International Journal of Electrical and Computer Engineering (IJECE)
Vol. 4, No. 5, October 2014, pp. 658~667
ISSN: 2088-8708

Robust Control Strategy for Pneumatic Drive System via Enhanced Nonlinear PID Controller

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Article Info

Article history:

Received Jun 22, 2014
Revised Aug 24, 2014
Accepted Sep 10, 2014

Keyword:

pneumatic positioning system
NPID
SNF
dead-zone compensation
robustness

ABSTRACT

This paper presents the pneumatic positioning system controlled by Enhanced Nonlinear PID (NPID) controller. The characteristic of rate variation of the nonlinear gain that are readily available in NPID controller is utilized to improve the performance of the controller. A Self-regulation Nonlinear Function (SNF) is used to reprocess the error signals with the purpose of continuously generating the values for the rate variation. Subsequently, the controller has successfully been implemented on dynamically changing loads and pressures. The comparison with the other available method such as. NPID and conventional PID are performed and evaluated. The effectiveness of this method with Dead Zone Compensator (DZC) has also been successfully demonstrated and proven through simulations and experimental studies.

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1. INTRODUCTION

Pneumatic actuators are widely used in industries to perform many tasks such as pick and place application, gripping, clamping, drilling and spraying. They are categorized under fluid power control and apply the principles of using compressed gas as a source of power to perform a variety of tasks. It deals with mechanical properties of gases and offer several advantages such as simple to maintain, fast motion, low cost, high power to weight ratio, free from overheating and reliable [1]. The ability to operate at a high number of cycles per workday is also one of the advantages of this drive. Due to these advantages, it has been promoted as an alternative to hydraulics and electric servo motors in many automated tasks. In spite of these advantages, pneumatic actuators are subject to nonlinearities due to compressibility of air, high friction forces and dead band of the spool movement in the valve [2]. These nonlinearities make an accurate position difficult to achieve, and it requires an appropriate controller for better performance.

In early 1900s, the use of this actuator was limited to a certain application due to the difficulty of obtaining a good performance.Thus, research on this component is rarely performed for decades until there is a demand to be applied in the automation industry circa 1950s [3]. Research on pneumatic positioning control has grown significantly in the 1990s when many control techniques have been examined on the system as reported in [4-6]. Although the conventional PID controller is not suitable for the systems with high nonlinearity, but it is still popular with the idea of modification as a study conducted by [7-9]. This controller is widely applied in industries compared to the other techniques due to its good characteristics, low cost and easy to implement as well as mature in theoretical analysis [10-12]. The advanced control strategies such as adaptive control, fuzzy logic control, neural network and others were aggressively investigated and applied on the early of 2000s. However, in the last decade many researchers found that the techniques that integrate with PID controller are more practical. It referred to the increasing number of publications written by [13-21] amongst others. For example, due to the drawback of adaptive controllers that are not fast enough to follow the parameter variation, [16] have proposed Multi-model controller based on several fixed PD-controllers. This technique is proposed for the position control of a pneumatic cylinder under variable loads. Five PD controllers were tuned based on the estimated model for the five fixed load that has been identified earlier. Experimental results show that this technique significantly improved the ability of producing a good performance even under variable load conditions.

This paper deals with the investigation on the robustness of the pneumatic actuators which controlled by the novel Self-regulation Nonlinear PID (SNPID) controller that had been published in the previous work [22]. The system is examined based on the variation of load and pressure in increasing and decreasing values. The DZC was added to the real system, and the consequence to the system was observed. The experiments were performed to confirm the capability of this controller. The comparisons with the other existing methods including PID and NPID controller are performed based on transient and steady-state performance. The rest of this paper is organized as follows: In section 2, research method is described starting with mathematical modelling and followed by the design of the controller. The simulated and experimental results using MATLAB/SIMULINK are described in section 3. Finally, section 4 contains some concluding remarks.

2. RESEARCH METHOD

2.1 System Model

The system under consideration is shown in Fig.1. It consists of 5/3 proportional directional control valve, double-acting with double rod cylinder, displacement transducer, pressure sensors, data acquisition system, PC and mass payload. The transfer function of the system is obtained using System Identification. For this purpose, 2000 data points representing the input and output signal of the open loop system were collected with a sampling time of 0.01 second. A state space model as shown in (1) and (2) is used as a model structure of the system.

$$x(t+Ts) = Ax(t) + Bu(t) + Ke(t) \tag{1}$$

$$y(t) = Cx(t) + Du(t) + e(t) \tag{2}$$

where, $A \in R^{nxn}$, $B \in R^{nxm}$, $C \in R^{1xn}$ and $D \in R^{1xm}$ are the matrices of the system. While $x(t) \in R^n$, $y(t) \in R$, $u(t) \in R^m$, and $K \in R^{n \times m}$ represent the state-vector, measured output, measured input signal and noise, respectively. The estimation of the values of the parameters is performed using the Prediction-Error Minimization (PEM) technique within MATLAB. Through this method, the parameters are calculated by minimizing a cost function of the prediction errors, $\varepsilon(t)$ giving;

$$V_N(\theta, Z^N) = \frac{1}{2N}\sum_{t=1}^{N} \varepsilon^2(t) \tag{3}$$

where Z^N and N denotes the set of data and number of data samples, respectively. For example, the input-output data over a time interval of $1 \le t \le N$ is represented by:

$$Z^N = [u(1), y(1), u(2), y(2), ... u(N), y(N)] \tag{4}$$

The data is used for estimating a model. The difference between the observed output and predicting output is known as the residual or prediction error which is given by

$$\varepsilon(t) = y(t) - \hat{y}(t) \tag{5}$$

where $y(t)$ and $\hat{y}(t)$ represent observed ouput and predicted output, respectively. In general, the output $y(t)$ can be represented as;

$$y(t) = q^{-n_k} G(q)u(t) + H(q)e(t) \tag{6}$$

The estimated parameter is obtained by minimizing (3) as follows;

$$\hat{\theta}_N = \hat{\theta}_N(Z^N) = \text{argmin} V_N(\theta, Z^N) \tag{7}$$

The following equation is the discrete state-space equation obtained through this identification process.

$$A = \begin{bmatrix} 2.846 & -1.350 & 8.544\times10^{-1} \\ 2 & 0 & 0 \\ 0 & 0.5 & 0 \end{bmatrix} \qquad B = \begin{bmatrix} 0.125 \\ 0 \\ 0 \end{bmatrix}$$

$$C = \begin{bmatrix} -3.494\times10^{-2} & 3.765\times10^{-2} & -3.967\times10^{-2} \end{bmatrix} \quad ; \quad D = [0]$$

The continuous transfer function can be obtained using the Zero Order Hold (ZOH) conversion method with sampling time, Ts = 0.01s. This conversion method generates the continuous time input signal by holding each sample value constant over one sample period.

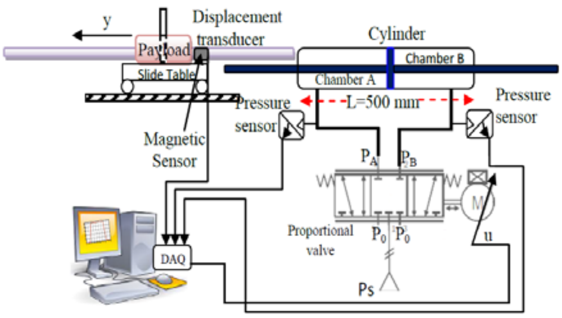

Figure 1. Experimental Setup

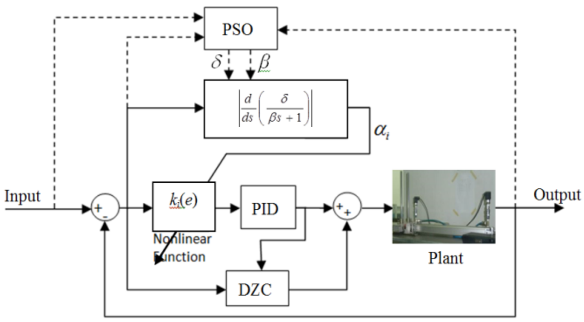

Figure 2. System with SNPID controller

a. Controller Design

In general, the transfer function of PID controller with noise filter is given by:

$$\frac{U_{PID}(s)}{E(s)} = K_p\left(1 + \frac{1}{T_i s} + \frac{T_d s}{\left(\frac{T_d}{N}s + 1\right)}\right) \tag{8}$$

where K_p is the proportional gain and T_i is the integral time. Both parameters can be tuned to improve the rise time and eliminate the steady state error, respectively. Meanwhile, the derivative time, T_d can give the effect of increasing the stability of the system by improving the transient response. In control system design, stability is the first criterion that needs to be considered. In order to maintain the stability of the system, the conditions as written in (9) must be complied.

$$|L(j\omega_{BT})| < 1 \tag{9}$$

where $|L(j\omega_{\beta T}|$ is a magnitude of the open loop system.

The speed of the response is one of the criterions that need to be considered to obtain the optimal performances. It leads to considering the bandwidth frequency of the system. In general, the speed of the response is increased with respect to the increasing of bandwidth. However, it involves a trade-off between speed and robustness of the response, and high bandwidth makes the system sensitive to the noise. Thus, in order to provide a good results in a wide range of performance including stability, speed and robustness, the design should correspond to various criteria including gain margin, phase margin, gain crossover frequency and maximum sensitivity. In order to ensure the optimum performance to be acquired, several simulations based on different Gain Margin (GM) and Phase Margin (PM) were conducted as depicted in Table 1. Based on these results, the optimum value of GM and PM are 15.7 dB and 71.9° at frequency 1.380 Hz and 0.286 Hz, respectively. These values provide an appropriate trade-off between speed performance and robustness. According to [23], in practice for well-tuned system the value of GM and PM should be between 6 dB to 20 dB and 35° to 80°, respectively. The maximum peak for the sensitivity function is less than 6dB.

Table 1. Performance of the System with respect to GM and PM

Gm	*Pm*	*tr (s)*	*ts (s)*	*Number of oscillation*	*Robustness criterion*
27.200	85.000	4.140	7.560	-	√
19.700	78.200	1.490	2.820	-	√
15.700	71.300	0.748	1.460	-	√
13.700	66.400	0.518	0.881	< 1 cycle	√
11.900	61.000	0.390	1.170	< 1 cycle	√
9.470	51.700	0.287	1.490	<2 cycle	√
2.340	13.500	0.143	5.360	6 cycle	×

2.3. Determination of Nonlinear Gain

The nonlinear gain, $k_x(e)$ which bounded in the sector $0 \le k_x(e) \le k(e_{\max})$ as indicated in (10) is used to increases the performance of the system in terms of speed. This gain represents the continuous dynamic nonlinear function. This function is then combined in cascade with PID controller.

$$k_x(e) = \frac{\exp(\alpha_i e) + \exp(-\alpha_i e)}{2} \tag{10}$$

where:

$$e = \begin{cases} e & |e| \le e_{\max} \\ e_{\max}\, sign(e) & |e| > e_{\max} \end{cases}$$

α_i and e_{max} represent the rate variation of nonlinear gain and range of variation, respectively. The value of nonlinear gain $k_x(e)$ is automatically varied depends on the value of α_i that is on-line generated using (11). Figure 2 shows the block diagram of the system with SNPID controller.

$$\frac{\alpha_i(s)}{e(s)} = \left| \frac{d}{ds}\left(\frac{\delta}{\beta s + 1} \right) \right| \tag{11}$$

In order to ensure the stability of the system, the maximum value of nonlinear gain $k(e_{max})$ should be determined in advance. This is performed via Popov stability criterion. The procedure of this criterion based on second order system has discussed by other researcher in [24]. Since the plant is represented as a third order system, the use of a MATLAB function is more practical to get the Popov plot. Figure 3 illustrate the Popov plot of the tested system. It is possible to construct a straight line with a positive slope passing through the intersect point between Popov and the real axis where the Popov plot is entirely to the right of this line. It can be seen that the Popov plot of $G(j\omega)$ is crossing the real axis at the point (-0.234, j0). The maximum value of the nonlinear gain can be obtained using (12). Therefore, the allowable range of nonlinear gain $k(e)$ is (0, 4.274).

$$k(e_{\max}) = -\frac{1}{|G(j\omega)|} \tag{12}$$

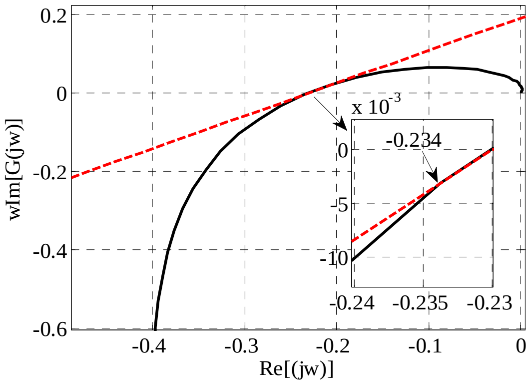

Figure 3. Popov plot

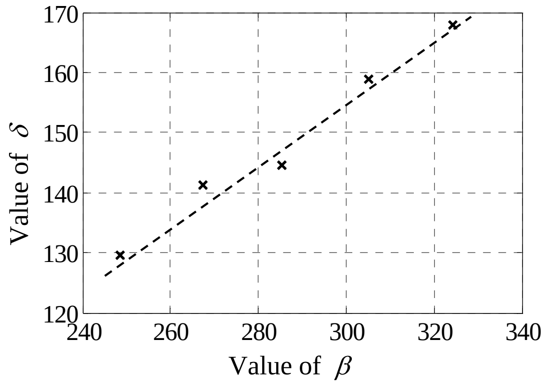

Figure 4. Relationship between δ and β

Subsequently, the design parameters are determined by identifying the relationships between δ and β in order to produce the maximum value of rate variation (α_i) with exponential decay. It performed using Particle Swarm Optimization (PSO) technique. Details on this technique were explained in [22]. Table 2 indicates the results of δ and β through this optimization technique. The relationships between δ and β can be plotted as shown in Figure 4. Thus, the equation as expressed in (13) can then be applied to determine the value of δ and β.

$$\delta = 0.519\beta \tag{13}$$

Table 2. Parameter determination via Particle Swarm Optimization

Parameter	Opt 1	Opt 2	Opt 3	Opt 4	Opt 5
δ	167.902	141.202	158.904	144.502	129.510
β	324.411	267.513	305.211	285.301	248.531
$\delta:\beta$	0.518	0.528	0.521	0.506	0.521

The rate variation (α_i) is designed to provide a certain value of nonlinear gain at the beginning for the purpose to overcome the static friction. This rate variation is then decreasing starting from this value and ending at 0 where the steady state response is achieved. For better interpretation, it can be elaborated through the following derivation. From (11), let;

$$G_{\delta,\beta}(s) = \frac{\delta/\beta}{s + 1/\beta}$$

Considered impulse response represents the error signal, thus;

$$g_{\delta,\beta}(t) = L^{-1}\left[G_{\delta,\beta}(s)\right] = 1 - \frac{\delta}{\beta}\exp^{-\frac{t}{\beta}}(t) \tag{14}$$

Perform differential of (14);

$$\alpha(t) = \frac{d}{dt}\left[g_{\delta,\beta}(t)\right] = \frac{\delta}{\beta^2}\exp^{-\frac{t}{\beta}}(t) \tag{15}$$

Based on the initial value theorem;

$$\lim_{t\to 0}\left|\alpha(t)\right| = \lim_{t\to\infty}\left|\frac{\delta}{\beta^2}\exp^{-\frac{t}{\beta}}\right| = \frac{\delta}{\beta^2} \tag{16}$$

Based on the final value theorem;

$$\lim_{t\to\infty}\left|\alpha(t)\right| = \lim_{t\to\infty}\left|\frac{\delta}{\beta^2}\exp^{-\frac{t}{\beta}}\right| = 0 \tag{17}$$

2.4. Dead-zone Compensation

In practice, the width of dead-zone is unknown. Thus, the DZC is employed to offset the deleterious effects of dead-zone. A similar compensator as in [22] is used to overcome this problem. It is implemented by using the following rules:

$$\begin{aligned}
&if\ \ |e| \le e_d\ \ then\ U_{DZC} = U_{e0}\\
&if\ \ |e| > e_d\ And\,U > 0\ \ then\ U_{DZC} = u_P\\
&if\ \ |e| > e_d\ And\,U < 0\ \ then\ U_{DZC} = u_n
\end{aligned}$$

where U_{e0} , u_p and u_n are input compensation based on error, positive dead-zone compensation and negative dead-zone compensation, respectively. Based on these conditions, there is no force imposed to the payload when the output of the DZC is represented by U_{e0}. For the other conditions in which the position error is exceeded, ed is in either positive or negative direction, the output of the controller is added to the dead zone compensator u_p and u_n, respectively.

3. RESULTS AND DISCUSSION

The performance of the pneumatic positioning system controlled by SNPID was examined using the different step input and tested to the various of load and pressure. The difference with the nominal load and pressure were tested to illustrate the robustness of this controller. The performance of this technique is compared to the other techniques namely conventional PID and NPID controller. The parameters of the proposed controller including SNF and other parameters are tabulated in Table 3. Figure 5 demonstrates the simulated result of the output response obtained from the system controlled by SNPID, NPID and PID controller. The result indicates that these controllers are able to follow the input with different position and direction. Though, it can be seen that the SNPID offer faster response with lower steady-state error compared to the other methods. The steady-state error for the system with NPID is close mimics the result obtained by the system with SNPID controller. However, it provides the slower response compared to the others. For a system with PID controller, the performance is doggerel compared to other due to the presence of overshoot that can reduce the system robustness. In order to validate the performance of the SNPID controller, the result from the simulation is compared to the result obtained from the real-time system. As can be seen from Figure 6, the response obtained based on experimental is quite similar with the simulation.

Table 3. Parameters of the controller

Control strategies	Control Parameters		
	Parameter	*Abbreviation*	*Value*
PID	Proportional Gain	K_p	2.099
	Integral Gain	T_i	104.603
	Derivative Gain	T_d	0.035
	Filter	N	12.207
SN-Function	Param 1	δ	129.510
	Param 2	β	248.531
Dead-zone compensator	Variation of Error	e_{max}	1.350
	Control value in the range of desired e_{ss}	u_{e0}	0.010
	+ve dead-zone compensation	u_p	0.500
	-ve dead-zone compensation	u_n	-0.650
	Desired e_{ss}	e_d	0.005

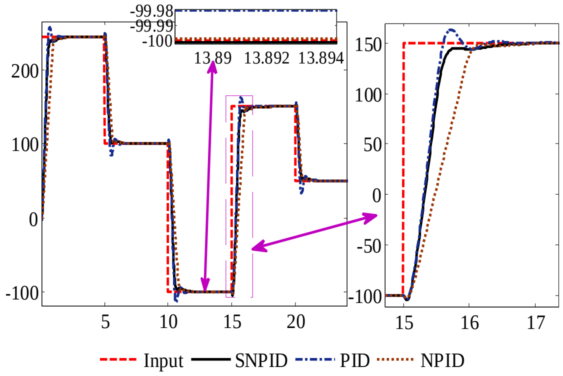

Figure 5. Simulation result for different controller

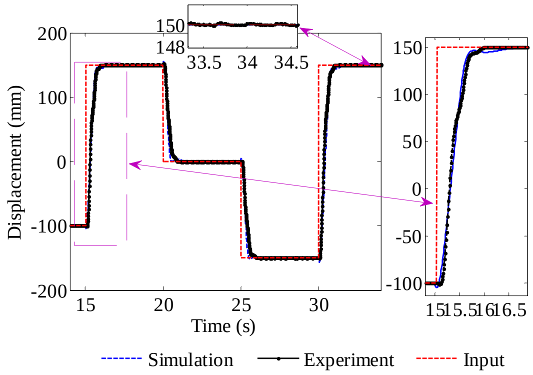

Figure 6. Validation result for SNPID

3.1. Robustness Tests

Robustness can be defined as the ability of a control system to be insensitive to the variation of the plant parameters. In order to test the system robustness, some investigations are performed to the system.The ability of the SNPID controller to compensate the system when there are changes occur in the load and pressure is examined. The performance is analyzed for both conditions in the case of the load/pressure is increasing or decreasing. The measurement of the performance is based on the distance of 200 mm. Comparison with the other methods are performed as a performance benchmark. The details of the performance based on the experiments for all cases are tabulated in Tables 4 through 7. The result indicates that the SNPID+DZC and NPID+DZC controller are more robust than PID+DZC. It can be seen that, when the moving mass is decreased from 8.4kg to 3.1 kg, the overshoot for the PID controller is significantly increased. It becomes more aggravated if the mass is increased from 8.4kg to 13.5 kg and ultimately affected the stability of the system. The same situation occurs when the pressure is decreased and increased from 0.6 MPa to 0.45 Mpa and 0.75 MPa, respectively. Meanwhile, the system with SNPID controller has succeeded to achieve better performance. The consistency of the performance for all cases indicates that this controller is less sensitive to the changes of load and pressure. The overall analysis on these finding is plotted in Figure 7. The numbers of 1, 2, 3 and 4 on the x-axes represent the experiments of table 1, table 2, table 3 and table 4, respectively.

Table 4. Performance of the system for M=3.1 kg with nominal load M=8.4 kg

Performance	Controller		
	SNPID+DZC	NPID+DZC	PID+DZC
Settling Time (t_s)	0.659	1.524	1.123
Rise Time (t_r)	0.314	1.268	0.317
Overshoot (%OS)	0.000	0.000	7.973
Steady-state error (e_{ss})	0.043	0.112	0.267

Table 5. Performance of the system for M=13.5 kg with nominal load M=8.4 kg

Performance	Controller		
	SNPID+DZC	NPID+DZC	PID+DZC
Settling Time (t_s)	0.679	1.803	1.403
Rise Time (t_r)	0.241	1.455	0.306
Overshoot (%OS)	0.000	3.375	20.869
Steady-state error (e_{ss})	0.046	0.118	0.269

Table 6. Performance of the system when Ps is reduced to 0.45 MPa

Performance	Controller		
	SNPID+DZC	NPID+DZC	PID+DZC
Settling Time (t_s)	0.797	1.612	1.115
Rise Time (t_r)	0.336	1.173	0.334
Overshoot (%OS)	0.000	0.000	8.081
Steady-state error (e_{ss})	0.0162	0.021	0.196

Table 7. Performance of the system when Ps is increased to 0.75 MPa

Performance	Controller		
	SNPID+DZC	NPID+DZC	PID+DZC
Settling Time (t_s)	0.699	1.276	1.321
Rise Time (t_r)	0.340	0.636	0.325
Overshoot (%OS)	0.000	3.304	19.992
Steady-state error (e_{ss})	0.019	0.025	0.367

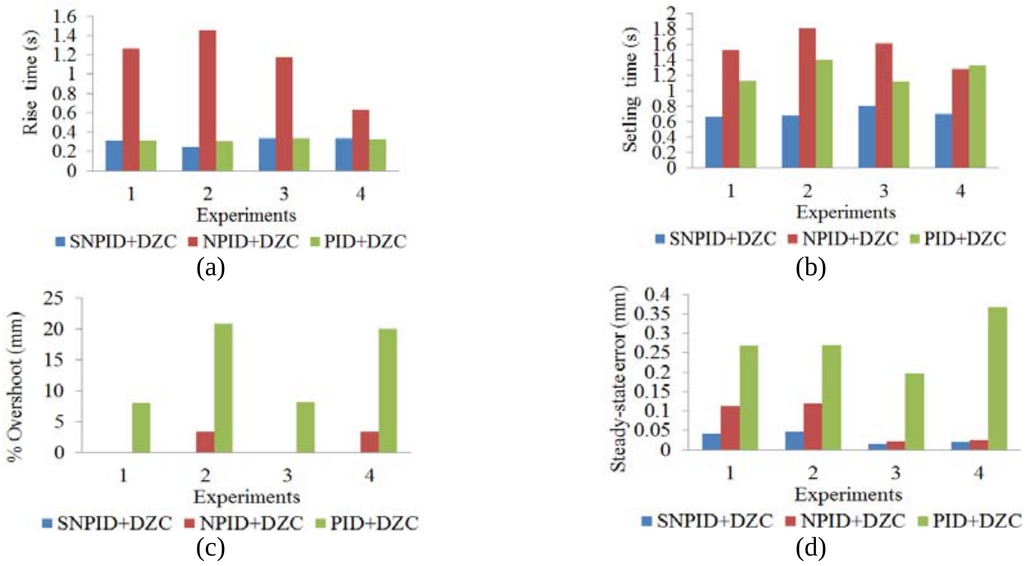

Figure 7. Robustness analysis based on decreasing and increasing of load

4. CONCLUSION

In this paper, a robustness of the SNPID controller is presented. Initially, the performances of the system with this controller are examined through simulations. Experiments to the real plant were performed for validation purposes and found only slight distinctions between them in the transient part. Subsequently, the robustness of the system was investigated chiefly by decreasing and increasing the load. Moreover, the effect caused by variation of pressures to the system performance is also examined. The system with SNPID shows a superior performance in terms of accuracy, speed and robustness compared to another method that were examined in this research. Overall, it provides a lower steady state error and is able to maintain the response without overshoot.

ACKNOWLEDGEMENTS

This research is supported by Ministry of Higher Education (MOHE) Malaysia, Universiti Teknologi Malaysia (UTM) and Universiti Teknikal Malaysia Melaka (UTeM) through Research University Grant (GUP) Tier 1 vote number Q.J130000.7123.00H36. Authors are grateful to the Ministry, UTM and UTeM for supporting the work.

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Zool H Ismail obtained his Ph.D. degree in Electrical Engineering from Heriot-Watt University, Edinburgh, United Kingdom in 2011. In 2005 and 2007, he received his B. Eng and M. Eng. degrees from Universiti Teknologi Malaysia, Skudai, Johor, Malaysia, respectively. He was appointed as a Senior Lecturer at Universiti Teknologi Malaysia in 2011 exactly after completing his Ph.D. He is a member of Society for Underwater Technology, IEEE Oceanic Engineering Society and Asian Control Association. Currently, he is involved in nonlinear control system and his main research interest is in the development of nonlinear and adaptive robust task-space control methods for regulation and trajectory tracking control of robotic systems.

NH Sunar is currently pursuing a Ph.D. degree in Electrical Engineering in Universiti Teknologi Malaysia. In 2008, she received his B. Eng degrees from Universiti Teknologi Malaysia, Skudai, Johor, Malaysia. She was appointed as a Component Design Engineer at Intel Microelectronic Malaysia in 2008 exactly after completing her B. Eng for 2 years. Currently, she is involved in adaptive control system and her main research interest is in the development of adaptive control methods for regulation and trajectory tracking control of pneumatic actuator system.

Shamsul Anuar Shamsudin received his Ph.D. in Engineering from the University of Dayton in Dayton, Ohio, USA in May 2013. His research interests include mechanisms, machine component design, engineering design methods, engineering graphics and automation. He has been with UTeM (and KUTKM) for 13 years. He is a senior lecturer at the Faculty of Mechanical Engineering in UTeM.