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PID Control for a Pneumatic Servo System

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Abstract

This paper examines the position control ability of a pneumatic cylinder in pneumatic servo system using PID control method. A pneumatic servo system including a pneumatic cylinder and 02 proportional flow control valves is firstly proposed. The system is then modeled by dynamic equations with consideration of the valve characteristics and of friction in the pneumatic cylinder. Proportional- Integral- Derivative controller (PID) is applied to control the cylinder position. Effects of the external load and the source pressure to the control ability of the PID controller are considered. Simulation and experimental results show that the PID controller gives good control performances under different operating conditions of the external load and the air source pressure.

Keywords: Pneumatic servo system, PID control, Pneumatic cylinder

1. Introduction

Pneumatic servo systems are widely applied in many industrial applications because they are cheap, lightweight, clean, easy to assemble and create a good force/weight ratio. However, it is very difficult to achieve high-precision position control using pneumatic cylinders due to the compressive properties of the air, the nonlinearity of the servo valve or proportional valve and nonlinear friction properties existing between the contact surfaces in the pneumatic cylinders. In order to improve the position control performance of pneumatic servo systems, many control methods have been proposed. In early applications, controllers were developed by linearization of the system model around the mid-stroke position [1] or other operating points [2]. Richardson et al. used self-tuning control for a low-friction pneumatic actuator under the influence of gravity [3]. In early applications of sliding control [4–6], the controllers were designed with consideration of the piston dynamics or simplified model only. Pandian et al. [7] proposed a sliding controller based on a reduced-order non-autonomous dynamic to include the effects of the piston behavior, pressure characteristics, and valve dynamics. In their design, the friction force was assumed to be neglected to nullify the possibility of existence of mismatched uncertainties. Acarman and Hatipog lu [8] proposed a feedback-linearization control strategy with consideration of various states of the chamber pressure in the system model. In the above control methods, friction is often omitted or modeled in simple form including static friction and viscous friction.

In this paper, a pneumatic servo system composed of a double-acting cylinder and two 2-position 3-port proportional valves is considered. Its mathematical model is firstly derived in detail in consideration with a suitable dynamic friction model that has been developed for pneumatic cylinders [9]. PID controller is then examined with different desired inputs, external loads, and source pressures. Both simulation and experiment are carried out to examine the position control performance of the PID controller for the pneumatic cylinder.

2. Pneumatic serrvo system

2.1. Experiental test setup

The pneumatic servo system under consideration is shown in Figs. 1 and 2. It consists of a pneumatic cylinder fixed horizontally on a flat plate made of steel. The cylinder has internal diameter of 0.025m, rod diameter of 0.01m and piston stroke of 0.3 m, respectively. The piston end was connected to a load mass which can slide on a guiding bar. The piston motion was controlled by two flow proportional control valves. These two valves can supply a flow rate up to 720l/min with a rated voltage of 5V. If the valve inputs u_1 or u_2 varies from 2.5 to 5V, the valves will provide air into the cylinder chamber (the valves are operated in left side) and if the valve inputs u_1 or u_2 varies from 0 to 2.5V, the valves will release air into the atmosphere (the valves are operated in right side). Therefore, by combining signals between u_1 and u_2 of the two valves, the extending and retracting motions of the piston can be obtained.

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The position of the piston was measured with a position sensor. Measuring accuracy of the displacement sensor is less than 0.5% F.S. The supply

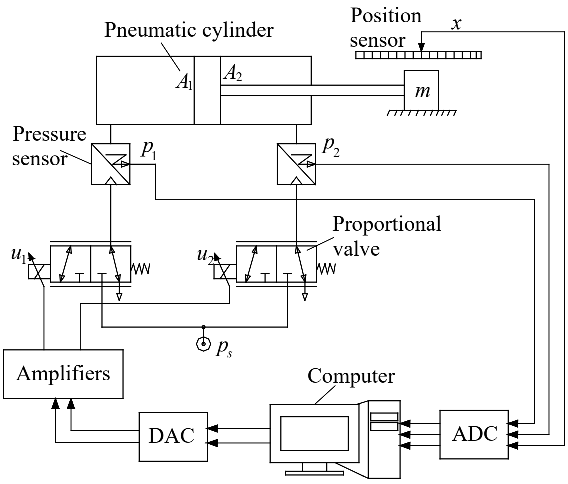

Fig. 1. Schematic of the experimental setup

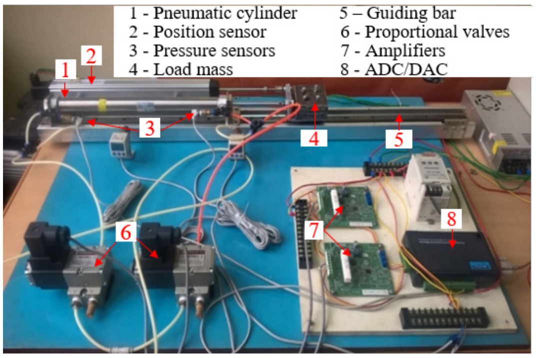

Fig. 2. Photo of the experimental test setup

pressure was set at 5bar. The position signal and the pressure signals were read via a personal computer through a 12bits Analog to Digital converter (ADC). The computer sent the control signals u_1 and u_2 of the two valves through a 12bits Digital to Analog converter (DAC). Two amplifiers were used to convert the voltage signals to the current signals of the valves. The program for data acquisition was done by using Microsoft visual C++. The position of the piston, x, was recorded at the interval of 1,1ms.

2.2. System model

This section develops mathematical equations of the system. In order to obtain the air flow dynamics in a cylinder, the following assumptions are used:

- The used air is an ideal gas and its kinetic energy is negligible in the chamber.
- The leakages of the cylinder are negligible.
- The temperature variation in cylinder chambers is negligible with respect to the supply temperature.
- The pressure and the temperature in the cylinder chambers are homogeneous.
- The evolution of the gas in each chamber is polytropic.
- The supply and exhaust pressures are constant.

As mentioned in the experimental setup in Section 2.1, if the supplied voltage to the proportional valve varies from 2.5 to 5V, the valve will provide air into the cylinder chamber (the building pressure) and if the supplied voltage varies from 0 to 2.5V, the valve will release air into the atmosphere (the exhausting pressure). In addition, it is considered that the proportional valves are overlap and thus present a dead zone in relation between the mass flow rate and the voltage signal of the valve. Therefore, the mass flow rate $\dot{m}_1$ that flows into or out from the left chamber of the pneumatic cylinder can be derived in terms of the voltage input u_1 of the left valve as follow [10]

$$\dot{m}_1 = \begin{cases} \gamma_{1b} p_s \sqrt{\dfrac{k}{RT_s}} K_{V1}\left(u_1 - 2.5\right) & if \quad u_m \le u_1 \le 5 \\ 0 & if \quad u_n < u_1 < u_m \\ \gamma_{1e} p_1 \sqrt{\dfrac{k}{RT_s}} K_{V2}\left(u_1 - 2.5\right) & if \quad 0 \le u_1 \le u_n \end{cases} \qquad (1)$$

where the operating condition $u_m \le u_1 \le 5$ corresponds to the case when the pressure in the left chamber is the building pressure, $0 \le u_1 \le u_n$ to the exhausting pressure and $u_n < u_1 < u_m$ to the case when all the valve ports are closed (dead-zone condition of the valve); u_m and u_n are respectively the right and left voltage limits of the dead-zone; p_s, p_1, and p_{atm} are respectively the source air pressure, the pressure in the left chamber of the cylinder, and the atmosphere pressure; R is the gas constant; k is the specific heat ratio; T_s is the temperature of the supply source; K_{V1} and K_{V2} are respectively the valve gains in the building pressure case and the exhausting pressure case; γ_{1b} and γ_{1e} are respectively the modifying factors when the pressure in the left chamber is building pressure and exhausting pressure as follows [10]

$$\gamma_{1b} = \begin{cases} \sqrt{\dfrac{2}{k-1}}\left(\dfrac{p_1}{p_s}\right)^{\frac{k+1}{2k}} \sqrt{\left(\dfrac{p_1}{p_s}\right)^{\frac{1-k}{k}} - 1}, & \dfrac{p_1}{p_s} \ge \left(\dfrac{2}{k+1}\right)^{k/k-1} \\ 0.58 & , \dfrac{p_1}{p_s} < \left(\dfrac{2}{k+1}\right)^{k/k-1} \end{cases} \qquad (2)$$

$$\gamma_{1e}=\begin{cases}\sqrt{\frac{2}{k-1}}\left(\frac{p_{atm}}{p_1}\right)^{\frac{k+1}{2k}}\sqrt{\left(\frac{p_{atm}}{p_1}\right)^{\frac{1-k}{k}}-1}\ , & \frac{p_{atm}}{p_1}\ge\left(\frac{2}{k+1}\right)^{k/k-1}\\ 0.58 & ,\ \frac{p_{atm}}{p_1}<\left(\frac{2}{k+1}\right)^{k/k-1}\end{cases} \tag{3}$$

Similarly, the mass flow rate $\dot{m}_2$ that flows into or out from the right chamber of the pneumatic cylinder can be derived in terms of the voltage input u_2 of the right valve by

$$\dot{m}_2=\begin{cases}\gamma_{2b}p_s\sqrt{\frac{k}{RT_s}}K_{V1}\left(u_2-2.5\right) & if\ \ u_m\le u_2\le 5\\ 0 & if\ \ u_n<u_2<u_m\\ \gamma_{2e}p_2\sqrt{\frac{k}{RT_s}}K_{V2}\left(u_2-2.5\right) & if\ \ 0\le u_2\le u_n\end{cases} \tag{4}$$

where

$$\gamma_{2b}=\begin{cases}\sqrt{\frac{2}{k-1}}\left(\frac{p_2}{p_s}\right)^{\frac{k+1}{2k}}\sqrt{\left(\frac{p_2}{p_s}\right)^{\frac{1-k}{k}}-1}\ , & \frac{p_2}{p_s}\ge\left(\frac{2}{k+1}\right)^{k/k-1}\\ 0.58 & ,\ \frac{p_2}{p_s}<\left(\frac{2}{k+1}\right)^{k/k-1}\end{cases} \tag{5}$$

$$\gamma_{2e}=\begin{cases}\sqrt{\frac{2}{k-1}}\left(\frac{p_{atm}}{p_2}\right)^{\frac{k+1}{2k}}\sqrt{\left(\frac{p_{atm}}{p_2}\right)^{\frac{1-k}{k}}-1}\ , & \frac{p_{atm}}{p_2}\ge\left(\frac{2}{k+1}\right)^{k/k-1}\\ 0.58 & ,\ \frac{p_{atm}}{p_2}<\left(\frac{2}{k+1}\right)^{k/k-1}\end{cases} \tag{6}$$

where p_2 is the pressure in the right chamber; γ_{2b} and γ_{2e} are respectively the modifying factors when the pressure in the right chamber is building pressure and exhausting pressure.

The dynamic relationship between the mass flow rates $\dot{m}_1$, $\dot{m}_2$ and the pressures p_1, p_2 in the cylinder chambers are can be given by

$$\begin{aligned}\dot{p}_1&=\frac{k}{V_1}\left(RT_s\dot{m}_1-p_1A_1v\right)\\ \dot{p}_2&=\frac{k}{V_2}\left(RT_s\dot{m}_2+p_2A_2v\right)\end{aligned} \tag{7}$$

where v is the piston velocity; V_1 and V_2 refer to the volumes of the left and right chambers of the cylinder, respectively and is calculated as

$$\begin{aligned}V_1&=V_{10}+A_1x\\ V_2&=V_{20}+A_2(L-x)\end{aligned} \tag{8}$$

where L is the piston stroke; x is the piston position; V_{10} and V_{20} are respectively the dead volumes in the cylinder chambers. Motion equation of the cylinder piston according to Newton's second law is given by

$$Ma=p_1A_1-p_2A_2-F_r \tag{9}$$

where M is the total mass of the piston, piston rod, and the load mass; a is the the piston acceleration; F_r is the friction force that is described by a dynamic friction model proposed by Tran et al [9]. The friction model is called the new modified LuGre model and is given as follows

$$F_r=\sigma_0z+\sigma_1\frac{dz}{dt}+\sigma_2(v+T\frac{dv}{dt}) \tag{10}$$

$$\frac{dz}{dt}=v-\frac{\sigma_0z}{g\left(v,h\right)}v \tag{11}$$

$$g\left(v,h\right)=F_c+\left[\left(1-h\right)F_s-F_c\right]e^{-\left(v/v_s\right)^n} \tag{12}$$

where F_r is the friction force; v is the piston velocity; z is the mean deflection of the elastic bristle between two contacting surfaces; σ_0 is the stiffness of the elastic bristle; σ_1 is the micro-viscous friction coefficient; σ_2 is the viscous friction coefficient; $g(v,h)$ is the Tribeck function; F_s is the static friction force; F_c is the Coulomb friction force; v_s is the Stribeck velocity; n is the exponent that affects the slope of the Stribeck curve; and T is the time constant for fluid friction dynamics; h is the dimensionless lubricant film thickness and is given by

$$\frac{dh}{dt}=\frac{1}{\tau_h}\left(h_{ss}-h\right) \tag{13}$$

$$\tau_h=\begin{cases}\tau_{hp} & \left(v\ne0,h\le h_{ss}\right)\\ \tau_{hn} & \left(v\ne0,h>h_{ss}\right)\\ \tau_{h0} & \left(v=0\right)\end{cases} \tag{14}$$

$$h_{ss}=\begin{cases}K_f\left|v\right|^{2/3} & \left(\left|v\right|\le\left|v_b\right|\right)\\ K_f\left|v_b\right|^{2/3} & \left(\left|v\right|>\left|v_b\right|\right)\end{cases} \tag{15}$$

$$K_f=\left(1-F_c/F_s\right)\left|v_b\right|^{-2/3} \tag{16}$$

where h_{ss} is dimensioness steay-state lubricant film thickness parameter; K_f is the proportional constant for lubricant film thickness; v_b is the velocity within which the lubricant film thickness is varied; and τ_{hp}, τ_{hn} và τ_{h0} are the time constants for acceleration, deceleration, and dwell periods, respectively. In Equation (14), $h < h_{ss}$ corresponds to the acceleration periods and $h > h_{ss}$ corresponds to the deceleration periods.

In steady-state condition, friction force is described by

$$F_{rss} = F_c + \left[\left(1 - h_{ss}\right)F_s - F_c\right]e^{-(v/v_s)^n} + \sigma_2 v \qquad (17)$$

The static parameters, F_s, F_c, v_s, v_b, n, and σ_2, of the friction model are identified from the measured steady-state friction characteristics using the least-squares method and the dynamic parameters, σ_0, σ_1, τ_h, and T, are identified from the measured dynamic friction characteristics by the methods proposed in [9].

3. PID controller design

The purpose is to design a controller so that the piston position can track well the desired position under the influence of nonlinear friction. In this study, we use a PID controller as shown in Figure 2. In the diagram, x_d is the desired control position, x is the piston position, and $e = x_d - x$ is the control error. u is the control law of the PID controller and is calculated according to the following formula:

$$u(t) = K_P e(t) + K_I \int e(t)dt + K_D \frac{de(t)}{dt} \qquad (18)$$

where, K_P, K_I, and K_D are respectively proportional, integral and derivative coefficients of the PID controllers. These coefficients are determined by trial and error method combining with tuning method [11]. Based on the flow-voltage characteristic, the valve signals u_1 and u_2 are calculated from the control law u as follows:

$$\begin{aligned} u_1 &= 2.5 + u \\ u_2 &= 2.5 - u \end{aligned} \qquad (19)$$

4. Results and disscution

In this section, simulation results are firstly presented and discussed to examine the control ability of the PID controller and to determine the appropriate controller's coefficients that are applied in the experiment. The experimental results are then given to verify the control performances of the PID controller.

4.1. Simulation results

Simulation was done by Matlab/Simulink software. The Runge-Kutta numerical solution method with a sampling time of 1.3ms was used. The parameters of the pneumatic servo system used in the simulation are shown in Table 1. The source pressure is set at 5 bar. Figure 4 shows the tracking result of the pneumatic cylinder position with the constant desired input of x_d = 0.2m. Controller's parameters K_P = 0.028, K_I = 0.00285, and K_D = 0.0098 are used. As can be seen from Fig. 4a that the piston starts from an original position of 0 m, after a time period of 0.55s, the piston reaches to the desired position of 0.2m. After that, the piston keeps at this desired position in the remaining control time. Overshoot behavior is not observed and tracking error in steady-state is $2\text{x}10^{-4}$m, an accuracy of 0.1% of the maximum moving distance (Fig. 4b). This result shows that the PID controller can give good tracking performance with a constant desired position.

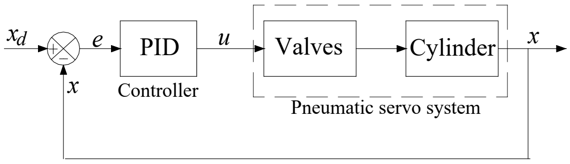

Fig. 3. Control diagram using PID controller

Table 1. System parameters

Parameter (unit)	Value	Parameter (unit)	Value
M (kg)	0.5	A_1 (m^2)	4.9x10^{-4}
p_{atm} (Pa)	0.1x10^5	A_2 (m^2)	4.12x10^{-4}
R (Nm/kgK)	287	L (m)	0.3
T (K)	295	V_{10} (m^3)	4.9x10^{-7}
k	1.3997	V_{20} (m^3)	4.12x10^{-7}
K_{V1} (m^2/V)	5x10^{-7}	u_m [V]	2.8
K_{V2} (m^2/V)	6x10^{-7}	u_n [V]	2.3

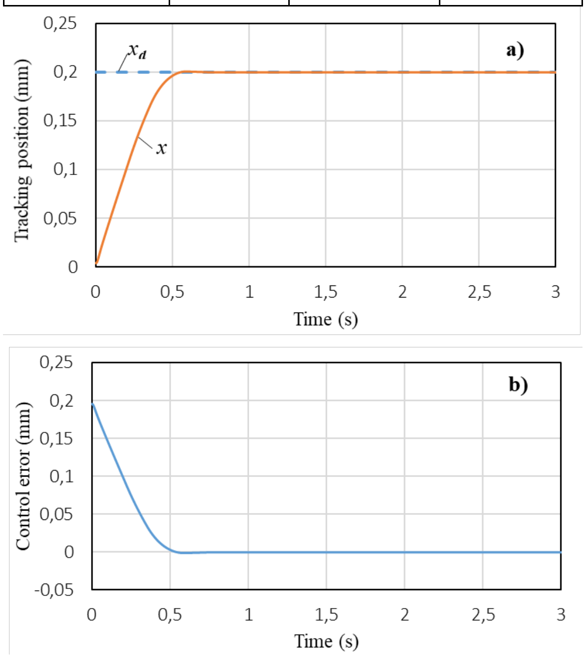

Fig. 4. Simulation results: a) tracking position, b) tracking error

4.2. Experiental results

Figure 5 shows the tracking results of the piston position obtained by experiment using the PID control method for the desired input of 0.2m. The source pressure was kept at 5 bar and the load mass at 0.5kg. The controller's parameters used in experiment were used as follows: K_P = 0.018 K_I= 0.0025 and K_D = 0.000125. These values of the parameters were adjusted from the values used in the simulation. The tracking results in Figure 5a show that the piston also can track very well the desired position, similarly to the result obtained by simulation. It takes about 0.4s for the piston to achieve the desired position of 0.2m and the steady-state control error is 0.1% as shown in Figure 5b. However, it can be noticed in the tracking results in Fig. 5a, in the first 0.15s, the piston remains at the original position of 0.02m. This result may be due to the dead-band of the valve or during the initial period, the air has not been fully compressed in the cylinder chamber to cause the piston displacement. In Figure 5c for the valve signals, the initial values of u_1 is large about 4.5 V and u_2 is about 0 V. In steady state, u_1 and u_2 keep constant near the value of 2.5 V.

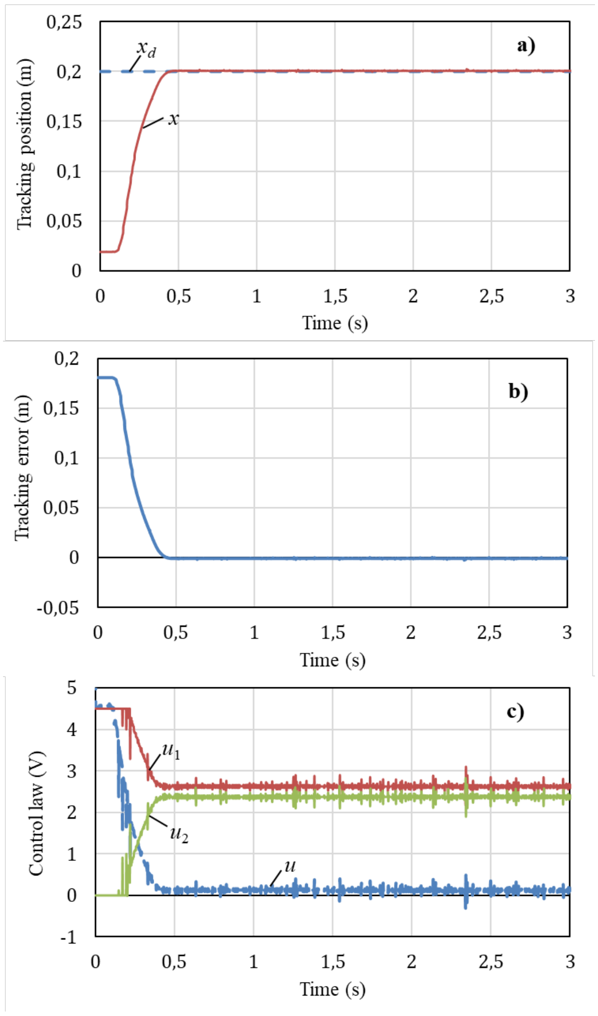

Fig. 5. Experimental result: a) tracking position, b) tracking error, c) control law (experiment)

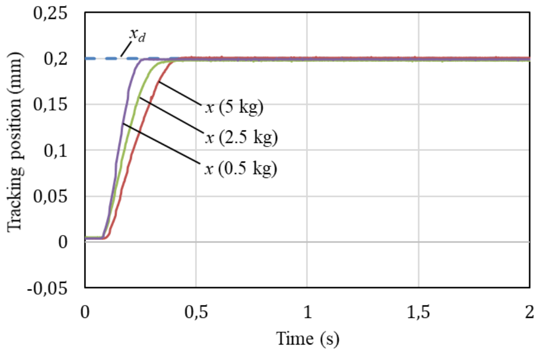

Fig. 6 Tracking results with different external loads (experiment)

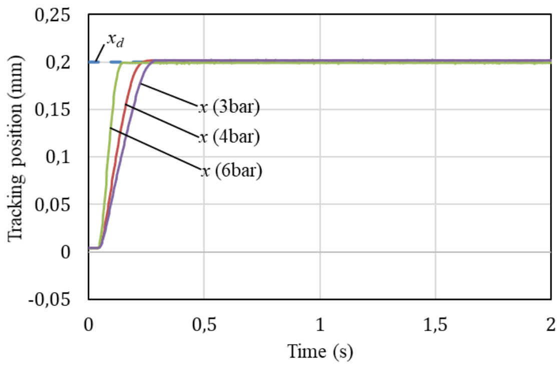

Fig. 7 Tracking results with different source pressure (experiment)

Figure 6 shows the control results under the influence of external load. Three different cases of the external loads 0.5, 2.5 and 5 kg were considered. Other system parameters and controller parameters were kept constant. The comparison results indicate that the PID controller can give good tracking results for all the three cases of the external load. However, it is shown that the rise time is increased with increasing the external load. The rise times of 0.28, 0.4 and 0.48s are respectively shown for the cases of the external load 0.5, 2.5 and 5kg.

The PID controller is also capable of giving good tracking results for cases of different source pressures as shown in Figure 7. Three cases of the source pressure 3, 4 and 6 bar were examined. When the source pressure is increased, the ability of the piston reaching to the desired value is faster. The rise times of 0.15, 0.25 and 0.3 s are respectively shown for the cases of the source pressure 6, 4 and 3 bar This can be explained by the fact that when the source pressure is greater, the flow rate fed into the cylinder chamber is more, resulting in a faster moving of the piston to the desired position.

4. Conclusion

In this paper, the study of the control capability of PID controllers for a pneumatic servo system is carried out by both simulation and experiment with step reference inputs. The results show that: i) a good tracking performance of the piston position can be obtained (the maximum rise time in transient state is less than 0.5s without overshoot and the maximum control error in steady state is less 0.1%); ii) the rise time in transient state is increased with increasing the external load and with decreasing the source pressure.

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