Existence scope of only primary vibration within one impact period of a hydraulic drifter piston via point transformation

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Abstract
The impact process of hydraulic drifter is a periodic vibration course with position feedback and piston control by shuttle valve. For the phenomena of secondary or multiple vibrations within one period during cyclic impact process of hydraulic drifter, this study establishes a dynamic model of the system during hydraulic impact and analyzes the vibration behavior of piston in the impact process of hydraulic drifter from the perspectives of velocity recovery coefficient $R$, acceleration ratio $K_F$, and acceleration switching time $K_t$. Then, the impact point position for the nonlinear periodic motion of the piston is observed through point transformation and Jacobian technique. Subsequently, a spatial scope defined by correlated system parameters $R$, $K_F$, and $K_t$ is identified, within this scope, the piston only stably vibrates once within one impact period. Finally, the position of the signal port is redesigned on the basis of the existence scope, and the design parameters for the hydraulic drifter are improved. The correctness of the derivation process and the conclusion of this study are experimentally verified. The present findings provide a basis for reasonably matching the relationships among design parameters and can help improve drifter impact efficiency.

Keywords: Hydraulic drifter, Dynamic model, Once vibration, Existence scope, Point transformation

1. Introduction

Hydraulic drifter act as a drilling device that use liquid as working medium to convert the pressure energy into the impact energy of the piston. They are widely applied in engineering work on mines and tunnels (Nygren, et al., 2009) (Fig. 1). The operation of the hydraulic drifter comprises propulsion (the hydraulic cylinder applies propulsion to the drifter to ensure that the drill shank is constantly in contact with the rock), impact (the hydraulic pump provides fluid pressure energy to the drifter; the impact piston reciprocally impacts the shank adapter; and the drill shank transfers the impact energy to the rock in the form of stress waves, thus causing the rock to rupture), gyration (the torque output by the hydraulic motor causes the drill bit to gyrate to a new position after each impact, and part of the cracked rock surface is removed), and flushing (the auxiliary water pump flushes the broken rock powder in the borehole) (Lin, 2014; Zou, 2017; Song, et al., 2015) (Fig. 2). Quite a few experimental and analytical studies have been conducted on impact system of the hydraulic drifter, however, the research on the vibration within one impact period of hydraulic drifter is limited.

The impact process and impact properties of the hydraulic drifter have attracted the interest of a growing number of experts and scholars in the past decades. Wasfy et al. (1997) proposed a computational procedure for simulating the contact/impact response of flexible multibody systems. The conservation of momentum and restitution equations are used as local velocity constraints to determine the “postimpact” velocities of impact nodes. Laforgia et al. (2005) simulated the working behavior of a hydraulic breaker. They realized a detailed parameterized model to simulate the physical phenomena that occurs during machine operation. Cavanough et al. (2008) derived and applied a self-optimizing control system for the percussive drilling of hard rocks. The control system superimposes an oscillating
force signal in the drill-feed force and demodulates the signal from the drill rotational torque. The resultant demodulated signal is used to control the drill. This control method is reliable and has improved drilling efficiency. Kahraman et al. (2003) analyzed system parameters, such as uniaxial compressive strength, Brazilian tensile strength, and point load strength, to calculate the penetration rate of the impactor and found that penetration rates are correlated with rock properties. Lundberg and Okrouhlik (2006) investigated the efficiency of a percussive rock drilling process by considering the wave energy radiation of the rocks and the influence of three-dimensional effects on efficiency. They found that the efficiencies of the drilling processes and elastic responses of rocks are directly correlated. Chiang et al. (2000) established impact modeling for down-the-hole rock drilling. They modeled rock–bit interaction by using a nonlinear spring and a variable gap on the basis of experimental parameter data obtained by other researchers and a normalized quasistatic penetration test described in this work. Arffman et al. (2011) established a flow field model by solving the equations that describe the time-average flow field with a commercial CFD solver. Guo et al. (2016) used the bond graph to establish a mathematical model for the nonlinear hydraulic coupling characteristics of the impact system. The experimental results prove the correctness of the simulation analysis.

Most of the scholars have focused on the concrete design parameters and detailed dynamic characteristics of hydraulic drifter. Joo-young et al. (2012) researched the hydraulic circuit of the drifter machine, analyzed the dynamic characteristics of the hydraulic impact system and the influence of rock hardness on the performance. Joo-young and Chang-heon et al. (2016) developed a back-controlled hydraulic drifter model and studied the influence of oil supply pressure, stroke regulator, acting area, and piston mass on the impact frequency and impact energy of a back-controlled drifter. Hu et al. (2014) have carried on the dynamic simulation and the experiment test to the back-controlled hydraulic drifter, so its impact performance and the influence mechanism were decided. Song et al. (2017) used the Taguchi method to effectively screen the impact performance of the hydraulic drifter design parameters, and concluded that supply pressure and rod diameter of the upper and lower piston are the main factors affecting the impact force. Other scholars (Li, et al., 2000; Wang, et al., 2015) established kinetic, system response and stress wave transfer models of the rock drill to analyze the movement law of the hydraulic drifter's impact mechanism. Moreover, they analyzed the change trend of impact performance by changing the structural parameters of rock drill. Yang et al. (2017) experimentally analyzed the dynamic characteristics of impact systems to obtain the stress wave curves of a jack rod at different impact velocities. Through the comparison between the analysis and experimental results, Seo et al. (2016) validated the reliability of the analytical model is verified, and the effect of the main factors on the impact frequency and impact energy is understood by using the verification results. Their analytical results indicated that the work area of the piston chamber is the main factor that affects impact energy and frequency. However, previous researchers did not study the piston state of secondary or multiple vibrations in the dynamic system process. If the energy of the piston is not completely exhausted after one impact period, the piston will have secondary or multiple types of vibration. At the same time, the amplitude decreases gradually, and the energy is gradually lost until the vibration stops, thus causing the piston to tremble. Trembling influences the drilling velocity of the drifter and reduces its impact efficiency.

In this paper, an experiment found that if the position of the signal port is not designed reasonably, i.e. the time of switching between the pressure in the front chamber and back chamber is inappropriate, the piston exhibits secondary or multiple vibrations. Through the use of point transformation and Jacobian technique, this study establishes a dynamic system model of drifter impact, analyzes the vibration state within the impact period of the drifter piston, and identifies the existence scope of impact point for only primary vibration within one impact period of the drifter piston in terms of recovery coefficient $R$, acceleration ratio $K_F$, and acceleration switching time $K_t$. Finally, the parameters for drifter design are optimized on the basis of on the existence scope, and the correctness of the scope is experimentally verified.
2. Irregular vibration phenomenon
2.1 Principle of drifter operation
2.1.1 Return process

The entire system is at the beginning of the return process after the last impact of the piston. At this moment, the piston is located at the front end of the drifter body, whereas the shuttle valve is on the body’s right side. The high-pressure oil $Q_1$ released by the hydraulic pump reaches the piston’s front chamber $V_1$ via the valve port $f$ and the oil duct $Q_{p1}$. This oil acts on the work face $A_1$ of the piston. Meanwhile, the low-pressure oil in the back chamber $V_2$ of the piston is connected to the return line via the oil duct $Q_{p2}$ and the valve port $h$; thus, the piston begins to accelerate the return process under the high-pressure oil in the front chamber $V_1$. Subsequently, the signal hole $c$ is opened and the oil in the left chamber $V_3$ of the valve is returned to the oil tank via the oil duct $Q_{p3}$, as shown in Fig. 3(a). The piston returns to open the signal hole $a$. The high-pressure oil in the front chamber $V_1$ of the piston is connected to the right chamber $V_4$ of the valve and pushes the shuttle valve to start moving toward the left. When the valve movement opens the valve ports $e$ and $g$ and closes the valve ports $f$ and $h$, the piston’s front chamber $V_1$ is connected to the oil tank, whereas its chamber $V_2$ is connected to the high-pressure oil $Q_1$. At this moment, the piston exhibits reverse deceleration until it stops, as shown in Fig. 3(b).
2.1.2 Stroke process

After the piston’s return process is completed, the piston is located at the back end of the drifter body, whereas the shuttle valve is on the body’s left side. The piston initiates stroke acceleration motion under the action of the high-pressure oil in the back chamber $V_2$. When the piston reaches a certain position, the signal hole $b$ opens and the right chamber $V_4$ of the shuttle valve is connected to the oil tank to prepare for the reverse process, as shown in Fig. 4(a). The piston continues its stroke acceleration motion to open the signal hole $d$. The left chamber $V_3$ of the valve is connected to the high-pressure oil in the back chamber $V_2$ of the piston, and the valve begins to reverse toward the right. The piston is accelerating at this moment, as shown in Fig. 4(b). When the valve moves toward the right to close the e and g ports and open the f and h ports, the piston’s front chamber $V_1$ is connected to the high-pressure oil, whereas its back chamber $V_2$ is connected to the oil tank. If the piston impacts the shank adapter, then it will start the next return process cycle under the action of the high-pressure oil and the shank adapter’s rebound force. Otherwise, the piston will begin to decelerate until it stops under the action of the high-pressure oil and then enters into the next return process.

Fig. 3 Return process

2.2 Trembling vibration phenomenon of piston

As shown in Fig. 5, the test bench is used to measure the acceleration, velocity, and displacement of the hydraulic drifter. The high-pressure oil released by the hydraulic pump enters the control valve, then the drifter, and finally returns to the oil tank, thereby circulating throughout the hydraulic testing system. The energy absorption device is used to absorb the impact and rotary loads of the drifter. The influence of torque on motion state is not considered in this study. It was found that if the position of the signal port is not designed reasonably, secondary or multiple vibrations exists after one impact of the piston is completed, and the piston does not return until all energy is consumed, thus causing trembling.

Hydraulic drifter impact is a periodic vibration course with position feedback and piston control by shuttle valve, as described in Section 2.1 entitled “Principle of drifter operation.” In Fig. 3(b), signal port $a$ is opened to cause the hydraulic oil to enter the back chamber of the piston via the shuttle valve, and the acceleration direction of the piston is the same as that of the stroke. In Fig. 4(b), signal port $d$ is opened to cause the hydraulic oil to enter the front chamber of the piston via the shuttle valve, and the acceleration direction of the piston is the same as that of the return. Signal ports $b$ and $c$ exert zero pressure on the left and right chambers of the valve before reversing. Thus, the position of the
signal port determines the time required for the hydraulic oil to switch between the front chamber and back chamber, thereby influencing the state of piston vibration. In turn, the state of piston vibration in the impact movement process of the drifter was investigated can serve as an actual reference for designing the signal port position and matching the pressure switching time of the front chamber and back chamber.

Signal port d is opened in advance before the piston arrives at the impact point, and the piston begins to decelerate under the damping action of pressure oil in the front chamber. If the piston arrives at the impact point and the opening of signal port d is delayed, the piston stays at the impact point and vibrates cyclically until all energy is consumed, thus causing trembling, as shown by position N in Fig. 6. The piston motion’s state curve is obtained by fitting the experimental data. The entire process from piston rebounding to vibration stopping is considered as the transitional phase of the piston from the stroke to the return motion. The piston exhibits secondary or multiple vibrations within this scope. The following section will establish the dynamic system model that was used to investigate the state of piston vibration.

![Image](image_url)

**Fig. 5** Experiment device of hydraulic drifter

**Fig. 6** Trembling of piston

### 3 Analysis of periodic vibration state

#### 3.1 Dynamic system model

The pressure and flow rate losses of hydraulic oil, which is the work medium, in the hydraulic system are assumed to be zero within the unit time when it is outputted from the shuttle valve to the working chamber of the drifter piston or the working chamber to the shuttle valve. Then, the continuous opening and closing of signal ports a and d form a periodically repeated rectangular pulse pressure in the front and back chambers of the drifter piston. This rectangular pulse pressure pushes the piston into periodic impact motion, and a periodic pulse pressure corresponds to a periodic impact motion. The piston is regarded as a particle upon which gross power \( F(t) \) acts. The dynamic system model of the hydraulic drifter impact is established (as shown in Fig. 7) and is expressed by Eq. (1):

![Image](image_url)

**Fig. 7** Dynamic system model of the hydraulic drifter impact

\[
\begin{align*}
    m\ddot{x} &= F(t), \\
    F(t) &= \begin{cases} 
    -F_+ & nT \leq t \leq (t_1 + nT) \\
    F_+ & (t_1 + nt) \leq t \leq (n + 1)T \\
    & n = 0, 1, 2, \ldots
    \end{cases}
\end{align*}
\]

(1)
where \( x \) is the coordinate of particle \( m \); \( t \) is time; \( t_1 \) is the switching time of power \( F(t) \), that is, the time signal port \( a \) is opened; and \( T \) is the vibration period, that is, the time signal port \( d \) is opened.

In the periodic movement of the piston, gross power \( F(t) \) is the resultant of viscous frictional resistance, hydraulic sticking force, sealing resistance, and axial thrust of liquid acting on the piston. New variables and parameters are introduced to utilize the minimal number of system parameters in system mode:

\[
\tau = \frac{t}{T}, \quad x = \frac{x^m}{F \cdot T}, \quad K_0 = \frac{I}{T}, \quad K_F = \frac{F}{F_i}. \tag{3}
\]

Substitute Eq. (3) into Eq. (1) to obtain:

\[
\begin{aligned}
2 \left( \tau \right) &= \tau \left( x \right) = F \left( \tau \right)
\end{aligned}
\]

\[
\begin{cases}
-2K_F & n \leq \tau \leq (K_t + n) \\
2 & (K_t + n) \leq \tau \leq (n + 1) \quad n = 0, 1, 2, \ldots
\end{cases}
\tag{4}
\]

The initial conditions of impact movement are:

\[
V_0 = -RV_c, \quad \tau = \tau_c, \quad x = x_0, \quad \dot{x} = \dot{x}_0 (\tau = 0),
\]

where \( V_c \) is the velocity at which particle \( m \) impacts the shank adapter, \( V_0 \) is the velocity after particle impact, \( \tau_c \) is impact time, and \( R \) is the velocity recovery coefficient.

On the basis of Eq. (4) and Eq. (5), the dynamic process of the impact system when the hydraulic drifter is operating can be expressed by the parameters of velocity recovery coefficient \( R \), acceleration switching time \( K_t \), and acceleration ratio \( K_F \).

### 3.2. Vibration state

Under different operating conditions of a hydraulic drifter, one impact period of a piston exhibits once, secondary or multiple vibration types. Point transformation is adopted to discuss the piston movement process and dynamically characterize the system.

Let \( x_1 = x \), \( x_2 = \dot{x} \), then \( V_0 = x_2 (\tau_c \rightarrow \tau_{c0}) \), \( V_0 = x_2 (\tau_c \rightarrow \tau_{c+0}) \).

As shown in Fig. 8(a), the function is interrupted at points \( \tau = 0, \tau = K_t \), and \( \tau = 1 \). Point \((x_1^{(0)}, x_2^{(0)})\) on plane \( \tau = 0 \) is transformed into point \((\bar{x}_1, \bar{x}_2)\) on plane \( \tau = K_t \), and the result is expressed as \( T_{\sigma} (r_c) \). Point \((\bar{x}_1, \bar{x}_2)\) on plane \( \tau = K_t \) is transformed into point \((\bar{x}_1, \bar{x}_2)\) on plane \( \tau = 1 \), and the result is expressed as \( T_{\sigma} (r_{c+0}) \). Where \( r_{c+0} \) is impact times when \( 0 < \tau \leq K_t \) and \( r_c \) is impact times when \( K_t < \tau \leq 1 \).

Thus, the vibration state of the hydraulic impact system can be expressed by Eq. (6):

\[
T_{\sigma} (r_c, r_{c+0}) \quad (i-1) < \tau \leq (i+K_t -1), \quad (i+K_t -1) < \tau \leq i,
\]

where \( N \) is the period of movement, and \( r_{c+0} \) are the impact times of the particle on the plane.

Thus, the movement state is divided into:

1. Within the scope of \( 0 < \tau \leq K_t \), the particle only vibrates once within the first negative half period of \( F(t) \) fluctuation, and the movement state can be expressed as \( T_{\sigma} (1,0) \), that is, the piston impacts the shank adapter once. This movement state is ideal, and the kinetic energy of the feedback to the system can be entirely reutilized when the piston returns because the impact energy is evenly distributed within each period.
(2) $F(t)$ produces impacts within the scope of $K_r < \tau \leq 1$ and may cause secondary or multiple impacts (Fig. 8(b)). The movement state can be expressed as $T_i (0, \tau_i) (i = 1, 2, 3, \ldots, N)$. In this state, the impact velocity of the piston is infinitely decaying until the stored energy is completely consumed, and the piston stops. Impact efficiency is influenced if the drifter cannot release kinetic energy but vibrates numerous times during impact. Thus, the existence scope of only primary vibration within the period is investigated to determine whether it can actually improve the work efficiency of the drifter.

In addition, other influencing factors, such as drilling environment, drifter hydraulic system, and drive system, are in the initial state when the drifter has just begun to operate, and the vibration type at the initial moment is complicated and irregular. However, such vibration type negligibly influences the general operating state. Thus, this condition is not discussed here.

4. Analysis of the existence scope of primary vibration within the impact period

Secondary or multiple piston vibrations can be avoided when the drifter completely releases kinetic energy after one impact. Thus, the transitional phase of shifting from stroke to return should be within $0 < \tau < K_r$.

Within $0 < \tau < K_r$, points $x_1^{(0)}, x_2^{(0)}$ on plane $t = 0$ are transformed into points $x_1, x_2$ on plane $t = 1$ to define the periodic vibration state of $T_i (1,0)$ type for the hydraulic drifter and analyze the existence scope of only primary vibration within one impact period of the piston, as shown in Fig. 8(a). Point transformation is expressed by the following function change:

$$
\bar{x}_1 = f_1(x_1^{(0)}) = \varphi_1(x_2^{(0)}),
\bar{x}_2 = f_2(x_1^{(0)}) = \varphi_2(x_2^{(0)}).
$$

Functions $f_1$ and $\varphi_1$ are solved by Eqs. (4), (5), and (7): $x_1 = (K_r - \tau_c) x_1^{(0)} + R V_c x_1^{(0)}$, $x_2 = -R V_c - 2K_f(K_r - \tau_c)$.

In Eq. (9), $V_c$ is impact velocity, and $\tau_c$ is impact time:

$$
V_c^2 = x_2^{(0)} + 4K_f x_1^{(0)} = (2K_f)^2 (x_1^{(0)} - V_c).
$$

Functions $f_2$ and $\varphi_2$ are solved with Eqs. (4), (5), and (8):

$$
\bar{x}_1 = x_1 + x_2 (1 - K_r) - (1 - K_r)^2, \quad \bar{x}_2 = x_2 + 2(1 - K_r).
$$

Eq. (9) is substituted into Eq. (11), and the resulting equation is combined with Eq. (10) to obtain transformation function $\psi_i$, as follows:

$$
\psi_i = \bar{x}_1 + R V_c (1 - \tau_c) + K_f (K_r - \tau_c)^2 + 2K_f (K_r - \tau_c)(1 - K_r)^2 - (1 - K_r)^2 = 0
$$
\[y_2 = \bar{V}_c + RV_c + 2K_c(K_r - \tau_c) - 2(1-K_r) = 0\]
\[y_3 = x_2^{(02)} - V_c^2 + 4K_c x_2^{(0)} = 0\]
\[y_4 = (2K_c)^{-1}(x_2^{(0)} - V_c) - \tau_c = 0.\]
\[\psi_i^{(i = 1, 2, 3, 4)}(x^{(0)}, x^{(0)}_0, x^{(1)}, x^{(1)}_0, \tau)\]

\[\psi_i^{(i = 1, 2, 3, 4)}(x^{(0)}, x^{(0)}_0, x^{(1)}, x^{(1)}_0, \tau)\]

When \(x^{(0)}_0 = \bar{x}_1 = \bar{x}_2\) and \(x^{(0)}_0 = \bar{x}_1 = \bar{x}_2\), solve fixed points \(\bar{x}_1^*, \bar{x}_2^*, \bar{x}_3^*, \text{ and } \bar{x}_4^*:\)
\[x_1^* = -RV_c^* - 2K_c(K_r - \tau_c^*)\]
\[x_2^* = V_c^* + 2K_c\tau_c^*\]
\[x_3^* = -0.5\tau_c^*(\bar{x}_1^* + \bar{x}_2^*).\]

The Jacobian technique is used to precisely analyze the existence scope of primary vibration within the period:
\[
\frac{A_1}{A_2} A_2 \ldots A_m \quad B_{11} \quad B_{12} \ldots B_{1m} \\
A_{21} A_{22} \ldots A_{2m} \quad B_{21} B_{22} \ldots B_{2m} \\
\vdots \quad \vdots \\
A_{n+m} A_{n+m} \ldots A_{n+m+n} B_{n+m+1} \ldots B_{n+m+m} = 0.
\]

Where
\[A_j = \left(\frac{\partial \psi_i}{\partial \tau_k}\right) \lambda + \left(\frac{\partial \psi_i}{\partial u_k}\right)^Y, \quad B_{ik} = \left(\frac{\partial \psi_i}{\partial u_k}\right), \quad i = 1, 2, \ldots, m+n; \quad j = 1, 2, \ldots, n; \quad k = 1, 2, \ldots, m;\]

where \(n\) is the number of variables; \(m\) is the number of parameters; \(u_k\) is the parameter of \(k\) when \(k = 1, u_k = V_c\); and \(k = 2, u_k = \tau_c\), which depends on transformation function \(\psi_i\).

Similarly, the displacement \(x_1\) and velocity \(x_2\) of drifter piston are variables; impact velocity \(V_c\) and impact time \(\tau_c\) are parameters; and \(n = 2, m = 2\). Function (12) \(\psi_j(i = 1, 2, 3, 4)\) is substituted into Eq. (20) to obtain determinant (21):
\[
\begin{vmatrix}
\lambda & 0 & R(1-\tau_c^*) & -RV_c^* - 2K_c(1-\tau_c^*) \\
0 & \lambda & R & -2K_c \\
4K_c & 2\tau_c^* & -2V_c^* & 0 \\
0 & (2K_c)^{-1} & -(2K_c)^{-1} & -1
\end{vmatrix} = 0.
\]

Calculate to obtain the polynomial \(N(\lambda)\) (22):
\[N(\lambda) = \lambda^2 + P(\lambda + q).\]

Where \(P = 2R + 2(1+R)K_c(V_c^*)^{-1}, q = R^2\). Then, the boundaries for the existence scope of fixed points include plane \(B_{11}\) (meeting Eq. \(1 - P + q = 0\)), plane \(B_{12}\) (meeting Eq. \(1 + P + q = 0\)), and plane \(B_{13}\) (meeting Eq. \(e^{-\varphi} + q = 0\), where \(0 < \varphi < \pi\)). The real and imaginary parts of the function are separated to obtain \(B_{13}\).
As shown in Fig. 9, the existence scope of all impact points can be expressed as follows:

\[-(1+q) \leq P \leq (1+q), \quad (23)\]

\[-1\leq q \leq 1. \quad (24)\]

Parameter \( q = R^2 \) fluctuates between 0 and 1. Thus, Eq. (24) is constantly true. The existence scope of \( T_1(1, 0) \) type vibration can be defined by Eq. (23), and Eqs. (13) and (22) are combined to expand Eq. (23):

\[-1 \leq \frac{K_F}{1-(1+K_F)K_t} \leq \left(\frac{1-R}{1+R}\right)^2. \quad (25)\]

If \( 1-(1+K_F)K_t > 0 \), then Eq. (25) is changed to:

\[-1 \leq K_F \leq \frac{(1-R^2)(1-K_t)}{(1+R)^2 + (1-R^2)K_t}. \quad (26)\]

When \( 1 < (1+K_F)K_t \), Eq. (25) is reversible. However, this finding is insignificant because \( K_F > 0 \). The condition for \( T_1(1, 0) \) periodic movement can only be \( 1 > (1+K_F)K_t \), that is:

\[K_F < \left(1-K_t\right)/K_t. \quad (27)\]

Compare the right part of Eq. (26) and right part of Eq. (27):

\[\frac{(1-R^2)(1-K_t)}{(1+R)^2 + (1-R^2)K_t} = \frac{1-K_t}{\left[(1+R)/(1-R)\right]^2 + K_t} \leq \frac{1-K_t}{K_t}, \quad (0 < R < 1). \quad (28)\]

Comparing Eqs. (18) and (26) reveals that the existence scope of \( T_1(1, 0) \) movement is restricted by the upper part of Eq. (26) and the lower part of Eq. (18):

\[-\frac{(1-K_t)[(1-K_t)-R(1+K_t)]}{K_t[2-(1+R)K_t]} \leq K_F \leq \frac{1-R^2(1-K_t)}{(1+R)^2 + (1-R^2)K_t}. \quad (29)\]

Accordingly, the scope defined by the correlated velocity recovery coefficient \( R \), acceleration switching time \( K_t \), and acceleration ratio \( K_F \) is shown in Fig. 10. The piston only vibrates once within one impact period. The figure shows that the values of \( R, K_t, \) and \( K_F \) exhibit a one-to-one correspondence (Fig. 10, Fig. 11).
Fig. 10 Existence scope of $T_1(1,0)$ type movement composed of system parameters $R, K_r$ and $K_F$

Fig. 11 Projection chart of $K_r$ cross-section for the existence scope of $T_1(1,0)$ type

If velocity recovery coefficient $R$ is fixed, then $K_r$ increases when $K_F$ decreases, because the returning acceleration of the piston decreases, thus prolonging return time. $K_F=0.34$ when $R=0$ and $K_r=0.5$.

If acceleration ratio $K_F$ is fixed, then $K_r$ increases when $R$ decreases because the coefficient of velocity recovery decreases, the conversion rate of impact energy increases, and the rebounding kinetic energy of the piston decreases. If the returning displacement is completed at a certain return acceleration, then the return time required increases.

If the switching time of acceleration $K_t$ is fixed, then $K_F$ decreases when $R$ increases because the coefficient of velocity recovery increases, the conversion rate of impact energy decreases, and the rebounding kinetic energy of the piston is large. Thus, the return acceleration decreases when the return displacement is completed within a certain return time.

Another experiment should be conducted to prove that the derived existence scope is correct and that the primary vibration is stable.

5. Verification of primary vibration stability within one impact period

The cyclic impact process of the shank adapter by the hydraulic drifter piston is the sum of complicated movement states. The positions of signal ports a and d are redesigned on the basis of the above optimization of the drifter impact mechanism. The movement period of particle is set to 1, and redesign is expressed by Eq. (30):

$$t_{r_1} + t_{r_2} + t_p = 1,$$

where $t_{r_1}$ is the time of return acceleration, $t_{r_2}$ is the time of return deceleration, and $t_p$ is the time of the
stroke, following is obtained:
\[ t_1 = \frac{1 - 2t_p}{1 - t_p}, \quad t_2 = \frac{t_p^2}{1 - t_p}. \]

The above analysis shows that the return of the piston begins to accelerate in the ending instant of stroke acceleration, causing the piston to impact only once and maximizing impact energy. The piston impacts the shank adapter, that is, the particle velocity becomes zero. Thus, \( t_r = K_r \), and stroke time is derived as \( t_r = \frac{1 - K_r}{2 - K_r} \).

Within the scope of Eq. (29), the velocity recovery coefficient \( R \) is set to solve the positions of return signal port \( a \) and stroke signal port \( d \) (Eq. (31), Eq. (32)). The dimensional parameters of the frontal and rear end faces of the piston are derived via \( K_r \).

\[
S_R = \frac{\dot{x}_0 (1 - k_1) t_p^2}{2}, \quad S_S = \frac{\dot{x}_0 (1 - k_2) t_p^2}{2}.
\]

The experiment is repeated using the basic design parameters to test the state of piston vibration during impact process, as shown in Table 1. It is found that the vibration of drifter piston is in the movement state of \( T_1(1,0) \), as shown in Fig. 12. The piston vibrates only once within one impact period. Its vibration is stable, thereby proving that the above analysis and derivation are correct.

### Table 1 Basic design parameters of drifter

| Parameters                  | Symbol | Value     |
|-----------------------------|--------|-----------|
| Mass of a piston            | \( m_p \) | 6.2 kg    |
| Supply flow rate            | \( Q \) | 120 L/min |
| Supply pressure             | \( P \) | 170 bar   |
| Area of \( A_1 \)          | \( A_1 \) | 0.00026 m² |
| Area of \( A_2 \)          | \( A_2 \) | 0.00038 m² |
| Return signal port \( a \) | \( S_R \) | 41mm      |
| Stroke signal port \( d \) | \( S_S \) | 62mm      |

Fig. 12 Primary vibration within one impact period

### 6. Discussion and conclusion

The experiment on the impact movement of the hydraulic drifter revealed that piston vibration exhibits secondary or multiple different periodic vibration types. This study solved the existence scope of only primary piston vibration within a period in the impact process of the hydraulic drifter in terms of velocity recovery coefficient \( R \), acceleration...
ratio $K_r$ and switching acceleration time $K_t$. Finally, the position of the signal port was redesigned, the design parameters of the hydraulic drifter were improved on the basis of the derived existence scope, and the correctness of derivation was verified by the results of the second test. Thus, the impact efficiency of the hydraulic drifter can be improved by optimizing the matching relation among system parameters, and an energy-saving and efficient drifter can be obtained.

This study merely investigated the condition of only primary vibration within the impact period. Several special conditions, such as compression, crushing, and rock exfoliation by the drill bit of the drifter, demands multiple stable and cyclic vibrations of the piston in the transitional phase from stroke to return process. Future research will focus on multiple stable and cyclic vibrations within the impact period.

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