IDENTIFYING THE BASIC PARAMETERS OF SELF-REGULATING DAMPERS IN CONSTRUCTION MACHINERY

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***Summary**: Used in mechatronics systems, self-regulating dampers are capable of braking in both directions and automatically adjusting the resistance force when the velocity and mass of the actuator change, helping vehicles to avoid being impacted by the external environment and mitigating the judder phenomenon caused by complex terrain. However, the application of self-regulating damper equipment in Vietnam is still limited, especially in the field of construction machinery, so excavators have to work on unfavourable terrain and are negatively affected by external forces, which leads to instability and a lack of balance during soil extraction. Therefore, this article conducts a survey of some basic parameters of self-regulating dampers in construction machinery, the adjustibility of these parameters and the appropriate paramaters for operating such equipment. This article also serves as a theoretical foundation for designing self-regulating dampers in Vietnam.*

***Keywords**: Hydraulic damper; self-regulating damper; construction machinery.*

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1. Introduction

In using any equipment like machinery or vehicles, it is extremely important to avoid the impact of the external environment. One of the components playing an active role in helping vehicles and machines reduce such shocks is hydraulic damping equipment. With the aim of ensuring the highest requirements for the fast and accurate operation of actuators, they help to reduce the velocity of actuators before a given stop. In the field of construction machinery, hydraulic dampers have been applied to excavators, elevators and lift machines… However, the dampers used on construction machinery are mainly viscous dampers and telescopic dampers. The limitation of these devices is the need to adjust them by hand when the velocity or mass of the actuator changes. Moreover, this type of damping structure only brakes in one direction [7,8,9].

In Document [8], the author has given the problem of calculating the basic dimensions of hydraulic self-regulating dampers automatically on mechatronic systems and means of transport. However, there is no scientific research on the application of hydraulic self-regulating dampers on construction machinery in Vietnam, especially large and medium-sized excavators that regularly work on adverse terrain and under the influence of external forces. In addition, the study in Document [8] paid no attention to external factors acting on the piston-rod. Therefore, research is needed to identify the basic parameters of hydraulic self-regulating dampers capable of braking in both directions.

2. Model of Hydraulic Self-Regulating Dampers

2.1 Introduction

Figure 1 presents the structure of hydraulic self-regulating damper equipment, including Body 1, containing the working fluid, Piston-Rods 2 and 3 and the large Piston 4. Body 1 is divided into two chambers, A and B, by Piston 4. In Middle Part 5 of the large diameter section and two Outside Parts 6 and 7 of the small diameter sections, the axial sections are created in Piston-Rods 2 and 3 and Piston 4, in which the small Piston 8 is arranged coaxial with Piston-Rods 2 and 3 and Piston 4.

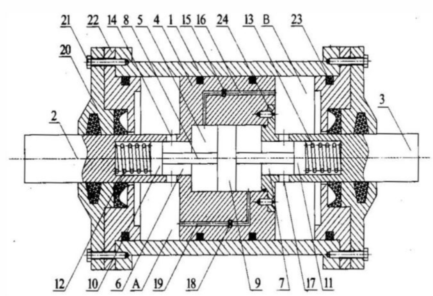

***Figure 1.** Diagramatic view of the hydraulic self-regulating damper [8].*

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Piston-Rod 8 with Middle-Strip 9 of the large-diameter sections and Top-Strips 10 and 11 of the small-diameter sections are arranged appropriately with Middle Cavity Section 5 and External Chamber Section 6 and 7. Meanwhile, Piston-Rods 8 are limited by two elastic Springs 12 and 13. External Chamber 6 is connected with Chamber B by Throttle Holes 14 on Piston Rod 2. It is also separated from Chamber B by a Throttle Disc 15 and located in Axial Pipe 16. Similarly, External Chamber 7 is connected with Chamber A by Throttle Hole 17 on the Handle 3. It is also separated from Chamber A by Throttle Disc 18, which is located in Axial Pipe 19.

The operating principle of the damper: in case of collision with Piston-Rod 2 at any small velocity, which affects 5 and adjusts Throttle Disc 15, moves the whole piston to the right. Then, the pressure in Chamber B is increased and fluid flows through Pipe Shaft 16, Throttle Disc 15, External Chamber 6 and Throttle Holes 14 to Chamber A. The throttling of fluid flow in Throttle Disc 15 makes the speed of the actuator decrease and smooth brake until Main Piston 4 reaches the edge of Body Block 1. When the actuator moves in the opposite direction, the damping will work the same in reverse.

When the velocity of the actuator is increased quickly and moved to the right (Figure 1), the pressure in Chamber B increases and Small Piston 8 takes on the elastic force of Spring 13 and moves to the right. Meanwhile, Throttle Holes 14 starts gradually getting clogged by 10, which reduces the cross-section area of Throttle Holes 14. Therefore, fluid flows through Throttle Holes 14 will be reduced and the braking force will increase, which leads to an increase in the velocity of Piston-Rod 2 and decreases in certain values, ensuring smooth braking and avoiding collision. When the velocity of the actuator increases in the opposite direction, the hydraulic self-regulating damper work similarly in the opposite direction.

2.2 The Dynamic Model and Motion Differential Equations

Figure 2. Where 1 and 3 are the piston-rods; 2 is the large piston; 4 is the small piston; 5 and 9 are springs; 6 and 8 are the external strips of the piston; and 7 is the middle-strip of the piston.

The system of coordinate axes has its root in the starting position with exposure between the actuator and Piston-Rod 1 or 3. Fluid in the cavity of the damper is incompressible. Then, the actuators move with Piston-Rods 1 and 3 at the same velocity and acceleration. Assume that the cross-section of Throttle Hole 14 is a square and Springs 5 and 9 have the same compression features, size and hardness ($k=k_1=k_2$) [8].

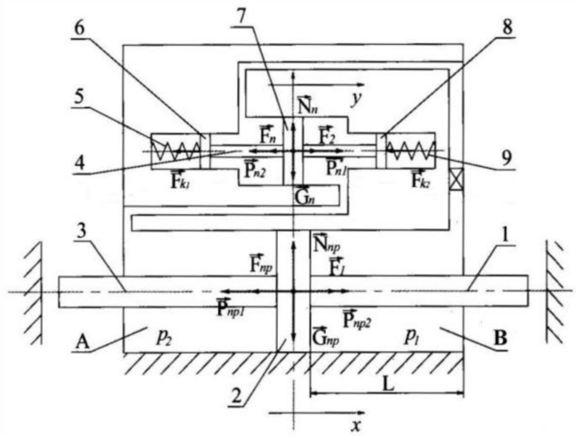

Figure 2. *Diagramatic calculation of the hydraulic self-regulating damper [8].*

The given hypothesis is that, if we ignore the dependence of the elastic modulus of working fluid on the pressure, ignore the dependence of the piston cylinder's volume on the displacement of the piston and ignore the leakage of liquid due to the piston-rod being sealed by the sealing elements, according to [1,8] the equations for the motion of the actuators are:

$$\begin{cases} m_{np}x'' + \lambda_{np}x' + \dfrac{\rho\xi S_{np}^{3}}{2.f^{2}(u)}x = f(t) = F_0; \\ m_n y'' - \lambda_n y' - 2.(k_1 + k_2).y = -\xi.\dfrac{u^2\rho}{2}\Delta S; \end{cases} \qquad (1)$$

where: u is the velocity of fluid flow through the cross-section of the throttle hole; f(u) is the cross-sectional area of the flow; m_{np} is the mass of Large Piston 2 and the actuators moving with Piston-Rods 1 and 3; m_n is the mass of Small Piston 4; λ_{np} is the resistance coefficient of Large Piston 2 on the body of the damper; λ_n is the resistance coefficient of Strips 6, 7 and 8 of Small Piston 4; $\Delta S = S_2 - S_1$ and S_1, S_2 are the working area of External Strips 6 and 8 and Middle Strip 7; S_{np} is the working area of Large Piston 2; ξ is the hydraulic drag coefficient; ρ is the density of the liquid; x, x' and x'' are respectively the x-coordinates, velocity and acceleration of Large Piston 2; y, y' and y'' are respectively the y-coordinate, velocity and acceleration of Small Piston 4; k_1, k_2 are the stiffness coefficient of Springs 5 and 9; F_0 is the external force effect on the rod of Piston 2.

We suppose that at time t = 0 we exert a constant force $F = F_0$ on the oscillating system. Mathematically, we can represent the force as follows: $f(t) = \begin{cases} 0 & \text{When } t \le 0 \\ F_0 & \text{When } t > 0 \end{cases}$ (2)

Divide both sides of Equation (1) for m_{np} and m_n and we obtain a system of differential equations in the form:

$$\begin{cases} x'' + 2.\delta_{np}.x' + \dfrac{\rho\xi S_{np}^3}{2.m_{np}.f^2(u)}.x = \dfrac{F_0}{m_{np}}; & \text{(a)} \\ y'' - 2.\delta_n.y' - \dfrac{2.(k_1+k_2)}{m_n}.y = -\xi.\dfrac{u^2\rho}{2.m_n}\Delta S; & \text{(b)} \end{cases} \quad (3)$$

where: $\omega_{0x} = \sqrt{\dfrac{\rho\xi S_{np}^3}{2.m.f^2(u)}}$, the oscillation frequency according to axis OX; $\omega_{0y} = \sqrt{\dfrac{2.(k_1+k_2)}{m_n}}$, the oscillation frequency according to axis OY; $\delta_x = \dfrac{\lambda}{2.m_{np}}$ and $\delta_y = \dfrac{\lambda}{2.m_n}$ are viscous drag coefficients. According to [1, 2], Equation (3a) has a general solution form:

$$x_{(t)} = \frac{F_0.2.m_{np}.f^2(u)}{\rho.\xi S_{np}^3}\left[1 - \frac{e^{-\delta_x.t}}{\sqrt{1-D_x^2}}\cos(\omega_x.t - \varphi_x)\right] \quad (4)$$

$$x'_{(t)} = \frac{F_0.2.m_{np}.f^2(u)}{\rho.\xi S_{np}^3}\frac{\omega_x}{\sqrt{1-D_x^2}}e^{-\delta_x.t}.\sin(\omega_x t - \varphi_x) \quad (5)$$

$$x''_{(t)} = \frac{F_0.2.m_{np}.f^2(u)}{\rho.\xi S_{np}^3}\frac{\omega_x^2}{\sqrt{1-D_x^2}}e^{-\delta_x.t}.\cos(\omega_x t - \varphi_x) \quad (6)$$

$$\omega_x = \omega_{0x}\sqrt{1-D_x^2};\ \cos\varphi_x = \sqrt{1-D_x^2} \quad (7)$$

Similarly, Equation (3b) has a general solution form:

$$y_{(t)} = -\xi.\Delta S.\frac{u^2.\rho}{4.(k_1+k_2)}.\left[1 - \frac{e^{-\delta_y.t}}{\sqrt{1-D_y^2}}\cos(\omega_y.t - \varphi_y)\right] \quad (8)$$

$$y'_{(t)} = -\xi.\Delta S.\frac{u^2.\rho}{4.(k_1+k_2)}.\frac{\omega_y}{\sqrt{1-D_y^2}}e^{-\delta_y.t}.\sin(\omega_y t - \varphi_y) \quad (9)$$

$$y''_{(t)} = -\xi.\Delta S.\frac{u^2.\rho}{4.(k_1+k_2)}.\frac{\omega_y^2}{\sqrt{1-D_y^2}}e^{-\delta_y.t}.\cos(\omega_y t - \varphi_y) \quad (10)$$

$$\omega_y = \omega_{0y}\sqrt{1-D_y^2};\ \cos\varphi_y = \sqrt{1-D_y^2} \quad (11)$$

where D is Lehr resistance, calculated by the formula: $D_x = \dfrac{\delta}{\omega_{0x}}$ and $D_y = \dfrac{\delta}{\omega_{0y}}$

3. Caculating and Investigating the Hydraulic Self-Regulating Damper in Construction Machinery

3.1 Preliminary Calculation of Some Parameters of the Hydraulic Self-Regulating Damper

We assume that both the size and rigidity of Springs 5 and 9 are equal and have compression properties serving to calculate [8] $k = k_1 = k_2 = 24,5$ (N/m). Serving computational simulation, the assumption is that each piston-rod of the hydraulic self-regulating damper will be affected by an external force F_0 = 47578.5 (N). According to [5], working cylinder diameter dk will be determined by the required pressure F_{cb}.

$$d_k = \sqrt{\frac{4.F_{cb}}{\pi.P_d}} = \sqrt{\frac{4.47578,5}{\pi.6.10^6}} \approx 0,1\ \text{(m)}; \quad (12)$$

where P_d is the working pressure of the hydraulic oil in the system P_d = 5 - 10 (MN/m^2), we choose P_d = 6 (MN/m^2);

The diameter of Large Piston 2 is $d_k \approx 0.1$ (m), so the mass of Large Piston 2 is

$$m_{pt} = \frac{\pi d_k^2}{4}.L_{pt}.\gamma = \frac{\pi.0,1^2}{4}.0,1.8100 \approx 6,3\ \text{(kg)}. \quad (13)$$

where γ = 8100 kg/m³ is the density of piston material made from stainless steel, and L_{pt} = 0,1 (m) is the length of the piston. Similarly, the diameter of the rod is $d_c = \frac{1}{3}.d_k = \frac{0,1}{3} = 0,034$ (m), the length of the rod is L_{ct}= 0,2 (m), and the mass of the rod is:

$$m_{ct} = \frac{\pi d_c^2}{4}.L_{ct}.\gamma = \frac{\pi.0,034^2}{4}.0,2.8100 \approx 1,5 \text{ (kg)}. \tag{14}$$

From (13) and (14), we calculated $m = m_{pt} + m_{ct}$ = 6,3 + 1,5 = 7,8 (kg).

According to [6], we chose hydraulic oil 46 HP Oil Hydraulic AW, which has an appropriate viscosity, viscosity index, purity and high-durability oxidation, abrasion resistance, water separation property, demulsifying property and high anti-foaming. Thus, we chose hydraulic 46 HP Oil Hydraulic AW for experimental study of the hydraulic self-regulating damper with the proportion at 20°C is 0.8750 kg/l, kinematic viscosity at 40°C being υ = 41-50 m²/s. According to [3,4] we calculate the coefficient of hydraulic resistance based on the length of the cylinder according to the formula:

$$\lambda = \frac{0,312}{d_t^{0,226}}.\left(1,9.10^{-6} + \frac{\upsilon}{v}\right)^{0,226} \tag{15}$$

where $d_t = d_k$ is the internal diameter of the throttle cylinder; (m); λ is the coefficient of hydraulic resistance by cylinder length; υ is the kinematic viscosity coefficient of hydraulic oil; υ = 45 (m²/s); v is the average velocity of hydraulic oil flow in the cylinder; v=0, 5 (m/s). The resistance coefficient of Large Piston 2 with d_t = 0, 1 (m):

$$\lambda_{nb} = \frac{0,312}{0,1^{0,226}}.\left(1,9.10^{-6} + \frac{45}{0,5}\right)^{0,226} \approx 1,45 \tag{16}$$

Similarly, the resistance coefficient of Small Piston 4 with d_t = 0,035 (m):

$$\lambda_n = \frac{0,312}{0,035^{0,226}}.\left(1,9.10^{-6} + \frac{45}{0,5}\right)^{0,226} \approx 1,84 \tag{17}$$

Cross-sectional area of the flow: $f(u) = \frac{\pi.d_t^2}{4} = \frac{3.14.0,1^2}{4} = 7,85.10^{-3}$ (m²).

According to [4], we can calculate the coefficient of local losses by using Formula (18) with (l_{td} = 0,015 m being equivalent to the length of the pipe).

$$l_{td} = d_t.\frac{\xi}{\lambda} \rightarrow \xi = \frac{l_{td}\,\lambda}{d_t} = \frac{0,015.1,45}{0,1} = 0,2 \tag{18}$$

In addition, according to the table of the coefficient of local losses through the throttle holes in [3] we have $\zeta_v = 2 \div 3 \rightarrow$ we choose $\zeta_v = 2$ (19)

From (18) and (19) we calculated: $\zeta = \zeta_{td} + \zeta_v = 2,2$ (20)

According to [8], the stable velocity of the actuators is v_{ycm} = 0,01 m/s, $\Delta S = S_2 - S_1$ = (50 – 100) mm², we choose ΔS = 80 mm² = 80.10⁻⁶ m².

Table 1. *The basic specifications of the hydraulic self-regulating damper.*

ST	Specifications	Notation	Value
1	Velocity of flow through the cross-section of throttle hole, (m/s)	u	0,5
2	Cross-sectional area of the flow, (m²)	f(u)	$7,85.10^{-3}$
3	Mass of Large Piston 2 and the actuators, (kg)	m	7,8
4	Resistance coefficient of Piston 2 on the body of the damper	λ_{nb}	1,45
5	Resistance coefficient of Strips 6, 7 and 8 and Small Piston 4	λ_n	1,84
6	Hydraulic drag coefficient	ξ	2,2
7	The difference between the cross-section area of the strips, (m²)	ΔS	80.10^{-6}
8	The density of the liquid, (kg/l)	ρ	0,8750
9	Elastic coefficient of Spring 5, (N/m)	k_1	24,5
10	Elastic coefficient of Spring 9, (N/m)	k_2	24,5

3.2 Investigating Some Basic Parameters of the Hydraulic Self-Regulating Damper

System Oscillator Equation (3), including the oscillation equation according to the OX axis, has General Solutions (6), (7) and (8), and the oscillation equation according to the OY axis has General Solutions (10), (11) and (12), corresponding to the variables x, x' and x'', being the coordinates, velocity and acceleration of Large Piston 2 according to the OX axis, and y, y' and y'' being the coordinates, velocity and acceleration of Small Piston 4, according to the OY axis. By using the parameters of equipment in Table 1, we conducted a survey of some basic parameters of the hydraulic self-regulating damper based on MATLAB software, and obtained the following results:

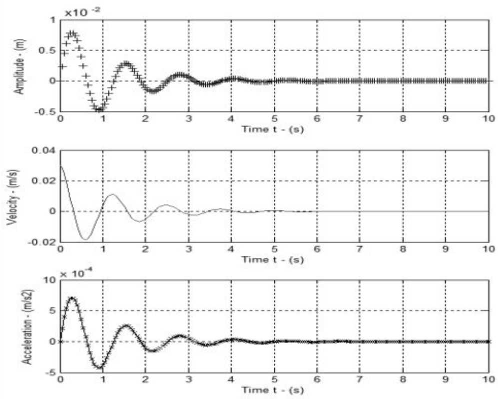

***Figure 3.** The amplitude, velocity and acceleration of the large piston depend on time t, (s).*

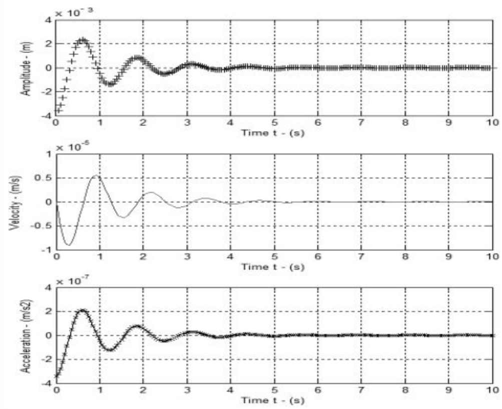

***Figure 6.** The amplitude, velocity and acceleration of the small piston depend on time t, (s).*

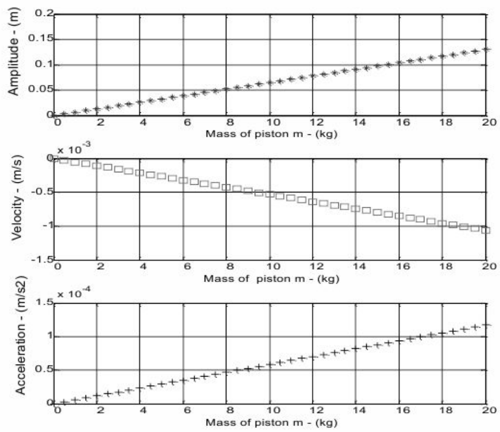

***Figure 4.** The amplitude, velocity and acceleration of the large piston depend on the mass of piston m_{np}, (kg).*

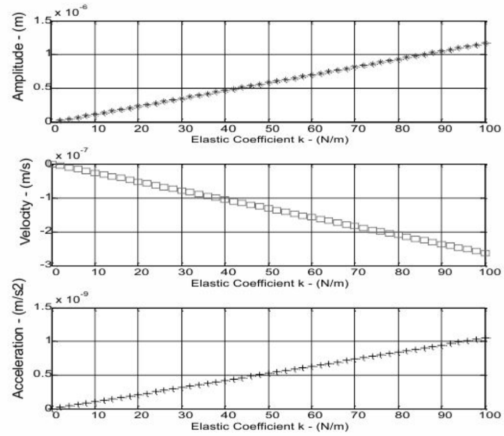

***Figure7.** The amplitude, velocity and acceleration of the small piston depend on elastic coefficient k, (N/m).*

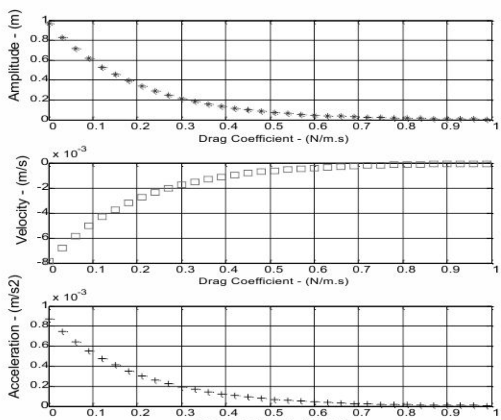

***Figure 5.** The amplitude, velocity and acceleration of the large piston depend on drag coefficient δ_x, (N/m.s).*

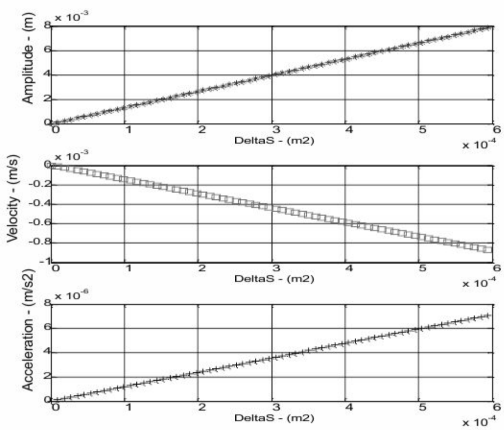

***Figure 8.** The amplitude, velocity and acceleration of the small piston depend on ΔS, (m^2).*

Discussion:

As can be seen from Figures 3 and 6, by using hydraulic self-regulating dampers when external forces affect the rod of Large Piston 2, the displacement, velocity and acceleration of the piston will decrease over time and gradually reach zero. The largest oscillation amplitude of Piston 2 is $A_{max} \approx 7 \div 8$ (mm). Balancing the external pressure effect, Small Piston 4 tends to have a reverse-phase oscillation frequency and eliminate vibrations, with a smaller oscillation amplitude $A_{max} \approx 2 \div 4$ (mm) due to the resistance of hydraulic oil. After a period of time $t = 2 \div 4$ (s), both oscillations are eliminated completely. This allows the excavator to be stable while moving on complex terrain.

Figure 4 demonstrates that the mass of Large Piston 2 has a linear relationship with the displacement, velocity and acceleration of the piston. When we change the mass of the piston, it also affects the working of the damper. In detail, when the mass of the piston increases then the displacement of the piston gradually increases, as in the process of working the piston will incur a partial load itself. Similarly with velocity, there is also a small change in velocity when we increase the mass of the piston. For example, at the mass value m=7, 8 (kg) the velocity value $v=x'(m) \approx 0.5.10^{-3}$ (m/s) and acceleration $a=x''(m) \approx 0.5.10^{-4}$ (m^2/s). The values of both the velocity and acceleration are very small, ensuring the mobility of the equipment is minimal during the working process.

Particularly, the coefficient of viscous drag has a nonlinear relationship with the displacement, velocity and acceleration of the piston (Figure 5). When the drag coefficient increases, the displacement, velocity and acceleration of the piston decrease quickly and reach zero. If the drag coefficient is more than 1 (N/m.s) then the piston stays unchanged.

Figure 7 illustrates the relationship between the displacement of Small Piston 4 and the stiffness drag coefficient of the springs. However, the fluctuation of the piston's oscillation amplitude is very small. In detail, at the value $k \approx 24, 5$ (N/m), the oscillation amplitude of piston $A \approx 0,3.10^{-6}$ (m).

Similarly, Figure 8, when the difference between the cross-sectional area of External Strips 6 and 8 and Middle Strip 7 is changed, it will make the cross-sectional area of the actuator change, which leads to the pressure flow changing and creates a difference in pressure, making the displacement of Small Piston 4 increase. For instance, when $\Delta S = 80.10^{-6}$(m^2), the displacement of Small Piston 4 is $A \approx 1,5$ (mm). Thus, in order to work more effectively, braking processes are correct and this equipment has a compact structure, so we should not choose too large a difference between the strip pistons' area.

4. Conclusions

In this limited research paper, the authors have surveyed the oscillation parameters of the piston clusters, such as amplitude, velocity and acceleration, as well as the relationship between them with the structural parameters of the hydraulic self-regulating damper. With the results of this theoretical research and numerical survey, this paper demonstrates the effectiveness of the hydraulic self-regulating damper in construction machinery, as well as the adjustibility of these parameters to achieve the best machine operation. The research results of this paper also serve as a basis for the experimental calculation, design and manufacture of hydraulic self-regulating dampers on mechatronic systems and construction machinery equipment in Vietnam.

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