RESEARCH ON PNEUMATIC CYLINDER'S EXHAUSTED-AIR RECLAIMING CONTROL DEVICES

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ABSTRACT

The air with certain pressure in a pneumatic cylinder is usually exhausted into the atmosphere after work process. It is of significant that the air energy can be saved and re-used. In this paper, the constitution of an exhausted-air reclaiming system for pneumatic cylinders is studied. To find the possible influence on the cylinder work process, the effect of the system on the cylinder velocity characteristics is also researched and different control switch points are tested. Experimental results show that attaching a reclaiming device would not cause bad influence on the velocity stability if the switch point could be properly controlled. Experiments also indicate that the switch control differential pressure ΔP_{sw} varies with P_c in the receiver and the supply pressure P_s would affect the velocity stability of cylinders. Therefore to reclaim more energy and make less influence on the cylinder velocity characteristics, the suitable differential pressure ΔP_{sw} and switching-point are also tested and suggested.

KEY WORDS

Air energy save, Air exhaust reclaim device, Velocity control characteristics, Control differential pressure

NOMENCLATURE

P_s	:	Supply pressure	[MPa]
p	:	Pressure	[MPa]
d	:	Piston rod diameter	[m]
D	:	Piston diameter	[m]
A	:	Piston area	[m^2]
A_e	:	Effective area of restriction	[m^2]
V	:	Chamber volume	[m^3]
M	:	Load Mass	[Kg]
F	:	Force load	[N]
T	:	Temperature	[K]
Q_m	:	Mass flow rate	[Kg/s]
P_{c0}	:	Initial pressure in receiver	[MPa]
ΔP	:	Differential pressure	[MPa]
ΔP_{cr}	:	Critical switch differential pressure	[MPa]
ΔP_{sw}	:	Switch control differential pressure	[MPa]
v	:	Piston velocity	[m/s]
x	:	Piston displacement	[m]
ρ	:	Air density	[kg/m^3]

Subscripts

1	:	Supply side
2	:	Exhaust side
c	:	Receiver

INTRODUCTION

In working cycles of pneumatic cylinders, the air with certain pressure in the discharging chamber is usually exhausted into the atmosphere directly. If a part of the exhausted-air could be reclaimed, great energy would be saved for an industrial pneumatic system. For this, a simple and direct method is to develop a system in which the discharging chamber of a cylinder should be connected with an air receiver and the exhaust air is reclaimed into the receiver for reuse. However, there might be some potential problems for this system. On the one side, when the pressure in the receiver is

increased near the pressure in the discharging chamber, the remaining air in the discharging chamber could not be reclaimed to the receiver and it should be discharged into the atmosphere. Moreover, the pressure in the receiver would vary with the increasing or decreasing of the air mass in it. Therefore it is necessary to control the connecting of the discharging chamber with the receiver and the air discharging from the cylinder (for short, expressed as exhaust switch). On the other side, the cylinder velocity characteristics might be changed when the cylinder circuit attached with an additional air-reclaiming device. Therefore, in order to reclaim more energy and make less influence on the cylinder velocity characteristics, the suitable control criterion and the switching-point for controlling the reclaiming process should be carefully studied and determined. Moreover, the reclaiming system should be simple in structure and low in cost.

In following contents, the constitution of the reclaiming system is introduced and the mathematical model of the system has been studied. Through theoretical analysis and experimental research, the varying law of the cylinder velocity characteristics when with reclaiming device is discussed and the criterion of control differential pressure at the switch points is determined. Then some suggestions for constitution and control of the exhausted-air reclaiming system are given.

THEORETICAL ANALYSIS

Mathematical Model of Exhausted-air Reclaiming System
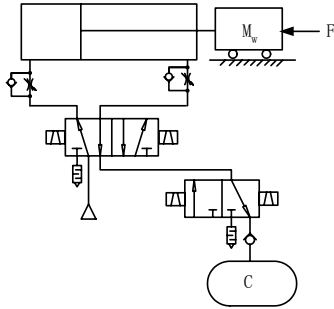

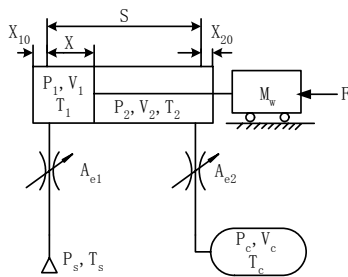Figure 1 Constitution of the exhausted-air reclaiming system

Figure 2 Working scheme of the exhausted-air reclaiming system

The exhausted-air reclaiming system is shown in Figure 1. To simplify calculation, some assumptions for the cylinder exhaust process should be made as follows [1]: (a) The thermodynamic process is adiabatic. (b) The supply pressure is constant and the temperature of supply air is equal to surrounding temperature. (c) The thermodynamic process of air in cylinder chambers is quasistatic process. (d) The internal and external leakage of the system is ignored.

The pneumatic working principle of the reclaiming system is shown in Figure 2. Applying the thermodynamics law to the charging and discharging chambers and the receiver, the mathematical model of the exhaust-air reclaiming system can be built up as follows [2][3]:

$$\frac{dp_1}{dt} = \frac{kRT_sQ_{m1}}{V_1} - \frac{kp_1}{V_1} \cdot \frac{dV_1}{dt} \qquad (1)$$

$$\frac{dp_2}{dt} = -\frac{kRT_2Q_{m2}}{V_2} - \frac{kp_2}{V_2} \cdot \frac{dV_2}{dt} \qquad (2)$$

$$\frac{dp_c}{dt} = \frac{kRT_cQ_{m2}}{V_c} \qquad (3)$$

$$\frac{d^2x}{dt^2} = \frac{1}{M_w}\left[p_1A_1 - p_2A_2 - F\right] \qquad (4)$$

Where P_1 and P_2 are the absolute pressure in the charging and discharging chambers respectively, P_c is the absolute pressure in the receiver, A_1 and A_2 are the piston effective area respectively, R is the ideal gas constant, M_w is the mass load, F is the force load and Q_m is the simplified mass flow rate and is given in literature[4].

$$Q_m = \frac{A_eP_u}{\sqrt{RT_u}}\psi(\sigma) \qquad (5)$$

$$\psi(\sigma) = \begin{cases} \dfrac{\sqrt{2}}{2}, \sigma = p_d/p_u \le 0.528 \\ \sqrt{2\sigma(1-\sigma)}, \ 0.528 < \sigma \le 1 \end{cases} \qquad (6)$$

Where A_e is the effective flow area, P_u and P_d are the upstream and downstream pressure respectively, T_u is the upstream temperature and σ is the pressure ratio.

Theoretical Analysis of the Critical Switch Differential Pressure ΔP_{cr} in Exhausted-air Reclaiming

As shown in Figure 1, after several working cycles in piston's to-and-pro movements, the pressure in the receiver would be increased and the differential pressure ΔP between the discharging chamber and the receiver would be also reduced. Thus the piston velocity will be decreased gradually. Generally the working velocity range of a standard cylinder is 50 to 500mm/s. If the piston velocity is smaller than 50mm/s, the piston might

stop or cause a "stick-slip" phenomenon due to the influence of the friction force and the compressibility of air.

Therefore to avoid the stick-slip motion, the discharging chamber of a cylinder should be opened to the atmosphere at right time before stick-slip motion after a period of exhausted-air reclaiming process. The time point when the discharging chamber is switched from exhausted-air reclaiming process to directly discharging into the atmosphere is called exhaust switch point. The differential pressure between the discharging chamber and the receiver at the exhaust switch point is called critical switch differential pressure, expressed as ΔP_{cr}. If the differential pressure between P_2 in the discharging chamber and the P_c in receiver reaches the critical switch differential pressure ΔP_{cr} in reclaiming process, the directional valve should be switched to the process of the directly discharging into the atmosphere. At this moment, the pressure P_1 in the charging chamber is close to the supply pressure P_s (that is, $P_1 \approx P_s$) and the piston motion is probably at stable state. From Eq. (4), it can be seen that to sufficiently overcome the possible force load F the maximum permissible pressure in the discharging chamber should be

$$P_{2\max} = \frac{P_s A_1 - F}{A_2} \qquad (7)$$

Assume that the thermodynamic process in chambers is isothermal process. The flow function $\psi(\sigma)$ adopts the subsonic phase in Eq. (6) when the differential pressure ΔP reaches the critical switch differential pressure ΔP_{cr} in the exhausted-air reclaiming process. Thus the mass flow through the effective flow area of the discharging pipes can be expressed as

$$Q_{m2} = \frac{\sqrt{2} A_{e2} p_2}{\sqrt{RT_s}} \sqrt{\sigma(1-\sigma)} \qquad (8)$$

Assuming $\Delta p = p_2 - p_c$:

$$\sigma = \frac{p_c}{p_2} = \frac{(p_c - p_2) + p_2}{p_2} = 1 - \frac{\Delta p}{p_2} \qquad (9)$$

Substituting Eq. (9) into Eq. (8), we get

$$Q_{m2} = \frac{\sqrt{2} A_{e2} p_2}{\sqrt{RT_s}} \sqrt{\frac{\Delta p}{p_2} - \frac{\Delta p^2}{p_2^2}} \qquad (10)$$

Because the mass flow through the discharging chamber can also be expressed as

$$Q_{m2}{}' = \rho A_2 \frac{dx}{dt} = \rho A_2 v \qquad (11)$$

Where, ρ is the air density in discharging chamber, x is the displacement of the piston and v is the piston velocity.

The ideal gas law can be written as

$$\rho = \frac{p}{RT} \qquad (12)$$

Substituting Eq. (12) into Eq. (11), the mass flow equation can be expressed as

$$Q_{m2}{}' = \frac{p_2}{RT_s} A_2 v \qquad (13)$$

From the continuity equation, we know that the mass flow out of the discharging chamber should be equal to the mass flow through the effective flow area of the exhausted-air side, which can be expressed as

$$Q_{m2} = Q_{m2}{}' \qquad (14)$$

Substituting Eq. (10) and Eq. (13) into Eq. (14), we get

$$\frac{p_2}{RT_s} A_2 v = \frac{\sqrt{2} A_{e2} p_2}{\sqrt{RT_s}} \sqrt{\frac{\Delta p}{p_2} - \frac{\Delta p^2}{p_2^2}} \qquad (15)$$

From Eq. (15), we obtain

$$\Delta p^2 - p_2 \cdot \Delta p + \frac{1}{2RT_s} \cdot \left(\frac{A_2 p_2 v}{A_{e2}}\right)^2 = 0 \qquad (16)$$

Substituting the piston velocity $v = 50 mm/s$ and the discharging chamber's pressure $P_2 \approx P_{2\max}$ into Eq. (16), the solution can be written as

$$\Delta P_{cr} = \frac{P_{2\max} - P_{2\max} \sqrt{1 - 0.0025 \cdot \frac{2}{RT_s} \cdot \left(\frac{A_2}{A_{e2}}\right)^2}}{2} \qquad (17)$$

Assuming $\xi = \frac{2}{RT_s} \cdot \left(\frac{A_2}{A_{e2}}\right)^2$ and substituting it into Eq. (17), the critical switch differential pressure ΔP_{cr} can be expressed as

$$\Delta P_{cr} = P_{2\max} \left(\frac{1 - \sqrt{1 - 0.0025 \cdot \xi}}{2}\right) \qquad (18)$$

Where, $P_{2\max}$ is the maximum permissible pressure in discharging chamber being sufficient to overcome the force load, which is defined by Eq. (7).

The critical switch differential pressure ΔP_{cr}, as expressed in Eq. (18), is considered as the theoretical value of the minimum critical switch differential pressure. The practical switch differential pressure should be determined considering other real factors.
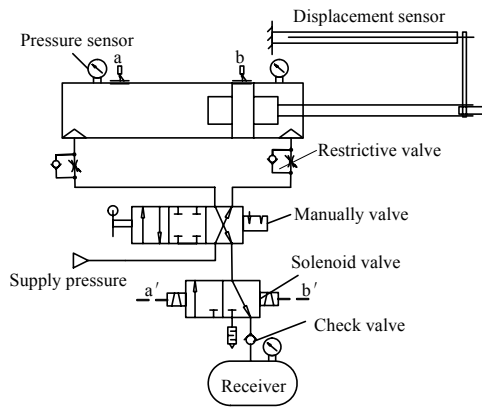
EXPERIMENTS ON EXHAUSTED AIR RECLAIMING SYSTEM

Experimental Equipment and Method

Figure 3 Experimental apparatus
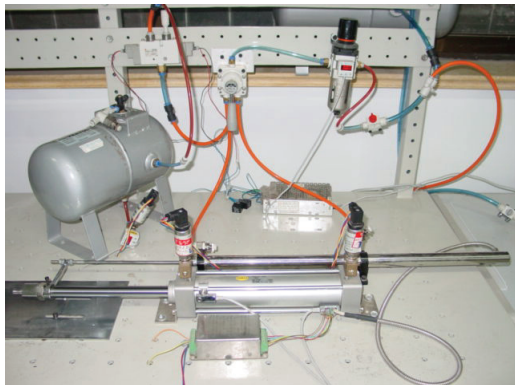
Figure 4 Photo of experimental device

The experimental apparatus is shown in Figure 3, which is used for validating the theoretical analysis and finding the real cylinder velocity characteristics when with the air reclaiming device. Figure 4 is a photo of the experimental device. Considering the similar process and working law with the extend stroke and the retract stroke of the cylinder, only the experiments for the retract stroke of the cylinder are reported and discussed.

When the piston is at the position shown in Figure 3, the solenoid valve has been switched to the right side. Along with the piston's return, the air in discharging chamber is being exhausted to the receiver. When the piston reaches the position of magnetic switch "a", the solenoid valve switches to the left side, the air in the piston rod chamber is exhausted into the atmosphere after the manually valve switches to the other side.

The diameter of the cylinder piston is *50mm* and the stroke of the cylinder is *200mm*, the volume of the receiver is 5 liter. In following sections to simplify the measuring in experiments, relative pressure is used if not demonstrative.

Influence on Cylinder's Dynamic Characteristics with Exhausted-air Reclaiming Device
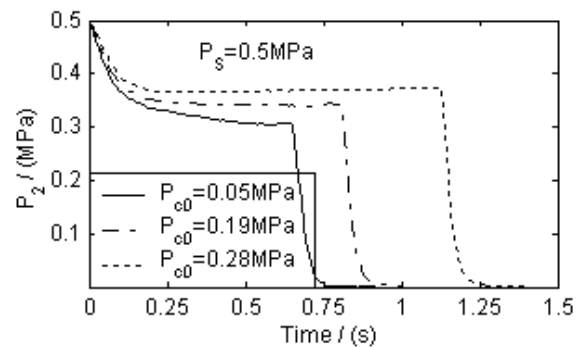

Figure 5 Variation of discharging chamber's pressure with different initial pressure in receiver

Figure 5 shows the variation of pressure in discharging chamber when the initial pressure P_{c0} in the receiver is 0.05MPa, 0.19MPa and 0.28MPa respectively in the process of exhausted-air reclaiming. From Figure 5, it can be seen that the initial pressure P_{c0} would affect the pressure curve of P_2 in discharging chamber.
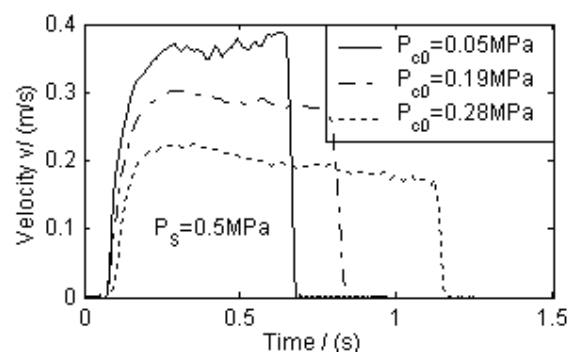

Figure 6 Variation of piston velocity with different initial pressure in receiver

Figure 6 shows the variation of piston velocity when the initial pressure in the receiver is 0.05MPa, 0.19MPa and 0.28MPa respectively in the process of exhausted-air reclaiming. From Figure 6, it can be seen that the higher

the initial pressure P_{c0} is, the lower the piston velocity and the longer the piston motion time interval. As shown in Figure 6, the higher the initial pressure P_{c0}, the smaller the velocity difference between the maximum and minimum piston velocity.   Furthermore, it could be deduced that attaching a reclaiming device would not cause bad influence on the velocity stability but would increase the piston motion time under the same load condition if the switch point could be properly controlled.
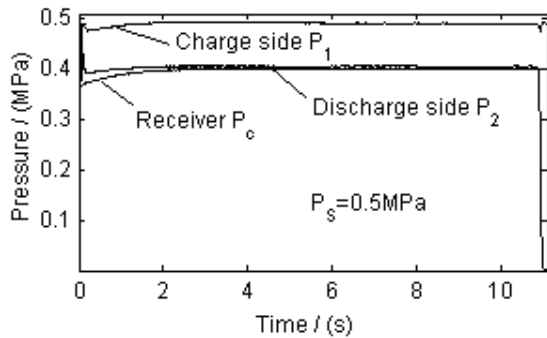
Figure 7 Variation of the pressure in cylinder chambers when P_c approaches P_2
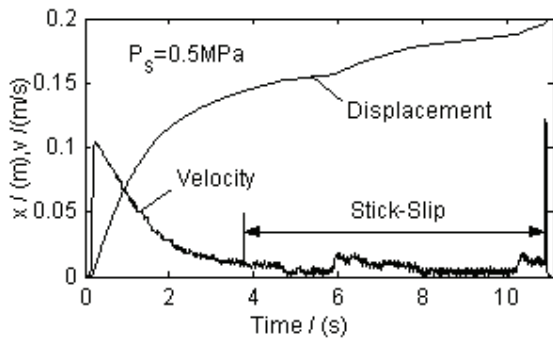

Figure 8 Variation of the piston velocity when P_c approaches P_2

The variation of the pressure in cylinder chambers and the piston velocity when the initial pressure P_{c0} in receiver approaches the stable pressure in discharging chamber are shown in Figure7 and Figure 8 respectively. Figure 7 shows the pressure condition under which a variation in motion and velocity of the piston occurs. Further analyzing the velocity curve in Figure 8 can be seen that during reclaiming when P_c approaches P_2 the stick-slip motion might take place if the cylinder discharging chamber is keeping connection with the receiver. So the exhaust from discharging directly into the atmosphere and switch off the connection between the discharging chamber and the receiver should be started before P_c equals P_2.
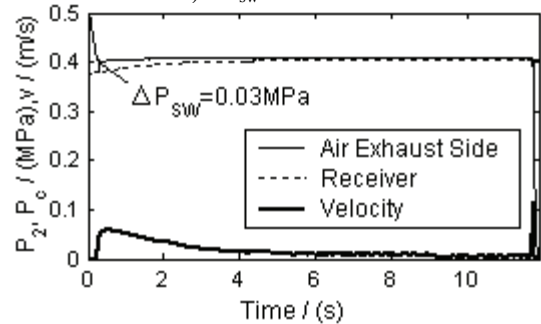
What is the more proper switch time point or the proper switch differential control pressure from reclaiming state

to the state of directly discharging into the atmosphere should be given by experiments.

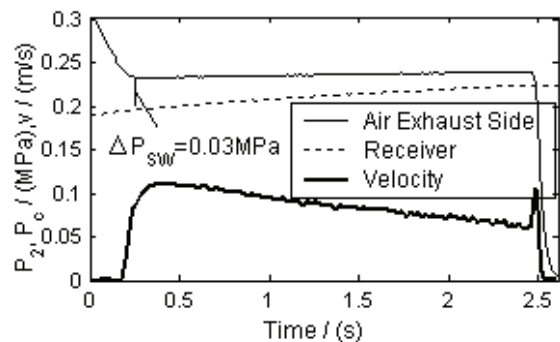**Effect of Switch Control Differential Pressure ΔP_{sw} under Different Supply Pressure**
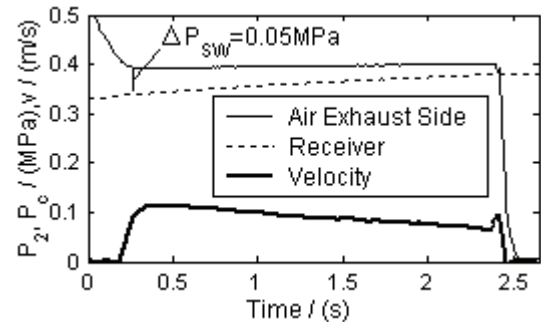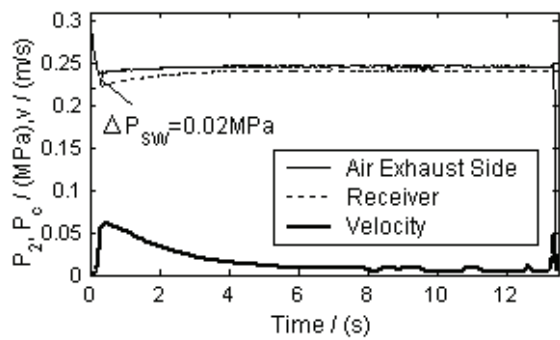

a) ΔP_{sw} =0.05MPa

b) ΔP_{sw} =0.03MPa

Figure 9 Piston velocity with different switch control differential pressure ΔP_{sw} (P_s=0.5MPa)

The piston velocity curves with different switch control differential pressure ΔP_{sw} are shown in Figure 9. When ΔP_{sw} =0.05MPa, the piston velocity is comparatively smooth and steady. However, when ΔP_{sw} =0.03MPa, the piston velocity fluctuates during the later stage and the stick-slip occurs.   In this case the pressure P_2 has approached the permissible maximum pressure P_{2max} for overcoming a certain load and the motion time is obviously prolonged.

a) ΔP_{sw} =0.03MPa

b) ΔP_{sw} =0.02MPa
Figure 10 Piston velocity with different switch control differential pressure ΔP_{sw} (P_s=0.3MPa)

The piston velocity curves with different switch control differential pressure ΔP_{sw} are shown in Figure 10. The piston velocity when ΔP_{sw} =0.03MPa is more smooth and steady that when ΔP_{sw} =0.02MPa and the motion time when ΔP_{sw} =0.02MPa is longer than that when ΔP_{sw} =0.03MPa.

The above experiments reveal that the ΔP_{sw} varies with supply pressure P_s and the pressure P_c in receiver, and the smaller the ΔP_{sw} is, the greater the possibility of stick-slip motion of the piston and the motion time are. Although use of the smaller ΔP_{sw} to control the switch point would increase the reclaimed energy, the fluctuation might happen. So selection of the real switch control differential pressure ΔP_{sw} should be considered in all its aspects.

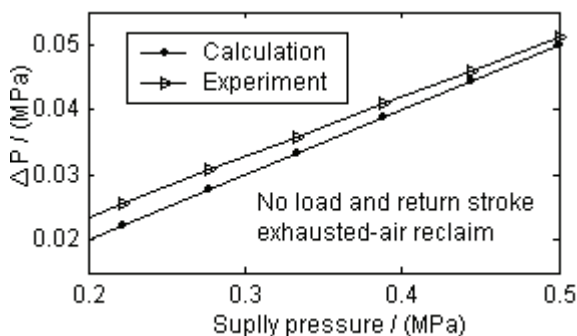**Variation of Switch Control Differential Pressure ΔP_{sw} with the Supply Pressure P_s**

Figure 11 Variation of ΔP_{sw} and ΔP_{cr} with the supply pressure P_s

Variation of ΔP_{sw} and ΔP_{cr} with the supply pressure P_s is shown in Figure 11. From Figure 11, it can be seen that the trends of both curves of the theoretical calculation and the experiment are coincident and the ΔP_{sw} as while as the ΔP_{cr} are increasing with the increase of supply pressure P_s. Considering the results shown in Figure 7 to

Figure9, we know that ΔP_{sw} also varies with P_c in receiver. The range of the ΔP_{sw} is normally within 0.02-0.05MPa. This means that with different pressure P_c and P_s the switch control differential pressure ΔP_{sw} has to be changed. This will obviously increase the complication of the control system. So for simplicity and lower cost, we intend to select a determined value for the ΔP_{sw}. For reclaiming more energy, we expect to select a smaller set value of ΔP_{sw}. But it would not satisfy the velocity requirement with a higher supply pressure P_s or higher pressure P_c in receiver. If we select a higher set value of ΔP_{sw}, the velocity requirement could be well satisfied within a wide range from lower values to higher values of P_s and P_c. So for practicality we suggest that the value of ΔP_{sw} is determined as a set value and it can be set about 0.05MPa under the given condition in our experimental reclaiming system.

CONCLUSIONS

In this paper, the constitution of an exhausted-air reclaiming system for pneumatic cylinders is studied. The effect of the system on the cylinder velocity characteristics and the control method are also researched through experiments. Experimental results show that attachment of a reclaiming device would not cause bad influence on the velocity stability but would increase the piston motion time under the same load condition if the switch point could be properly controlled, and experiments also indicate that the switch control differential pressure ΔP_{sw} varies with P_c in receiver and the supply pressure P_s of the cylinder. So the selection of ΔP_{sw} should consider some factors, such as the efficiency of reclaiming energy, simplicity and cost of reclaiming control method and the velocity stability of cylinder, etc. Therefore, we suggest that the value of ΔP_{sw} is determined as a set value and it can be set about 0.05MPa under the given condition in our experimental reclaiming system.

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