

A GENERATION STEP FOR FORCE REFLECTING CONTROL OF A PNEUMATIC EXCAVATOR BASED ON AUGMENTED REALITY ENVIRONMENT

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ABSTRACT: Nowadays, excavators play an important role in construction machinery. Among them, pneumatic excavators are suitable selections for some specific applications, because of their advantages. As a new trend, automation of excavation helps improving productivity, safety, and reducing of operation costs, consequently, making construction works become feasible even in hostile environment. In traditional auto-operating methods, excavators are controlled by joysticks through a so-called virtual environment in which the excavation performance is determined by some sensors attached to the real machines. However, the control accuracy by using these methodologies is reduced proportionally to the number of used sensors. The aim of this paper is to develop a mathematic model to apply to force reflecting control method based-augmented reality environment for a pneumatic excavator. Here, the mathematic model was constructed and optimized by practical excavation data. Once, the optimized model is obtained, it can be used in a combination with potential meters attached on the excavator to estimate loading forces without using any force sensor as well as to monitor current states of the excavator in the virtual environment. By using the estimated force to create a reflecting force to the joysticks, the driver can feel the real working conditions as directly-driving method. A mini-scaled pneumatic excavator was used in this study for investigating the proposed modeling method.

Keywords: *Excavator, Pneumatic Actuator, Modeling, Augmented Reality, Force Reflecting*

1. INTRODUCTION

There are hundreds of thousands of excavator machines manufactured every year and widely used in the construction, forestry, and mining industries. Among them, pneumatic excavators are suitable selections for some specific applications, because of their advantages. As a new trend, teleoperation of excavation helps improving productivity, safety, and reducing of operation costs, consequently, making construction works become feasible even in hostile environment. In the teleoperated excavator design, when there is a contact between the bucket and the environment at the remote site, a proper force signal fed back to operator is important to make operator feel as the physically present at the remote site. Therefore, there were many studies in the literature relating on force reflecting control methods [1-2]. In these studies, the force sensors

were used to measure the contacting force. However, the system cost was high and the control accuracy was reduced due to using lots of sensors.

The aim of this paper is to develop a position-error based force reflecting control method with augmented reality environment for a pneumatic excavator. In order to construct the proposed system, modeling of excavator and building its 3D model have been considered as the first two generation steps. As the first step, the excavator boom model was constructed and optimized in this paper. Dynamic models of a pneumatic actuator have been studied by some literatures [3-4]. In these researches, there was very little information on how to obtain accurate values for the model parameters [3] or the algorithm to derive them was so complicated [4]. Hence, this paper presents a simple method to estimate the model unknown

parameters by the practical data. And the second step to build the 3D excavator also was also described in this paper.

2. OVERVIEW OF PROPOSED FORCE REFLECTING CONTROL METHOD

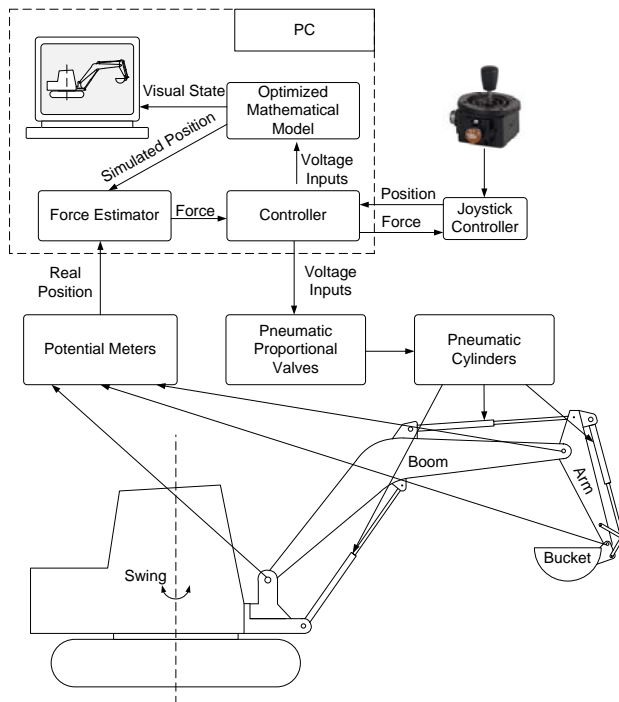


Fig.1 Schematic diagram of the proposed force reflecting control method

In this study, the mini-scaled pneumatic excavator was used to investigate the proposed force reflecting control method. This excavator has four degrees of freedom consisting of the swing, boom, arm, and bucket, which are actuated by the pneumatic cylinders. The position of each cylinder depends on the differential pressure between its two chambers, which is directly controlled by a pneumatic proportional valve. The excavator coordinate can be determined by a potential meter attached on each joint of two excavator links. Therefore, to implement the proposed control method, firstly, a mathematic model of the valves and the cylinders needs to be built and optimized. Once, the optimized model is obtained, it can be used in a combination with potential meters to estimate the loading forces without using any force sensor as well as to monitor current states of the excavator in the virtual environment. Fig. 1 depicts detailed about the proposed force reflecting

control method of a pneumatic excavator based on augmented reality environment as seen in this figure. The operator provides the trajectory commands to the real excavator by interacting with the joysticks. The joystick controller transfers these commands into proper voltage signals. These signals are acquired and fed through the A/D converter to the controller built in the computer. Here, according to the measured values, the controller creates two sets of output values. The first one includes the voltage values which are sent to the pneumatic proportional valves in order to control the pressures inside the corresponding cylinders. The pressure changes inside the cylinders then cause the excavator to move follow the operator commands. And the second one contains the voltage values are sent to the mathematic model. In no-load condition, if the mathematic model is well optimized, the simulated excavator performance yielded from the optimized mathematic model is similar with that of the real excavator. Otherwise, in the case the bucket end-point contacts with the working environment, subsequently the loading forces are attacked to the excavator links. Therefore, it exists a position difference between the mathematic model and the real excavator. Based on this different value, the force estimator creates a reflecting force to the joysticks. Thus, the operator can feel the real working conditions as directly-driving method. In addition, to make a visual feeling for the operator to control the excavator easier, the virtual excavator is also graphically rendered and viewed upon the computer screen by using the optimized mathematic model.

To prepare for the proposed force reflecting control method, the following sections describe first two generation steps of proposed control method: the procedure to build and optimize the mathematic model of excavator boom, as well as the implementation of the virtual excavator environment.

3. MATHEMATIC MODEL ANALYSIS OF THE EXCAVATOR BOOM

The mini-scaled excavator has the boom is operated by the pneumatic actuators. Therefore, an analysis of the machine dynamic behaviors is necessary. The boom actuator is then firstly investigated in this study. The boom system consists

of three components: the pneumatic proportional valve, the pneumatic cylinder, and the load. For convenience of analysis, the following nomenclatures are used:

- a, b Subscripts of inlet and outlet chambers, respectively.
- A_a, A_b The piston effective area (m^2)
- k Specific heat constant
- K_v Viscous frictional coefficient (Ns/m)
- F_s Static friction force (N)
- F_d Coulomb friction force (N)
- \dot{m} Mass flow rate (Kg/s)
- M Payload
- P_d Absolute down-stream pressure (N/m^2)
- P_u Absolute up-stream pressure (N/m^2)
- P_{atm} Atmospheric pressure (N/m^2)
- P_{cr} Critical pressure ratio
- R Universal gas constant (J/(K.kg))
- V Volume (m^3)
- L Stroke length (m)
- x Piston position (m)

The constants appearing in the mathematic model are:

$k=1.4$ $R=287 \text{ J/(K.kg)}$

$$P_{cr} = \left(\frac{2}{k+1}\right)^{\frac{k}{k-1}} = 0.582$$

3.1. Valve model

Fig. 2 displays the schematic diagram of the pneumatic proportional valve used in this paper. According to the standard orifice theory, the mass flow rate through the valve orifice can be expressed as:

$$\dot{m} = \begin{cases} A(u)P_u \sqrt{\frac{k}{RT} \frac{2}{k-1} \sqrt{\left(\frac{P_d}{P_u}\right)^{\frac{2}{k}} - \left(\frac{P_d}{P_u}\right)^{\frac{k+1}{k}}}} & , \frac{P_d}{P_u} \geq P_{cr} \\ A(u)P_u \sqrt{\frac{k}{RT} \left(\frac{2}{k+1}\right)^{\frac{k+1}{k-1}}} & , \frac{P_d}{P_u} < P_{cr} \end{cases} \quad (1)$$

where: A(u) is a equivalent valve orifice coefficient that is a function of valve supplied voltage(u).

3.2. Cylinder model

With an assumption that the thermodynamic behavior of the air inside the cylinder is an isentropic process, the following equation is applicable to each chamber of the cylinder:

$$P \left[\frac{V}{m} \right]^k = constant \quad (2)$$

where: P (N/m^2), V (m^3) and m (kg) are pressure, volume, and mass of the air inside the cylinder chamber, respectively. Differentiating Eq. (2) with respect to time, one can have:

$$m(\dot{P}V + kP\dot{V}) = k\dot{m}PV \quad (3)$$

By combining Eq. (3) with the ideal gas law $PV=mRT$, the relationship between the pressure and the mass flow rate in each of cylinder chambers can be obtained:

$$kRT\dot{m} = kP\dot{V} + V\dot{P} \quad (4)$$

where: T is temperature of the incoming air in the filling process ($\dot{m} > 0$) and temperature of the air inside the chamber in the discharging process ($\dot{m} < 0$). In practice, the difference between these temperatures is not usually significant. To simplify the model, it is assumed that $T=20^{\circ}C=273^{\circ}K$ for both the filling and discharging processes.

By applying Eq. (4) to the cylinder chambers, the changes of the cylinder pressures can be derived as:

$$\begin{cases} \dot{P}_a = \frac{kRT\dot{m}_a - kP_a\dot{V}_a}{V_a} \\ \dot{P}_b = \frac{kRT\dot{m}_b - kP_b\dot{V}_b}{V_b} \end{cases} \quad (5)$$

where: the chamber on the rodless side of the cylinder is denoted chamber A, and the other is denoted chamber B.

The chamber volumes V_a and V_b are defined as:

$$V_a = A_a x + V_{a0} \text{ and } V_b = A_b (L - x) + V_{b0} \quad (6)$$

where: V_{a0} and V_{b0} are the inactive volumes at the end of the stroke and the admission ports. Substituting (6) into (5), the following equations are obtained:

$$\begin{cases} \dot{P}_a = \frac{kRT\dot{m}_a - kP_a A_a \dot{x}}{A_a x + V_{a0}} \\ \dot{P}_b = \frac{kRT\dot{m}_b + kP_b A_b \dot{x}}{A_b (L - x) + V_{b0}} \end{cases} \quad (7)$$

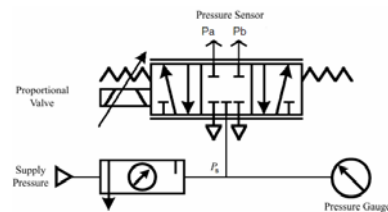


Fig. 2 Schematic diagram of the pneumatic proportional valve

3.3. Load model

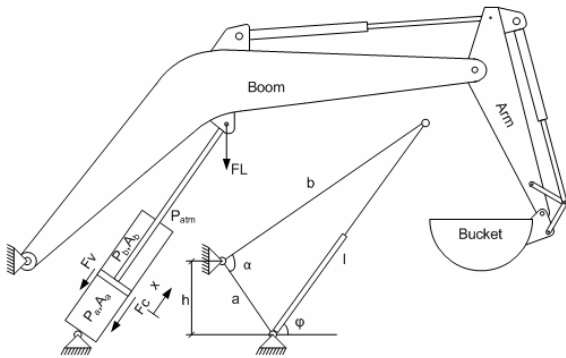


Fig. 3 Forces acting on the boom cylinder analysis

Analysis of forces acting on the boom cylinder is shown in the Fig. 3. By using the Newton's second law, the cylinder motion equation is given as:

$$P_a A_a - P_b A_b - P_{atm} (A_a - A_b) - K_v \dot{x} - F_c (P_a, P_b, \dot{x}) - F_L \sin(\varphi) = M \ddot{x} \quad (8)$$

where: \dot{x} , \ddot{x} , $F_c(\dot{x}, P_a, P_b)$ are velocity (m/s), acceleration (m/s^2), and combination of the static and Coulomb friction forces, respectively; $F_L(N)$ is an external force generated by the total mass of the excavator backhoe acting on the boom cylinder; and $\varphi(\text{rad})$ is angle between the cylinder axis and the horizontal orientation. The function $F_c(\dot{x}, P_a, P_b)$ and angle φ are described as:

$$F_c = \begin{cases} P_a A_a - P_b A_b - P_{atm} (A_a - A_b) - F_L \sin(\varphi), & \text{if } \dot{x} = 0 \\ \& |P_a A_a - P_b A_b - P_{atm} (A_a - A_b) - F_L \sin(\varphi)| \leq F_s \cos(\varphi) \\ F_d \text{sign}(\dot{x}) \cos(\varphi), & \text{if elsewhere} \end{cases} \quad (9)$$

$$\varphi = \text{Pi} - \arcsin\left(\frac{h}{a}\right) - \arccos\left(\frac{a^2 + l^2 - b^2}{2al}\right) \quad (10)$$

$$l = x + l_{\min} \quad (11)$$

where: a , b , h are in turn geometric parameters (m) of the excavator boom; l and l_{\min} are the current cylinder length (m) and the minimum cylinder length (m).

4. CYLINDER MODEL OPTIMIZATION USING PRACTICAL DATA

4.1 Test rig setup

A test rig was set up to collect the practical excavator boom data as shown in Fig. 4. The detailed specifications of the rig components are listed in Table I. The relative pressures in two chambers, termed P_{ar} and P_{br} , are measured by two Festo pressure sensors. To control the pressures P_{ar} and P_{br} ,

the pneumatic proportional valve Festo MPYE-5-1/4 was utilized. The potential meter was attached on the joint between the swing and the boom for measuring the boom angle α (rad). The pressure sensors, valve and potential meter were fed back to the computer through the PCI card 1711. The sampling time was set at 1ms for data acquisition. And the collected data in this test rig was then filtered by applying a second order low-pass filter in order to reduce the noise influence. Hence, the position of the piston x (m) in the boom cylinder can be calculated as:

$$x = \sqrt{a^2 + b^2 - 2.a.b.\cos(a)} - l_{\min} \quad (12)$$

Table I. Specifications of test rig components

Cylinder	Type	DSNU 32-60-PPVA
	Manufacturer	Festo Plc
	Length	60mm
	Bore diameter(mm)	32mm
	Rod diameter(mm)	12mm
Festo Valve	Type	MPYE-5-1/4-010B
	Input voltage range	0-10V
Position sensor	Type	10K \pm 5%
Pressure Sensors	Type	SDE5-D10-Q4-V
	Manufacturer	Festo Plc
PCI card	Type	Advantech PCI 1711

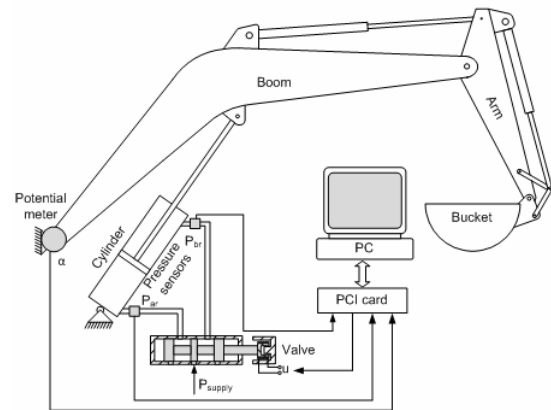


Fig. 4 Configuration of test rig for boom data acquisition

4.2 Experimental data acquisition and cylinder model validation

The method for estimating the parameters F_s , F_d , K_v , F_L , M in Eq. (8) to the boom cylinder modeling is described in

this section. These parameters can be identified by using the system responses corresponding to multi-step voltage command. The command input was a series of voltage values (u) that was applied to the proportional valve. In the current setting, the piston movements are determined as: $u > 5V$, the piston moves in the extended direction; $u < 5$, the piston moves in the opposite direction. The voltage series has three periods as displayed in the Fig. 5. At the beginning of each period, the piston was retracted to the position $x=0$ by setting $u=10V$ and waiting for the steady state condition. Next, three values of u , termed U_1 , U_2 , and U_3 , were set at the rests of three periods, respectively. These values should be large enough so that the piston acceleration could reach approximately constant values before moving to the end of the stroke. These values furthermore depend on the valve, cylinder and loading force acting on the cylinder. For the boom system in this study, the suitable values were chosen as $U_1=1V$, $U_2=2.5V$, $U_3=4V$. The period duration also should be large enough that the system could reach the stable condition before the next period started.

Consequently, experiments with the defined driving voltages were then carried out. The pressure responses and the position displacements of the boom cylinder were obtained as shown in Fig. 5. It was observed that as soon as the force acting on the piston, $P_{ar}A_a - P_{br}A_b - F_L \sin(\varphi)$, was greater than the static friction force, the piston started to move. Therefore, F_s and F_L can be estimated by the following equations:

$$F_L = \frac{F_{StartP} \cos(\varphi_{StartN}) - F_{StartN} \cos(\varphi_{StartP})}{\sin(\varphi_{StartP} + \varphi_{StartN})} \quad (13)$$

$$F_s = \frac{F_{StartP} - F_L \sin(\varphi_{StartP})}{\cos(\varphi_{StartP})}$$

$$\begin{aligned} F_{StartP} &= P_{arStartP} A_a - P_{brStartP} A_b \\ F_{StartN} &= P_{arStartN} A_a - P_{brStartN} A_b \end{aligned} \quad (14)$$

where: $P_{arStartP}$, $P_{brStartP}$, $P_{arStartN}$, $P_{brStartN}$ are the relative pressures (MPa) inside the cylinder chambers when the piston starts to move in the positive and negative direction, respectively; φ_{StartP} , φ_{StartN} are calculated by Eq. (10).

From the results in Fig. 5 and substituting the piston areas: $A_a=8.0425 \times 10^{-4} \text{ m}^2$, $A_b=1.131 \times 10^{-4} \text{ m}^2$ into (13), the static and external forces: $F_s=113.255 \text{ N}$, $F_L=261.177 \text{ N}$.

From Eq. (8), denoting that:

$$\begin{cases} A_i = P_{ar}^i A_a - P_{br}^i A_b - F_L \sin(\varphi_i) \\ B_i = \dot{x}_i \\ C_i = \cos(\varphi_i) \\ D_i = \ddot{x}_i \end{cases} \quad (15)$$

Where, index i ($i=1, 2, 3$) represents for the system response states when the piston acceleration reaches approximately the constants value corresponding to the valve voltage inputs $u=U_1$, $u=U_2$, $u=U_3$. The following equations are derived from Eq. (8):

$$A_i - B_i K_v - C_i F_d - D_i M = 0, \quad i=1,2,3 \quad (16)$$

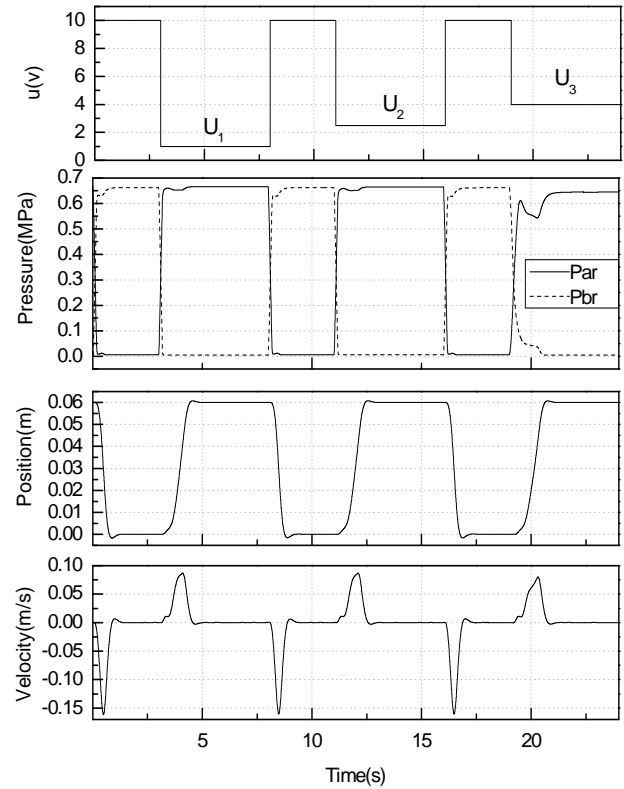


Fig. 5 System responses with the multi-steps input

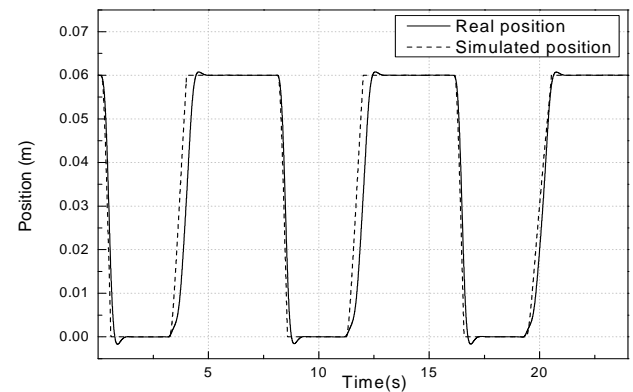


Fig. 6 Comparison of model responses and real system responses

It is clearly that Eq. (16) is the three equations system of three variables K_v , F_d , M . Consequently, it is possible to identify the parameters K_v , F_d , M from (16).

From the data in Fig. 5, Eq. (16) can be rewritten as:

$$\begin{cases} 297,2217 - 0,0784.K_v - 0,6169.F_d - 0,0517.M = 0 \\ 290,3454 - 0,0783.K_v - 0,5853.F_d - 0,0269.M = 0 \\ 289,2136 - 0,0739.K_v - 0,5846.F_d - 0,0250.M = 0 \end{cases} \quad (17)$$

By solving Eq. (17), $K_v=2.829 \cdot 10^3$ Ns/m, $F_d=85.727$ N, $M=22.7$ kg.

Next, the estimated parameters were employed into the cylinder model for validating this design model. The same driving signal was then given to the optimized model. As a result, a comparison between the actual boom response and the model output was obtained as plotted in Fig.6. It can be seen that the modeling result was similar whit the real system.

5. VIRTUAL ENVIRENMENT

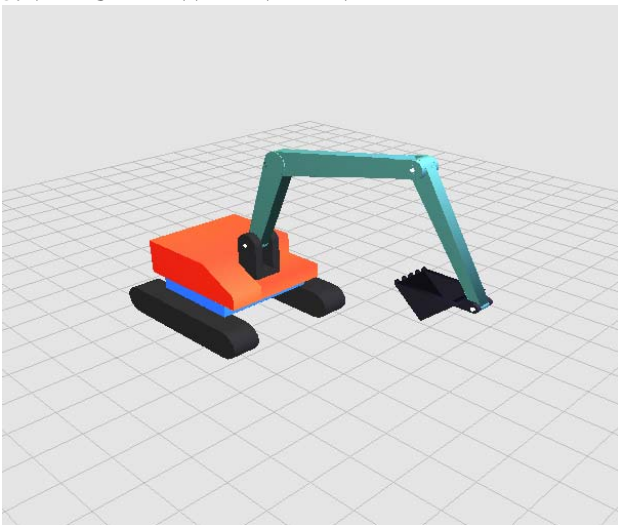


Fig. 7 The excavator in virtual environment

In order to make the visual feeling for the operator during driving the real excavator, it is necessary to construct a virtual excavator, whose manner imitates the real excavator posture. Therefore, a combination of the Visual Basic program and the graphics library built upon OpenGL was implemented to create the virtual environment as displayed in Fig. 7. By employing this virtual environment, the virtual excavator actuation is defined by the optimized excavator model and the joystick commands.

6. CONCLUSIONS

In this paper, the methodology for deriving the mathematic model of the pneumatic excavator boom was presented. And the method for estimating the model unknown parameters from the practical excavator data was also described. It was found that the optimized boom model could identify the real excavator boom in the no-load condition. This paper also presented the idea for the augmented reality development of the teleoperated excavator.

However, the method for valve model validation did not proposed in this paper. Consequently, improving this problem and constructing the whole excavator model for the force reflecting force control are the subjects for the future research.

ACKNOWLEDGEMENTS

This work was supported by Brain Korea 21(BK21)

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