383	201.52	24.43	26.10	26.10	50.53	649.68	1.94	25.90	1.47		0.459	7.068	5.1
4	6.47	2.916	3.367	201.71	22.86	26.94	26.94	49.80	649.68	1.85	25.06	2.20		0.687	6.872	7.7
5	6.02	2.931	3.352	201.88	21.29	27.78	27.78	49.07	649.68	1.77	24.22	2.93		0.914	6.674	10.3
6	5.57	2.947	3.336	202.04	19.72	28.62	28.62	48.34	649.68	1.69	23.38	3.66	**Xs max**	1.140	6.475	12.9
7	5.13	2.963	3.321	202.19	18.14	29.45	29.45	47.60	649.68	1.62	22.55	4.40	**(mm)**	1.366	6.275	15.4
8	4.68	2.978	3.305	202.32	16.57	30.28	30.28	46.85	649.68	1.55	21.72	5.15	7.458	1.590	6.074	18.0
9	4.24	2.994	3.289	202.45	15.00	31.11	31.11	46.10	649.68	1.48	20.89	5.90		1.814	5.871	20.6
10	3.79	3.009	3.274	202.56	13.42	31.93	31.93	45.35	649.68	1.42	20.07	6.65		2.037	5.668	23.1
11	3.34	3.025	3.258	202.65	11.84	32.75	32.75	44.59	649.68	1.36	19.25	7.41		2.258	5.463	25.7
12	2.90	3.040	3.243	202.74	10.27	33.57	33.57	43.83	649.68	1.31	18.43	8.17		2.479	5.257	28.3
13	2.45	3.056	3.227	202.81	8.69	34.38	34.38	43.07	649.68	1.25	17.62	8.93		2.699	5.049	30.9
14	2.01	3.072	3.212	202.88	7.11	35.19	35.19	42.30	649.68	1.20	16.81	9.70		2.918	4.841	33.4
15	1.56	3.087	3.196	202.92	5.53	35.99	35.99	41.52	649.68	1.15	16.01	10.48		3.136	4.632	36.0
16	1.11	3.103	3.181	202.96	3.95	36.80	36.80	40.75	649.68	1.11	15.20	11.25		3.353	4.421	38.6
17	0.67	3.118	3.165	202.99	2.37	37.59	37.59	39.96	649.68	1.06	14.41	12.04		3.568	4.210	41.1
18	0.22	3.134	3.149	203.00	0.79	38.39	38.39	39.18	649.68	1.02	13.61	12.82		3.783	3.997	43.7
19	-0.22	3.149	3.134	203.00	-0.79	39.18	39.18	38.39	649.68	0.98	12.82	13.61		3.997	3.783	46.3
20	-0.67	3.165	3.118	202.99	-2.37	39.96	39.96	37.59	649.68	0.94	12.04	14.41		4.210	3.568	48.9
21	-1.11	3.181	3.103	202.96	-3.95	40.75	40.75	36.80	649.68	0.90	11.25	15.20		4.421	3.353	51.4
22	-1.56	3.196	3.087	202.92	-5.53	41.52	41.52	35.99	649.68	0.87	10.48	16.01		4.632	3.136	54.0
23	-2.01	3.212	3.072	202.88	-7.11	42.30	42.30	35.19	649.68	0.83	9.70	16.81		4.841	2.918	56.6
24	-2.45	3.227	3.056	202.81	-8.69	43.07	43.07	34.38	649.68	0.80	8.93	17.62		5.049	2.699	59.1
25	-2.90	3.243	3.040	202.74	-10.27	43.83	43.83	33.57	649.68	0.77	8.17	18.43		5.257	2.479	61.7

APPENDIX B : Belt Calculation Using Microsoft Excel