THEORETICAL ANALYSIS AND DESIGN/CALCULATION FORMULAE FOR HYDRAULIC IMPACT MECHANISM[①]

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ABSTRACT The period of the piston motion of hydraulic impact mechanism was divided into three stages, and a non-dimensional parameter—the accelaration ratio β was also generalized. And then a series of formulae describing the mechanics and the kinetics of the piston were derived, which provided a new approach for designing and studying the hydraulic mechanism and new theoretical bases for further research.

Key words: hydraulic impact mechanism acceleration ratio unretrieved oil

1 INTRODUCTION

The vital working mechanism of all hydraulic drill or splitters is the impact mechanism consisting of a reciprocately moving piston. In order to improve the performance of the impact mechanism and search an accurate method for its design and analysis, it is of great importance to study the motion law of the piston. Although it is well known that the period of the piston motion can be divided approximately into three stages, i. e. the return acceleration, the return deceleration and the impact acceleration, it is impossible to describe them with a precise mathematics model on account of the complexity of the actual motion of the piston.

Up to now many mechanicians used a linear model to analyze the motion of the piston, supposing the pressed oil is supplied at a constant pressure and the piston is accelerated evenly. In addition being equal in values the return deceleration and the impact acceleration were considered to be the same. And thus some simple formulae describing the relationship among the motion parameters of the piston under an ideal working condition were derived.

In this paper, a three-staged analysis method was put forward. Considering innegligible differences exist between the acceleration values of the return and impact stages under actual working condition, and so the accelerations of the three stages need to be considered respectively. For accuracy and applicability some factors are added.

2 PARAMETERS DESCRIBING HYDRAULIC IMPACT MECHANISM

Performance parameters include impact efficiency f, impact energy E_i, total efficiency η, volume efficiency η_v and mechanism efficiency η_m. Structural parameters include mass of the piston m_p, areas of the piston's front-and rear-chamber A_f, A_r and gas-charged volume of the accumulator V_a.

Working parameters include input oil pressure P_{in}, input oil flow Q_{in}, leaking oil flow Q_l, feature speeds of the piston including the impact speed u_{im} and maximum speed during the return acceleration u_{rm}, piston stroke S_p including return acceleration stroke S_r and return deceleration stroke S_d, period of piston motion T including return acceleration time t_r, return deceleration time t_d and impact time t_i,

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piston accelerations including return acceleration a_r, return deceleration a_d and impact acceleration a_i, gas-charged pressure of the accumulator p_a, oil-charged volume V_{ri} and oil-discharged volume V_{ro} during the return acceleration, oil-charged volume V_i and oil-discharged volume V_o during the return deceleration and the impact.

There are two kinds of valve-controlled hydraulic impact mechanisms widely used around the world. One has a constant-pressurized front-chamber, and the oil pressure in the rear-chamber changes alternatively. The other has no constant-pressurized chamber, the oil pressures in both the front-chamber and the rear-chamber changeing alternatively. Hereafter these two kinds of valve-controlled hydraulic impact mechanisms are abbreviated to RC and DC respectively.

3 THE ANALYSIS OF THE PISTON KINEMATICS

During the return deceleration stage the seal resistance, the adhesive resistance, the hydraulic locking resistance, the deceleration resistance, caused by the oil pressure act in the same direction. While during the impact acceleration the seal resistance, the adhesive resistance and the hydraulic locking resistance act in the opposite direction of the oil pressure which leads the piston to accelerate. So the absolute value of the acceleration during the speed-down is larger than that during the speed-up. For convenience, two non-dimensional parameters are introduced. One is the acceleration ratio $\beta(\beta = a_r/a_i)$, and the other is $\beta_1(\beta_1 = a_1/a_d)$. In the case of the frequency f and the impact energy E_i being given, the impact velocity u_{im}, the impact oil pressure P_{in} and β being properly chosen, and the pause time after impacting being neglected, a series of formulae describing the kinematics relationship of the piston can thus be derived.

$$\psi_{SP} = \frac{\beta}{2[\beta + \sqrt{\beta(1 + \beta\beta_1)}]} \quad (1)$$

$$\psi_{ur} = \sqrt{\frac{\beta}{1 + \beta\beta_1}} \quad (2)$$

$$m_p = \frac{2E_i}{u_{im}^2} \quad (3)$$

$$T = \frac{1}{f} = t_r + t_d + t_i \quad (4)$$

$$S_p = \psi_{sp} u_{im} T \quad (5)$$

$$S_d = \beta_1 \psi_{ur}^2 S_p = \beta_1 \psi_{ur}^2 \psi_{sp} u_{im} T \quad (6)$$

$$S_r = \beta^{-1} \psi_{ur}^2 \psi_{sp} u_{im} T \quad (7)$$

$$a_i = \frac{u_{im}}{2\psi_{sp} T} \quad (8)$$

$$a_r = \beta a_i = \frac{\beta u_{im}}{2\psi_{sp} T} \quad (9)$$

$$a_d = \frac{a_i}{\beta_1} = \frac{u_{im}}{2\beta_1 \psi_{sp} T} \quad (10)$$

$$u_{rm} = \psi_{ur} u_{im} \quad (11)$$

$$t_i = 2\psi_{sp} T \quad (12)$$

$$t_d = 2\beta_1 \psi_{ur} \psi_{sp} T \quad (13)$$

$$t_r = \frac{2\psi_{ur}\psi_{sp} T}{\beta} \quad (14)$$

Where ψ_{sp} and ψ_{ur} are coefficients.

Suppose the time for control valve to change the direction of oil flow from piston impact to return and vice versa are $t_{rv}(t_{rv} = k_{rv}t_r)$ and $t_v(t_v = k_v t_i)$ respectively (where k_{rv} — the coefficient of time for control valve itself to change direction during piston return, k_v — the coefficient of time for control valve to change its direction during piston impact). Then the following formulae can be derived.

The distance from where the piston starts its return to the oil-port through which the oil flows into valve chamber for actuating the control valve to change its direction is

$$S_{rc} = \frac{a_r(t_r - t_{rv})^2}{2} = \frac{a_r(1 - k_{rv})^2 t_r^2}{2} = (1 - k_{rv})^2 S_r \quad (15)$$

The distance between where the piston starts the impact and the oil-port mentioned just now is

$$S_c = \frac{a_i(t_i - t_v)^2}{2} = \frac{a_i(1 - k_v)^2 t_i^2}{2} = (1 - k_v)^2 S_p \quad (16)$$

Formulae (3) — (6) indicate that the

working parameters related to the piston kinematics can be expressed using the given impact period/frequency, the impact speed/energy and the acceleration ratios but the input working oil-pressure.

4 THE PRESSURE-APPLIED AREAS OF THE PISTON FRONT-CHAMBER A_f AND THE REAR-CHAMBER A_r

The piston movement is affected by several kinds of resistance. However, only two are considered for practicality. One is the return oil resistance coefficient K_0. For convenience, the return oil pressure P_0 is expressed by the supplied oil pressure P_{in} as following:

$$P_0 = K_0 P_{in} \quad (K_0 = 0.06 \sim 0.12) \qquad (17)$$

The other is the comprehensive resistance coefficient K_y. K_y represents the effect of the seal resistance and the hydraulic locking resistance, which actually exists yet not clear. Since these resistances are increased with the increase of supplied oil pressure, K_y can be calculated as

$$P_y = K_y P_{in} \quad (K_y = 0.05 \sim 0.1) \qquad (18)$$

According to the conditions of pressurization, the following formulae are available:

$$\psi_{Ar} = \begin{cases} \dfrac{1 - K_y + (1 + K_y)\beta}{2\psi_{sp}[(1 - K_y)^2 - K_0(1 + K_y)]} & \text{(RC)} \\ \dfrac{1 - K_y + K_0\beta_2}{2\psi_{sp}[(1 - K_y)^2 - K_0^2]} & \text{(RC)} \end{cases} \qquad (19)$$

$$\psi_{Af} = \begin{cases} \dfrac{K_0 + (1 + K_y)\beta}{2\psi_{sp}[(1 - K_y)^2 - K_0(1 + K_y)]} & \text{(RC)} \\ \dfrac{(1 - K_y)\beta + K_0}{2\psi_{sp}[(1 - K_y)^2 - K_0^2]} & \text{(RC)} \end{cases} \qquad (20)$$

$$\beta_2 = \begin{cases} \dfrac{K_0 + (1 - K_y)\beta}{1 - K_y(1 + K_y)\beta} & \text{(RC)} \\ \dfrac{K_0(1 - K_y)\beta}{1 - K_y + K_0\beta} & \text{(RC)} \end{cases} \qquad (21)$$

$$A_r = \psi_{Ar} \frac{m_p u_{im}}{P_{in} T} \qquad (22)$$

$$A_f = \psi_{Af} \frac{m_p u_{im}}{P_{in} T} = \beta_2 A_r \qquad (23)$$

where ψ_{Ar}, ψ_{Af} and β_2 are coefficients.

$$\beta_l = \begin{cases} \dfrac{(1 - K_y)^2 - K_0(1 + K_y)}{(1 - K_y)(1 + K_y - K_0) + 4K_y\beta} & \text{(RC)} \\ \dfrac{(1 - K_y)^2 - K_0^2}{(1 + K_y)^2 + 2K_0\beta + K_0^2} & \text{(DC)} \end{cases} \qquad (24)$$

The acceleration ratio β_l is determined according to (24) once β has been determined, therefore it is not an independent parameter. In addition, we can get β according to A_f and A_r which are got formerly.

$$\beta = \begin{cases} \dfrac{(1 - K_y)A_f - K_0 A_r}{(1 - K_y)A_r - (1 + K_y)A_f} & \text{(RC)} \\ \dfrac{(1 - K_y)A_f - K_0 A_r}{(1 - K_y)A_r - K_0 A_f} & \text{(DC)} \end{cases} \qquad (25)$$

It should be pointed out that when the mechanism is operated under an ideal working condition ($K_0 = K_y = 0$, $\beta_l = 1$ and no resistance) the acceleration ratio β is equal to the area ratio of the front-chamber and the rear-chamber.

$$\beta = \begin{cases} \dfrac{A_f}{A_r - A_f} & \text{(RC)} \\ \dfrac{A_f}{A_r} & \text{(DC)} \end{cases} \qquad (26)$$

$$\beta = \begin{cases} \dfrac{\beta}{1 + \beta} & \text{(RC)} \\ \beta & \text{(DC)} \end{cases} \qquad (27)$$

5 THE INPUT OIL FLOW Q_{in}

During impact cycle the high-pressure oil is mainly consumed in the following ways.

5.1 *The Effective Oil Flow* V_e

Theoretically the effective oil flow consumed in each piston stroke can be calculated as

$$V_e = \begin{cases} (A_r - A_f)S_p & \text{(RC)} \\ A_r S_p & \text{(DC)} \end{cases} \qquad (28)$$

5.2 *The Leaked Oil Flow* V_l

Based on the oil-leakage time and leakage coefficient of the front-chamber and rear-chamber, K_{fl} *and* K_{rl} calculated using the flow formula for eccentrically circular gap in fluid mechanics, the oil loss in leakage can be calculated as formula (29).

$$V_l = \begin{cases} [K_{fl} + 2K_{rl}\psi_{sp}(1 + \beta_l\psi_{ur})]P_{in}T & \text{(RC)} \\ 2\psi_{sp}[\dfrac{K_{fl}\psi_{ur}}{\beta} + K_{rl}(1 + \beta_l\psi_{ur})]P_{in}T & \text{(DC)} \end{cases}$$
$$= \psi_l P_{in} T \qquad (29)$$

5.3 *The Unretrieved Oil Flow* V_u

During the piston return acceleration, a portion of oil($V_r = A_f S_r$) is consumed in converting into kinematic energy of the piston. And during the return deceleration this energy is used to push the high-pressure oil flow($V_d = (A_r - A_f)S_d$ for RC and $V_d = A_r S_d$ for DC) into the accumulator. Under an ideal working condition $V_r = V_d$ and all the return energy is recoverable. But $K_0 \neq 0$, $K_y \neq 0$, and thus $\beta_1 < 1$, the return decelaration stroke is shorter than that of the ideal working condition. That's to say the consumed oil flow is actually larger than that of the retrieved oil flow($V_d < V_r$). So the unretrieved high-pressure oil flow in one cycle is

$$V_u = V_r - V_d = \begin{cases} A_f S_p - A_r S_d \\ A_f S_r - A_r S_d \end{cases}$$

$$= \begin{cases} \dfrac{2\psi_{Ar}\psi_{sp}(\beta_2 - \beta_1\psi_{ur}^2)E_i}{P_{in}} & \text{(RC)} \\ \dfrac{2\psi_{Ar}\psi_{sp}\psi_{ur}^2(\dfrac{\beta_2}{\beta} - \beta_1)E_i}{P_{in}} & \text{(DC)} \end{cases} \quad (30)$$

Therefore the actual oil flow is

$$Q_{in} = \frac{V_e + V_l + V_u}{T}$$

$$= \begin{cases} \varphi_1 P_{in} + \dfrac{2\psi_{Ar}\psi_{sp}\psi_{ur}^2 E_i}{\beta P_{in} T} & \text{(RC)} \\ \varphi_1 P_{in} + \dfrac{2(1+\beta_2)\psi_{Ar}\psi_{sp}\psi_{ur}^2 E_i}{\beta P_{in} T} & \text{(DC)} \end{cases} \quad (31)$$

$\dfrac{V_e}{T}$ is called effective oil flow Q_c.

6 CHARGED AND DISCHARGED OIL FLOW OF ACCUMULATOR

The accumulator is charged and discharged oil twice an impact cycle. (See Fig. 1).

So it is of great importance to analyze the variables for the design and study of the accumulator. In Fig. 1, we can also see that the time t_{rg} or t_g, at which the needed oil flow of the front-chamber or the rear-chamber equals to the supplied oil flow (excluding the leaked oil flow) is the transitional point of the oil charging and discharging. Suppose the speeds of the piston corresponding to t_{rg} and t_g and u_{rg} and u_g, the volumes of the charged and the discharged oil can be calculated as follows (see the four shadow areas shown in Fig. 1)

$$\Delta V_{ri} = \frac{Q_e t_{rg}}{2} \begin{cases} \dfrac{2\psi_{Ar}\psi_{ur}^4\psi_{sp}^2}{\beta_2\beta^3}\dfrac{E_i}{P_{in}} & \text{(DC)} \\ \dfrac{2(1+\beta_2)^2\psi_{Ar}\psi_{ur}^4\psi_{sp}^2}{\beta_2\beta^3}\dfrac{E_i}{P_{in}} & \text{(DC)} \end{cases} \quad (32)$$

$$\Delta V_{ro} = \frac{(A_f u_{rm} - Q_e)(t_r - t_{rg})}{2}$$

$$= \begin{cases} \dfrac{2\beta_2\psi_{Ar}\psi_{ur}^2\psi_{sp}}{\beta}(1 - \dfrac{\psi_{ur}\psi_{sp}}{\beta_2\beta})^2\dfrac{E_i}{P_{in}} & \text{(RC)} \\ \dfrac{2\beta_2\psi_{Ar}\psi_{ur}^2\psi_{sp}}{\beta}[1 - \dfrac{(1+\beta_2)\psi_{ur}\psi_{sp}}{\beta_2\beta}]^2\dfrac{E_i}{P_{in}} & \text{(DC)} \end{cases} \quad (33)$$

$$\Delta V_i = \begin{cases} \dfrac{[(A_r - A_f)u_{im} + 2Q_e]t_d}{2} + \dfrac{Q_e t_g}{2} \\ \dfrac{(A_r u_{im} + 2Q_e)t_d}{2} + \dfrac{Q_e t_g}{2} \end{cases}$$

$$= \begin{cases} 2\psi_{Ar}\psi_{ur}^2\psi_{sp}[(1-\beta_2)\beta_1 + \dfrac{2\beta_1\psi_{ur}\psi_{sp}}{\beta} + \dfrac{\psi_{ur}^2\psi_{sp}}{(1-\beta_2)\beta^2}]\dfrac{E_i}{P_{in}} \\ 2\psi_{Ar}\psi_{ur}^2\psi_{sp}[\beta_1 + \dfrac{2(1+\beta_2)\beta_1\psi_{ur}\psi_{sp}}{\beta} + \dfrac{(1+\beta_2)^2\psi_{ur}^2\psi_{sp}}{\beta^2}]\dfrac{E_i}{P_{in}} & \text{(DC)} \end{cases} \quad (34)$$

$$\Delta V_0 = \begin{cases} \dfrac{[(A_r - A_f)u_{im} - Q_e](t_i - t_g)}{2} \\ \dfrac{(A_r u_{im} - Q_e)(t_i - T_g)}{2} \end{cases}$$

$$= \begin{cases} 2(1-\beta_2)\psi_{Ar}\psi_{sp}[1 - \dfrac{\psi_{ur}^2\psi_{sp}}{(1-\beta_2)\beta}]^2\dfrac{E_i}{P_{in}} & \text{(RC)} \\ 2\psi_{Ar}\psi_{sp}[1 - \dfrac{1+\beta_2)\psi_{ur}^2\psi_{sp}}{\beta}]^2\dfrac{E_i}{P_{in}} & \text{(DC)} \end{cases} \quad (35)$$

where ΔV_{ri} —charged oil flow during the piston return; ΔV_{ro} —discharged oil flow during the piston return; ΔV_i —charged oil flow during the piston impact; ΔV_0 —discharged oil flow during the piston impact

when $A_f u_{rm} \leqslant Q_e$ (36)

We can see in Fig. 1 that the accumulator doesn't undergo the discharging process. So ΔV_{ro} is the volume of the charged oil, the sum of ΔV_{ri}, ΔV_{ro} and ΔV_i is the volume of the charged oil which is equal to that of the discharged oil ΔV_0. When calculating, we may assume that $\Delta V_{ri} = \Delta V_{ro} = 0$, $\Delta V_i = \Delta V_o$.

If the design fits in with the formula (36), the diaphragm of the accumulator vibrates only once to undergo oil charge and discharge once respectively, which is very helpful to prolong the service life of the accumulator diaphragm. Formula (36) can also be formulated as

$$\begin{cases} \beta_2 \leqslant \dfrac{\psi_{ur}\psi_{sp}}{\beta} & \text{(RC)} \\ \dfrac{\beta_2}{1+\beta_2} \leqslant \dfrac{\psi_{ur}\psi_{sp}}{\beta} & \text{(DC)} \end{cases} \tag{37}$$

Actually these are the formulae describing the relationship of β, K_0 and K_Y. The value of β can be worked out once the formulae above are converted into a four-dimensional equation of β (or it is better to use calculation-trying method on computer). When β is smaller than the corresponding values listed in Table 1 the design can prolong the service life of the diaphragm. (the numbers in the brackets are for DC).

However, it should be pointed out that the maximum value of β is 1/3 under an ideal working condition($K_0 = K_Y = 0$)

7 THE RELATIONSHIP BETWEEN THE INPUT OIL FLOW AND SOME OTHER PARAMETERS

If the input oil flow Q_{in} of a given hydraulic impact mechanism is varied, it's working oil pressure P_{in}, impact frequency f, impact period T and the impact energy E_i are varied too. Their analytical relationships are shown in formulae (38)—(40).

$$u_{im}^2 = \frac{A_f S_p P_{in}}{\psi_{Ar}\psi_{sp} m_p} \tag{38}$$

$$E_i = \frac{A_r S_p P_{in}}{2\psi_{Ar}\psi_{sp}} \tag{39}$$

$$T = \sqrt{\frac{\psi_{Ar} S_p m_p}{\psi_{sp} A_r P_{in}}} \tag{40}$$

Table 1 Values of K_0, K_Y and β

K_0	0.00	0.02	0.04	0.06	0.08	0.10	0.12
K_Y	0.00	0.02	0.04	0.06	0.08	0.10	0.12
β	0.33	0.32	0.31	0.30	0.28	0.27	0.25
	(0.33)	(0.33)	(0.32)	(0.31)	(0.30)	(0.29)	(0.28)

The following formulae are derived according to formulae (38)—(40).

$$Q_{in} = \varphi_l P_{in} + \varphi_p \sqrt{P_{in}} \tag{41}$$

$$\varphi_l^2 P_{in}^2 - (\varphi_p^2 + 2\varphi_l Q_{in})P_{in} + Q_{in}^2 = 0 \tag{42}$$

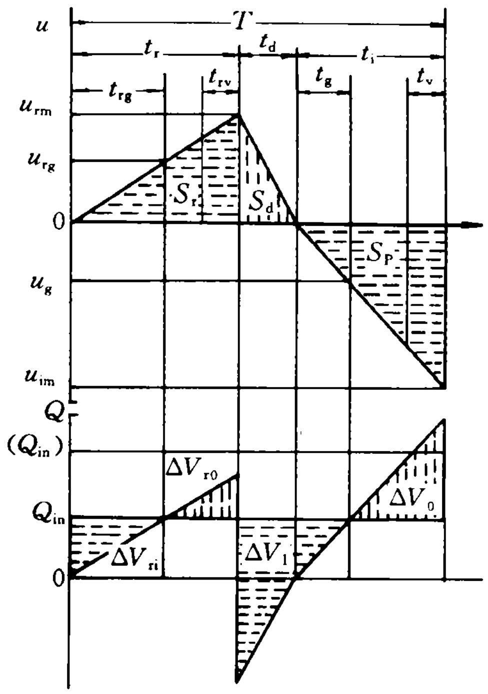

Fig. 1 Variation of the piston speed, displacement, flow quantity, and the oil-charge and discharge of accumulator

$$P_{in} = \frac{\varphi_p^2 + 2\varphi_l Q_{in} - \sqrt{(\varphi_p^2 + 2\varphi_l Q_{in})^2 - 4\varphi_l^2 Q_{in}^2}}{2\varphi_l^2} \tag{43}$$

$$\varphi_p = \begin{cases} \dfrac{\psi_{ur}^2}{\beta}\sqrt{\dfrac{\psi_{sp} A_r^3 S_p}{\psi_{Ar} m_p}} & \text{(RC)} \\ \dfrac{(1+\beta_2)\psi_{ur}^2}{\beta}\sqrt{\dfrac{\psi_{sp} A_r^3 S_p}{\psi_{Ar} m_p}} & \text{(DC)} \end{cases} \tag{44}$$

The impact energy, the impact period/frequency and the working pressure P_{in} of a given hydraulic impact mechanism supplied with varied oil flow and the oil flow Q_{in} needed under given working pressure as well can be calculated using formulae (38)—(40).

When the leaked oil flow is small ($\varphi_l \approx 0$), the formula (42) is converted into a one-dimensional equation of P_{in}, its approximate solution is

$$P_{in} \approx \frac{Q_{in}^2}{\varphi_p^2} \tag{45}$$

Actually when the impactor operates under an ideal working condition, the precise formula for P_{in} or reversely for Q_{in}, is

$$P_{in} = \begin{cases} \dfrac{(1+\beta)^3 m_p}{2\psi_{sp}^2 A_r^3 S_p} Q_{in}^2 & (\text{RC}) \\ \dfrac{m_p}{2\psi_{sp}^2 A_r^3 S_p} Q_{in}^2 & (\text{DC}) \end{cases} \quad (46)$$

The formulae (45) and (46) confirm that the working pressure is approximately proportional to the square of input oil flow, which was generalized using measurement data by other investigators.

8 CONCLUSION

The analysis above can be summed up as following:

(1) It is more practical to divide the piston cycle of the ordinary impact mechanisms, such as RC and DC, into three stages. And the three-stage analysis is formed based on the series of formulae derived above.

(2) With the help of the acceleration ratio β introduced in this paper the therotical analysis of the hydraulic impact mechanism becomes more applicable, and the corresponding design method more simple and practical.

(3) The movement law of the hydraulic impact mechanism is discovered on the analysis of the relationship among the unretrived energy, oil flow quantity and pressures.

The analysis of efficiencies and the design of the mechanism using the formulae in this paper will be discussed further in other perpers.

REFERENCES

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