POSITION TRACKING OF SLIDER CRANK MECHANISM USING PID CONTROLLER OPTIMIZED BY ZIEGLER NICHOL'S METHOD

Fauzi Ahmad[1], Ahmad Lukman Hitam[2], Khisbullah Hudha[3], and Hishamuddin Jamaluddin[4]

[1, 2,3)] Smart Material and Automotive Control (SMAC) Group
Center of Vehicle Research & Development (CeVReD)
Faculty of Mechanical Engineering
Universiti Teknikal Malaysia Melaka (UTeM)
Karung Berkunci 1200, Hang Tuah Jaya, Ayer Keroh 75450 Melaka, Malaysia
E-mail1): fauzi.ahmad@utem.edu.my,
E-mail2): lukh2u@yahoo.com,
E-mail3): khisbullah@utem.edu.my

[4)] Faculty of Mechanical Engineering,
Universiti Teknologi Malaysia (UTM),
81310 UTM Skudai, Johor, Malaysia
E-mail: hishamj@fkm.utm.my

ABSTRACT

This paper presents a study on the position tracking response of a Propotional-Integral-Derivative (PID) controlled- slider crank mechanism, which is driven by a two phase stepper motor. In this study, the rod and crank are assumed to be rigid where the Newton second law is applied to formulate the equation of motion. A position tracking control of the slider crank mechanism is then developed by using PID controller. Several tests such as saw tooth, step function and square function are used in order to examine the performance of the proposed control structure. The results show that, the proposed control structure is able to tracking the desired position with a good response. The slider crank mechanism rig is then developed to investigate experimentally the ability of the proposed controller structure. The results show that the proposed control structure is able to track the desired displacements with accepTABLE error.

***KEYWORD**: PID control, slider crank mechanism, stepper motor.*

1.0 INTRODUCTION

In mechanical engineering, the slider crank mechanism is a basic structure that have been applied in many usage such as fretsaws, petrol and diesel engines. Due to its mechanical coupling, the physical sense is not enough to derive its dynamic equations. Jasinski *et al*.(1971); Zhu and Chen (1983) and Badlani *et al.* (1979) have solved the steady state solutions of a slider crank few years ago. According to Viscomi *et al.* (1971), the response of slider crank is dependent on length, mass, damping, external piston force and frequency. Based on the viewpoints of the ratios, length and speeds of the crank to the connecting rod, the transient responses have been investigated (Fung, 1996). Recently, the slider crank mechanisms are actuated by the field oriented control PM synchronous (Leonard, 1996; Novotny *et al.* 1996 and Lin *et al.* 1998). The slider crank mechanism driven by PM synchronous is used to transfer rotational motion to translation motion.

Nowadays, advancements in magnetic materials, semiconductor power devices, and control theory have made the PM synchronous servo motor drive plays a vitally important role in motion-control applications in the low-to-medium power range. The desirable features of the PM synchronous servo motor are its compact structure, high air-gap flux density, high power density, high torque-to-inertia ratio, and high torque capability. Moreover, compared with an induction servo motor, a PM synchronous servo motor has such advantages as higher efficiency, due to the absence of rotor losses and lower no-load current below the rated speed; and its decoupling control performance is much less sensitive to the parametric variation of the motor (Leonard, 1996 and Novotny *et al.* 1996). To achieve fast four-quadrant operation and smooth starting and acceleration, the field-oriented control or vector control, is used in the design of the PM synchronous servo motor drive (Lin *et al.* 1998),.

However the control performance of the PM synchronous servo motor drive is still influenced by the uncertainties of the controlled plant, which usually comprise unpredicTABLE plant parametric variation, external load disturbances, unmodelled and nonlinear dynamics. During the past decades controlling of slider crank position have resulted with various control strategies to be developed. Numerous control methods such as: adaptive control; neural control; and fuzzy control have been studied (Visioli, 2001; Seng *et al.* 1999; Krohling *et al.* 2001; Mitsukura *et al.* 1999 and Kawabe T *et al.* 1997). Among these the best known is the proportional integral derivative (PID) controller, which has been widely used in the industry because of its simple structure and robust performance within a wide range of operating conditions (Huang HP *et al.* 2002; Cominos P *et al.* 2002; Chuang *et al*. (2006); Ahmad *et al.* (2010) and Kristiansson *et al.* 2002). In this paper, the formulation and dynamic behavior of a PM synchronous motor coupled with a complexity mechanical system is introduced where a slider crank mechanism system actuated by a PM synchronous servo motor is investigated. MATLAB-Simulink software is chosen as a computer simulation tool used to simulate the system's behavior and evaluate the performance of the control structure.

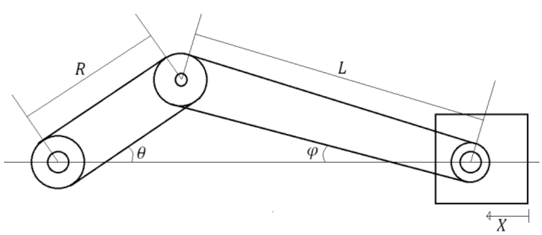
In controlling the mechanical system with a good response, a PID controller is designed to control the position of the coupled mechanism roller The proposed control structure based feed back control is consists of inner loop controller and outer loop controller (Ahmad *et al.* (2010); Kristiansson *et al.* (2002 Yukitom *et al.,* 2004). The inner loop controller is used for position tracking control of the motor actuator while the outer loop controller is used to track the position of the slider shaft. Simulation studies for the slider crank mechanism model are presented in order to demonstrate the effectiveness of using the proposed controller. Several test have been performed namely sine wave function test, square function test, step function test and saw tooth function test. The simulation results show that the use of the proposed PID control technique proved to be effective in controlling the position of the slider crank with a good accuracy.

Since the explorations of the proposed controller have been done in the simulation study, the slider crank mechanism rig is needed to investigate experimentally the capability of the proposed controller. The mechanism of the slider crank system is consists of an actuator such as stepper motor to actuate the slider crank in a real slider mechanism system. The result of the experiment studies show that the control technique is able to track the desired position with a small deviation and accepTABLE error.

The remainder of this paper is organized as follows: The first section contains the introduction and the review of some related works, followed by mathematical derivations of slider crank kinematics model with stepper motor model in the second section. The third section presents the proposed control structure for the position tracking control of slider crank mechanism system. The fourth section will explain about The performance evaluation of the proposed control structure. The following section will discuss about the setup and the performance of the slider crank mechanism in the experimental study and the last section contains some conclusion.

2.0 SLIDER CRANK MECHANISM MODELING

The slider crank mechanism is a basic structure in mechanical application. It is also widely used in practical application (Nagchaudhuri, 2002; Fung *et al.,* 1999; Ranjbarkohan *et al.,* 2011). For examples, fretsaws, petrol and diesel engines are the typical application of velocity control. Hence, the slider crank mechanism considered in this study is based on the basic operation of engine which consists of a crankshaft, R, connecting rod, L, and piston. The purpose of the slider-crank mechanism is to convert rotational motion of the crankshaft to the linear motion of the piston. Like shown in FIGURE 1, the kinematic of slider crank mechanism can be described in equation 1 to 7.



The piston displacement from top dead centre, x, can be determined from the geometry of the mechanism, in terms of the lengths of the con-rod, L, and crank, R, and the crank angle, θ. From the geometry and noting that $\theta = \varphi = 0$ when $x = 0$, x can be expressed as:

$$X = R - R\cos\theta + L - L\cos\varphi \qquad (1)$$

Also from the geometry, it can be seen that

$$L \sin \varphi = R \sin \theta \tag{2}$$

and

$$[L \cos \varphi]^2 = L^2 - [L \sin \varphi]^2$$
(3)

Substituting for $L \sin \varphi$ from Equation 2 in Equation 3 and leaves θ as the only variable on the right hand side of the expression,

$$[L \cos \varphi]^2 = L^2 - [R \sin \varphi]^2$$
(4)

Equation 4 can be substituted into Equation 1 to obtain the kinematic equation for the slider crank mechanism such as Equation 5,

$$X = R - R \cos \theta + L - \sqrt{L^2 - [R \sin \theta]^2}$$
(5)

Equation 5 can then be rearranged by introducing another parameter, *n*, the ratio of the length of the conrod, *L*, to the radius of crankshaft, *R*, as:

$$X = R \left\{ 1 - \cos \theta + n \left[1 - \sqrt{1 - \left(\frac{\sin \theta}{n} \right)^2} \right] \right\} \tag{6}$$

where

$$n = \frac{L}{R} \tag{7}$$

The values of parameters *R* and *n* are determined by measurement of the slider crank mechanism. The technical specifications of the slider crank mechanism are listed in **TABLE 1**.

TABLE 1 slider crank mechanism parameters

parameters	**value**
R	100mm
L	300mm

TABLE 2 Stepper motor model parameters

Parameters	**value**
Operating value U	24 V
Static holding torque T_m	0.5 Nm
Winding resistance R	100 ohms
Electric time constant $_E$	5 ms
Stepping angle	1.8 degree
Numbers of pole pairs n	50
Load inertia J	1.0 kgm^2

2.1 STEPPER MOTOR MODELING

In this study, the slider crank mechanism is driven by two phase stepper motor which is used to provide the rotational motion to the crank shaft. The stepper motor is consisting of one stator side and one rotor with one pole pair that function as the permanent magnet. When the windings of one phase are energized, a magnetic dipole is generated on the stator side. The basic principle of the stepper motor is given in FIGURE 2. For example, if phase 2 is active, winding 3 produces an electrical south pole and winding 4 produces an electrical north pole.

The number of steps per revolution of the rotor is calculated as:

$$S = 2.n.m \tag{8}$$

Where n is the number of rotor pole pair and m is the number of stator phases. For the hybrid stepper motor, n is half number of rotor teeth.

The stepping angle is

$$\Delta\emptyset = \frac{360}{S} \tag{9}$$

For example, if n=1 and m=2, it will have 4 steps per revolution and stepping angle is 90 degrees. If sinusoidal characteristic of the magnetic field in the air gap is assumed, the contribution of each phase j on the motor torque T_{Mj}
Can be written as

$$T_{Mj} = k_m \sin\left[n\emptyset(t) + \emptyset_{0j}\right].I_j(t) \tag{10}$$

Where

k_m	motor constant, depending on the design of the rotor
$\emptyset(t)$	actual motor position
$\emptyset_{0j}$	location of the coil j in the stator
$I_j(t)$	the current in the coil as function of time.

The current $I_j(t)$ in the coil is a function of the supplied voltage U_j and the coil properties. A general between U_j and $I_j(t)$ is given by:

$$U_j = emf + R.I(t) + L\frac{dI(t)}{dt} \tag{11}$$

Where

emf_j the electronic force induced in the phase j

R the resistance of the coil

L the inductance of the coil

The *emf* in each coil can be expressed as:

$$emf_j = k_m \sin\left[n\emptyset(t) + \emptyset_{0j}\right].\omega \tag{12}$$

Resistance and inductance of all coils in the motor are the same so that no indices are required. The differential equation can be expressed in the LAPLACE domain as shown in the equation below:

$$I = \frac{U}{L_S+R} \tag{13}$$

The total torque produced by the stepper is :

$$T_m = \sum_{j=1}^{m} T_{Mj} \tag{14}$$

Considering the equation of motion of the stepper motor

$$\sum_{j=1}^{m} T_{Mj} = J\frac{d\omega}{dt} + D\omega + T_F \tag{15}$$

Where

J the inertia of the rotor and the load

D the viscous damping constant

T_f frictional load torque

Where ω is the rotational velocity of the rotor and the data of the stepper motor is given in the **TABLE 2**.

2.2 DESCRIPTION OF THE SIMULATION MODEL

The slider crank simulation model was developed based on the mathematical equations presented in the previous section by using MATLAB SIMULINK software. The relationship between slider crank mechanism and stepper motor are clearly described in FIGURE 3. There are two inputs that can be used in the analysis of the slider crank namely torque input and position input which come from the stepper motor. But in this study, the position of rotor is used to become the input to the slider crank mechanism. It simply explains that the model created is able to perform the position tracking control analysis of the slider crank mechanism.

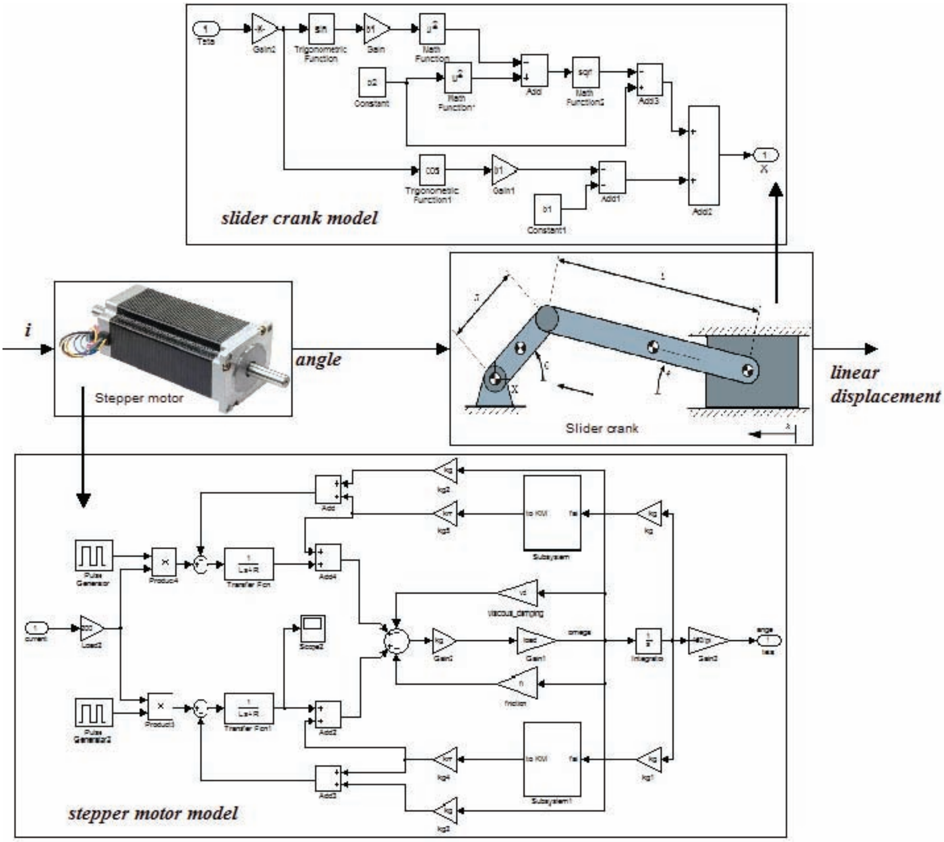


FIGURE 3 Slider crank mechanism model in Matlab Simulink Software

3.0 POSITION TRACKING CONTROL OF SLIDER CRANK MECHANISM

There are two loops used in the controller structure which are inner loop and outer loop controller. The inner loop controller is used to evaluate the deviation from the commanded position and the encoder which detects the position of the rotor with robust and accuracy tracking performance and an outer loop controller is used as the position controller for the tracking of periodic reference inputs. The proposed control structure is shown in **FIGURE 4**.

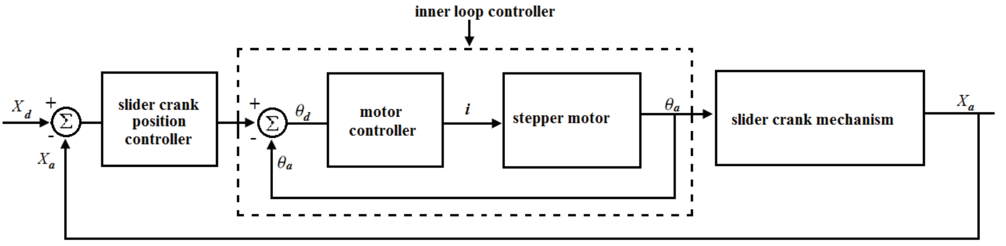


FIGURE 4 The proposed control structure for position tracking control of slider crank mechanism

In this control scheme, the proportional plus integral plus derivative (PID) controller is used in the outer loop control to isolate the piston displacement from top dead centre, *x*. The reason for using PID controller is because the PID controller has already proven effective in many applications where it is easy to maintain and easy to implement in the real system (Chuang *et al.,* 2006). The PID controller that applied in the system mathematically can be described by equation (16).

$$X(t) = K_P(T) + K_i(t)\int e(t)\,dt + K_d(e)\frac{d}{dt}e(t) \qquad (16)$$

Where $X(t) = X_d(t) - X_a(t)$ and X_a is the piston actual displacement, X_d is the piston desired displacement, the proportional gain, $K_p(t)$, integral gain, $K_i(t)$ and derivative gain, $K_d(t)$ are the function of the position error piston displacement.

Even PID controllers are probably the most commonly used controller structures in industry, however it present some challenges to control and instrumentation in the aspect of **tuning** of the gains required for stability and good transient performance. Because of that, there are several prescriptive rules used in PID tuning such as Ziegler-Nichole's method.

3.1 ZIEGLER NICHOLES FOR AUTO TUNING

In 1942 Ziegler and Nichols, both employees of Taylor Instruments, described simple mathematical procedures for tuning PID controller. The procedures are now accepted as standard in control systems practice. The Ziegler-Nichols formulae for specifying the controllers are based on plant step responses. The method is applied to plants with step responses of the form displayed in FIGURE 5. This type of response is typical of a first order system with transportation delay, such as that induced by fluid flow from a tank along a pipe line. It is also typical of a plant made up of a series of first order systems. The response is characterized by two parameters, *L* the delay time and *T* the time constant. These are found by drawing a tangent to the step response at its point of inflection and noting its intersections with the time axis and the steady state value. The plant model is therefore:

$$G(s) = \frac{K_{e-sl}}{T_s+1} \tag{17}$$

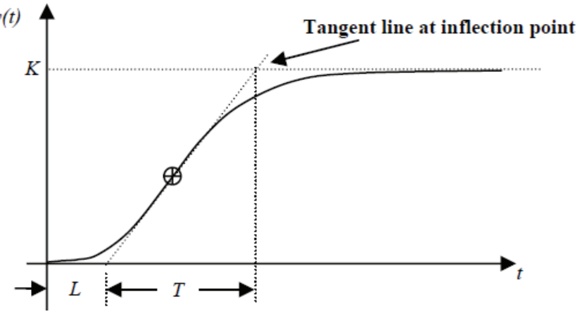


FIGURE 5 Response curve for Ziegler-Nichols method

Based on the equation, Ziegler and Nichols derived the control parameters such as in TABLE 3:

TABLE 3 Ziegler and Nichols control parameters

PID Type	K_p	$T_i = K_p/K_i$	$T_d = K_d/K_p$
P	$\frac{T}{L}$	∞	0
PI	$0.9\frac{T}{L}$	$\frac{L}{0.3}$	0
PID	$1.2\frac{T}{L}$	$2L$	$0.5L$

4.0 PERFORMANCE ASSESMENT OF THE PROPOSED CONTROL STRUCTURE TO THE POSITION TRACKING CONTROL OF SLIDER CRANK MECHANISM

The performance of the PID controller in tracking the desired position is examined through simulation studies using SIMULINK toolbox of the MATLAB software package. For comparison purposes, the performance of the proposed PID control structure is compared with the desired displacement which is the reference. The desired specifications are settling time t_s= 0.5 sec, rising time t_r=0.25 sec, maximum overshoot M_p< 5% and steady state error e_{ss} < 1%.

4.1 SIMULATION PARAMETERS

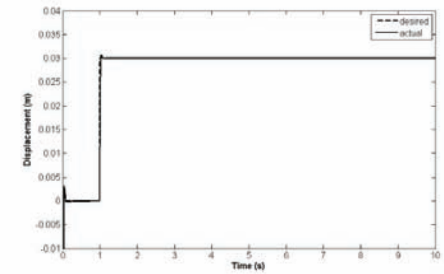
The simulation study was performed for a period of 10 seconds using Heun solver with a fixed step size of 0.01 second. The controller parameters are obtained using trial and error technique as shown in TABLE 4. The numerical values of the slider crank model are defined in **TABLE 1** and the stepper motor model parameters are as in TABLE 2 adopted from Morar, (2003).

TABLE 4 Controller parameters

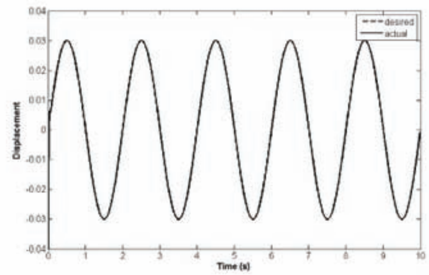
PID	Value
K_p	995
Ki	79030
K_d	0

4.2 SIMULATION RESULT

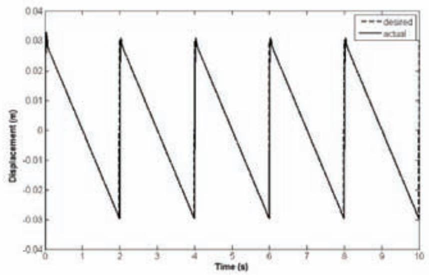
FIGUREs 6(a) to FIGURE 6(b) show the response for system with the PID controller. The parameters of the PID controller is optimized using Ziegler Nichols method for optimal performance under various condition. Several test procedures such as step function, sine wave function, square function and saw tooth function are applied to verified the effectiveness of the control structure. All the input signals are generated from 'step function' and 'Signal Generator' block with the amplitude set at 0.03m and frequency at 0.03 Hz.The results of the simulation runs corresponding to the trajectory profiles of the slider as discussed before. With appropriate tuning of the PID gains, excellent results are achieved as illustrated in FIGUREs 6(a) to FIGUREs 6(d). In the graphs, the dashed line corresponds to the desired motion of the slider position at the 0.03m and the solid line indicates the actual motion achieved by controlling the position piston of the slider crank. It can be seen that the proposed control structure with PID controller in driving the crank is very encouraging as shown in step function response in FIGURE 6(a). In term of sine wave function, FIGURE 6(b), saw tooth function, FIGURE 6(c), and square function, FIGURE 6(d) the controller structure shows it's capability in achieving control design criteria such as discussed in section 4.0 via providing a good response in tracking the desired position..



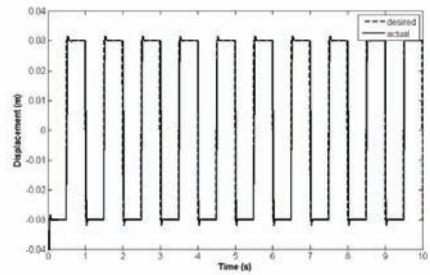
(a). Step function



(b). Sine wave function



(c). Saw tooth function



(d). Square function

FIGURE 6 Simulation responses of the position tracking control of the slider crank mechanism

5. PERFORMANCE ASSESMENT OF THE PROPOSED CONTROL STRUCTURE USING HARDWARE IN THE LOOP SIMULATIONS OF SLIDER CRANK MECHANISM

In order to demonstrate the effectiveness of the proposed control rule, slider crank mechanism rig has been setup as shown in FIGURE 7.The experimental instrument of slider crank is divided into five parts such as actuator, slider crank, controller, host PC and target PC. In this system, the simulation of the slider crank is simulated in the host PC, while the host PC is used to give the direction to the target PC to interact with the hardware which means the slider crank mechanism. In this study, the slider crank is coupled with the stepper motor that consists of a stepper motor driver. The driver is worked on 2-phase, 220 V and 60 Hz. To measure the translation position, a sensor namely linear variable displacement transducer (LVDT) is used. The output of the LVDT is 0~5V, which mapped to real translation position is 0~0.2m. On the other hand, another one sensor is employed to measure the angular position of the crank namely angular encoder. The used of the encoder is to give feed back control in the inner loop control. The data acquisition system namely National instrument (NI) interface card (Advantech CO., PCL-1800) is installed in the ISA bus to handle the A/D and D/A process.

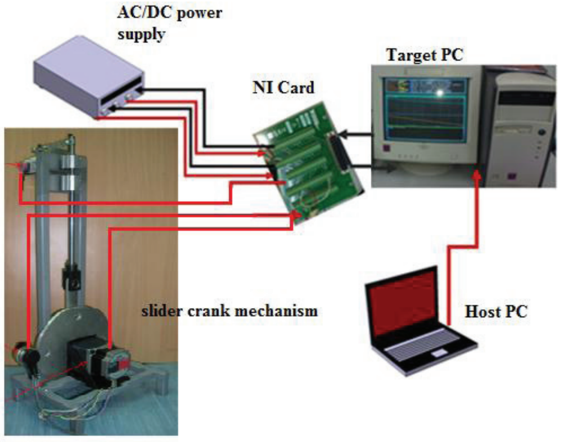
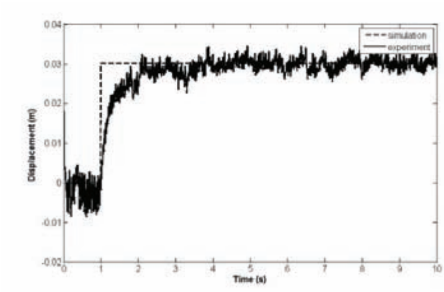


FIGURE 7 Slider crank mechanism experimental setup

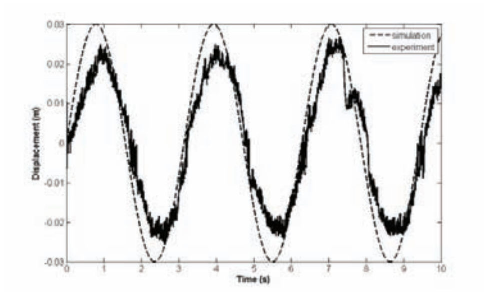
5.1 Experimental result

Based on the same requirement of simulation, the experimental results are shown in **FIGUREs** 8(a) to 8(d) with the differences signal applied for the test. From the graphs, it is necessary to note that the dashed line is the desired position, while the solid line is the response of the system. FIGURE 8(a) shows the step function responses of the system. It can be seen that the trends between the desired position and actual responses are slightly similar but didn't fulfill the control design criteria as discussed in section 4.0. The control strategy is over damp where the rise time of the system is more then 0.25 second, no over shoot and the steady state error is more then 1 %. From observation, this is happened because of the frictions that happened between the piston with the bearing where the surface of the piston is rough. Other than that, the control optimization was also the most factors contributing to this problem. The parameters of the controller were optimized (with assume) the system is in ideal conditions and no friction happen at all the joints and the touching surface.

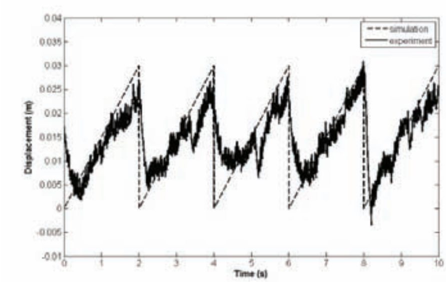
In term of sine wave function in FIGURE 8(b) and saw tooth function in FIGURE 8(c), the system shows the tendencies to follow the desired position with the similar shape but have a little bit differences in term of magnitude. These differences are caused by the friction that lowering down the speed of the system where the system was not yet finished implementing the old instruction but already arrived the new order from target PC. In term of square function, (FIGURE 8(d)), the result shows that, the control structure try to force the system (following) the desired position and as a result, there is a small difference in term of trends and magnitude. Again, this is due to the frictions that occur in the system. This is due to the fact that, during the simulation, the efficiency of the stepper motor's rotating shaft, the mass of connecting rod and the sliding shaft or piston were ignored and neglected while in the actual condition, all of these should be taken into account.



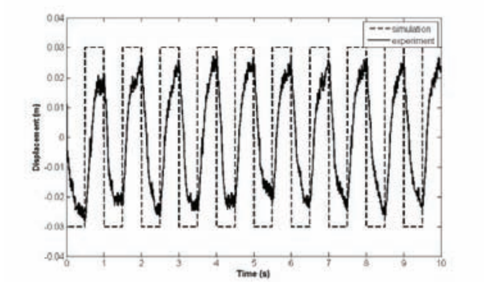
(a). Step functions



(b) Sine wave function



(c). Saw tooth function



(d). Square function

FIGURE 8 Response of hardware in the lop simulation of position tracking control of slider crank mechanism

6.0 CONCLUSION

As a conclusion, the kinematics model of slider crank mechanism has been developed and integrated with stepper motor model by using MATLAB SIMULINK software. The position tracking control of the slider crank mechanism is then has been developed which consist of two close loop function namely inner loop and outer loop controller. The inner loop controller is used to command the stepper motor to give the necessary motion such desired by the reference, while the outer loop control structure is used to instruct the slider crank to give the required position as needed. In this study, the Proportional-Integral-Derivative (PID) controller is used as the controller strategy. The reason for using PID controller is because the PID controller has already proven effective in many applications where it is easy to maintain and easy to implement in the online system. Simulation studies for the slider crank mechanism model are presented to demonstrate the effectiveness of using the proposed controller. Several tests have been performed in order to verified the effectiveness of the proposed controller namely sine wave function test, square function test, step function test and saw tooth function test. The simulation results show that the use of the proposed PID control technique proved to be effective in controlling the position of the slider crank with a good accuracy. A slider crank mechanism test rig is then has been developed to validate experimentally the efficiency of the proposed control technique. Same testing methods have been implemented and the results show that the control technique is able to track the desired position with a small deviation and accepTABLE error.

7.0 REFERENCES

Ahmad, F.,Imaduddin, F., Hudha, K. and Jamaluddin, H. (2010). Modelling, validation and adaptive PID control with pitch moment rejection of active suspension system for reducing unwanted vehicle motion in longitudinal direction. *International Journal of Vehicle System Modeling and Testing (IJVSMT) Vol. 5 No. 4 pp. 312-346*, Inderscience Publishers, UK

Viscomi, B. V. and Arye, R. S. (1971). Nonlinear dynamic response of elastic slider crank mechanism," *ASME J. Eng. Ind.,* pp. 251-262.

Boot, R., Richert, J. (1998). Automated test of ECUs in a Hardware-In-The-Loop Simulation Environment. ASIM 1998 12th Symposium on Simulation Technology,

Chuang, C. W., Lee, C. D. and Huang, C. L. (2006). Applying experienced self-tuning PID control to position control of slider crank mechanisms. *International Symposium on Power Electronics, Electrical Drives, Automation and Motion, 2006. (SPEEDAM 2006).* Taormina, Sicily May 2006, pp. 652 – 657, ISBN: 1-4244-0193-3

Cominos, P, Munro N. (*2002).* PID Controllers: Recent Tuning Methods And Design To Specification. *IEE Proceeding on Control Theory Application.* pp. 46–53.

Novotny, D. W. and Lipo, T. A. (1996). Slider Crank Mechanism Control Using Adaptive Computed Torque Technique "Vector Control And Dynamics Of AC Drivers," *Oxford University,*

Lin F.I., Lin Y.S. and Chiu S.L. Slider-crank mechanism control using adaptive computed iorque technique, JEE Proceedings Control Theory and Applications, 1998, pp.364-276.

Hanselmann, H. (1993). Hardware-in-the-Loop Simulation as a Standard Approach for Development, Customization, and Production Test. SAE 1993-930207

Huang, H.P., Roan, M.L. and Jeng, J.C. (2002). On-line Adaptive Tuning for PID Controllers". *IEEE Proceeding on Control Theory Application 2002,* pp. 60–72.

Kawabe, T. and Tagami, T. (1997). A real coded genetic algorithm for matrix inequality design approach of robust PID controller with two degrees of freedom". *IEEE Proceeding on 12th International Symposium Intelligent Control.* Istanbul, Turkey, July 1997. pp. 119–24.

Kristiansson, B. and Lennartson. B. (*2002).* Robust and Optimal Tuning of PI and PID Controllers. *IEEE Proceedingon Control Theory Application. Pp. 46–53.*

Krohling, R. A. and Rey, J.P. (2001).Design Of Optimal Disturbance Rejection PID Controllers Using Genetic Algorithm. *IEEE Transaction on Evolution Computer 2001;Vol. 5, pp.78–82.*

Badlani, M. and Kleinhenz, W. (1979). Dynamic Stability Of Elastic Mechanisms. *Journal ofMechanism Des.* pp. 149-153.

Mitsukura, Y., Yamamoto, T. and Kaneda, M. (1999). A Design Of Self-Tuning PID Controllers Using A Genetic Algorithm. *Proceeding of Am Control Confefernce, San Diego, CA,* June 1999. Vol. 5. pp. 1361.

Morar, A., (2003). Stepper Motor Model for Dynamic Simulation. *Acta Electrotehnica.* Vol. 44(2), pp. 117-122.

Jasinski, P. W., Lee, H.C. and Sandor, G. N. (1971).Vibrations of elastic connecting rod of a high-speed slider crank mechanism. *ASME J. Eng. Ind.,* pp. 636-644.

Park, T. J., Oh, S. W., Jang, J. H. and Han, C.S. (2002). The Design of a Controller for the Steer-by-Wire System Using the Hardware-In-the-Loop-Simulation System. *SAE Automotive Dynamics and Stability.*No.01, pp.1596.

Fung, R. F. (*1996).* Dynamic analysis of the flexible connecting rod of a slider crank mechanism. *ASME J. Vibration Acoust.* pp. 687-689,

Seng, T. L., Khalid, M. B., Yusof, R. (1999). Tuning of a Neuro-Fuzzy Controller by Genetic Algorithm. *IEEE Trans on Syst Man Cybern B 1999.* No.29, pp.226–36.

Terwiesch, P., Keller, T. and Scheiben, E. (1999). Rail vehicle control system integration testing using digital hardware-in-the-loop simulation. IEEE Trans Control Syst Technol 1999;7(3):352–62.

Visioli A. (2001) Tuning of PID controllers with fuzzy logic. *Proceeding of Electrical Engineering Control Theory Application.*Vol.148, No. 1, pp.1–8.

Yukitomo, M., Shigemasa, T. and Baba, Y. and Kojima, F. (2004). A Two Degrees of Freedom PID Control System, its Features and Applications. *2004 5th Asian Control Conference*. Melbourne, Australia. Vol. 1, pp: 456ISBN: 0-7803-8873-9, July 2004

Zhu, Z. G. and Chen, Y. (1983). The Stability of the Motion of a Connecting Rod," *J. Mechanisms, Transmissions, Automation Des.* pp. 637-640,

Nagchaudhuri, A. (2002). Mechatronic Redesign Of Slider Crank Mechanism. *Proceeding of IMECE2002, ASME International Mechanical Engineering Congress & Exposition.* No. IMECE2002-32482. November 17-22, New Orleans, Louisiana United State of America.

Fung, R. F., Chen, K. W. and Yen, J. Y. (1999). Fuzzy Sliding Mode Controlled Slider Crank Mechanism Using A PM Synchronous Servo Motor Drive. International Journal of Mechanical Sciences . No. 41(1999), pp. 337-355.

Ranjbarkohan, M., Rasekh, M., Hoseini, A. H., Kheiralipour, K. and Asadi, M. R. (2011). Kinematics And Kinetic Analysis of The Slider-Crank Mechanism In Otto Linear Four Cylinder Z24 Engine. Journal of Mechanical Engineering Research Vol. 3(3), pp. 85-95, March 2011