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Influence of Valve-Orifice Geometry on Dynamic Response and Synchronization of Master–Slave Hydraulic Cylinders

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ABSTRACT: This study investigates the influence of valve-orifice geometry on the dynamic response and synchronization performance of master-slave hydraulic cylinder systems, with particular emphasis on transient fluid-structure interactions in electrically driven hydraulic systems. A combined theoretical and experimental investigation is conducted to elucidate the interaction between orifice geometry and the transient response of the master-slave system. A dynamic coupling model is developed that incorporates the geometry-dependent area gradient (dA/dx) and mechanical feedback delay, enabling the underlying mechanisms of pressure surges, post-surge time lag, and disturbance recovery during nonlinear throttling to be identified. Based on these mechanisms, adaptive composite valve design guidelines are proposed. The results show that the rectangular orifice provides the highest flow gain and the fastest response, with the active-cylinder spool flow reaching stabilization in 0.19 s, representing reductions of 9.1% and 14.8% compared with trapezoidal and U-shaped orifices, respectively. The corresponding follower-cylinder delay is only 0.03 s, 16.7% and 30.2% shorter than those of the trapezoidal and U-shaped orifices. In contrast, the trapezoidal throttle slot provides superior buffering performance, with a buffering time of 0.12 s, while also exhibiting a faster response to disturbances. The rectangular orifice, however, provides greater load stiffness.

KEYWORDS: AMESim; digital hydraulic cylinder; master-slave control; valve port structure

1 Introduction

Digital fluid power has developed from early parallel on/off-valve concepts into a broader family of switching, independently metered, and digitally controlled linear-actuation architectures [1–5]. Foundational studies demonstrated accurate hydraulic-cylinder trajectory tracking with parallel on/off valves [2], while later work introduced control-oriented models and model-based switching for digital independent-metering cylinder drives [3] and robust force/position tracking with digital hydraulic valves [6]. System-level reviews and industrial perspectives have further emphasized the potential of digital fluid power for compactness, robustness, software-defined functionality, and energy-efficient actuation [1,4,7]. Optimal switching and feed-forward strategies have also been investigated to suppress oscillations in fast digital hydraulic drives [8]. Within this broader field, stepper-driven digital hydraulic cylinders are attractive because they combine digital command compatibility with high force density. Sharipov and Mesrpyan [9] investigated the stepper-motion problem in digital servo control of hydraulic actuators, while an earlier AMESim study provided the schematic basis for the digital servo stepping cylinder considered in this work [10]. More recent studies have examined multi-pressure actuation, multi-actuator digital-hydraulic control, multi-chamber energy efficiency, miniaturized digital valve packages, and robust model-based tracking [11–15].

Collectively, these results show that valve configuration, switching strategy, compressibility, friction, leakage, dead zones, and feedback design can strongly affect tracking accuracy and transient response. However, synchronized operation of mechanically feedback-controlled digital hydraulic cylinders remains comparatively underexplored.

High-precision synchronization is a broader coordinated-motion problem. Aarnoudse [16] demonstrated that cross-coupled control can substantially improve coordinated multi-axis contour tracking by explicitly incorporating contour-error information into the control of individual motion axes. Recent studies have integrated cross-coupled control with iterative learning control to improve contour-tracking accuracy in repetitive precision motion systems, while addressing convergence and computational efficiency [17]. Synchronization concepts have also been applied to hybrid hydraulic/electromechanical aircraft actuation [18], twin-axis table drives [19], multi-axis electro-hydraulic manipulators [20], and electro-hydraulic press tooling [21]. These studies demonstrate that explicit coordination of inter-axis error can substantially improve synchronism despite disturbances and model uncertainty. Nevertheless, digital hydraulic cylinders introduce additional electromechanical-hydraulic coupling through the motor, valve spool, screw transmission, and mechanical feedback loop; achieving high-precision master-slave synchronization therefore remains challenging.

Valve-port geometry is another key determinant of digital-valve and actuator performance because it governs the relationship between spool displacement and effective metering area. Gillella and Sun [22] developed a systematic method for analyzing and synthesizing spool-notch geometries to realize prescribed metering-area schedules. Kosek and Sliwinski [23] investigated how control-edge geometry influences the flow characteristics of proportional directional valves through CFD-based design and experimental validation. Wei et al. [24] compared non-circular-opening spool-valve structures for digital hydraulic cylinders and provided the schematic basis for the three valve-port configurations considered in this study. Collectively, these studies establish a clear link between notch geometry and valve dynamics; however, most emphasize valve-level or single-actuator performance rather than the coupled response of digital hydraulic cylinders operating in a master-slave synchronization architecture.

The current research on digital hydraulic cylinders mainly concentrates on the optimization of control algorithms, the design of the whole system structure, or the performance study of certain valve types. However, when it comes to the special architecture of “synchronous control”, there are still few investigations related to the effects of various configurations of valves’ ports upon the entire performance of digital hydraulic cylinders. There exist many problems that need further discussions, such as the bad synchronization performance, the poor disturbance resistance capacity, and the performance variation in different valve configurations due to their complicated interaction during synchronization. The theoretical analysis and simulation have not yet been conducted to clarify which valve configurations can better improve the overall synchronization performance of the system. In the field of numerous control schemes, the master-slave control scheme possesses high value in complex digital hydraulic cylinder systems because of its simple structure, easy implementation of multidirectional synchronized control, and disturbance resistance ability.

2 Structural Composition and Working Principle of Digital Hydraulic Cylinders

Fig. 1 shows the construction principle of the investigated digital hydraulic cylinder. Its major components include a stepper motor, a four-way directional control valve, a screw-nut mechanism, a feedback nut, a piston, a ball screw, and a cylinder block. The principle of its operation is illustrated below. First of all, when receiving a command signal, the stepper motor turns depending on the command signal. Then,

the screw-nut mechanism transforms this rotary movement of the stepper motor into an axial movement of the valve spool. In other words, it opens the valve port, and as a result, movement of the hydraulic cylinder occurs. When the hydraulic oil enters the cylinder body, the piston rod moves in a linear way. The piston rod is a part of the feedback nut and ball screw system. As a result, it moves the feedback nut and the latter drives the ball screw rotation. In other words, the screw-nut system causes rotary movement of the valve spool in the opposite direction up to valve port closing. In this way, the piston rod reaches the required location.

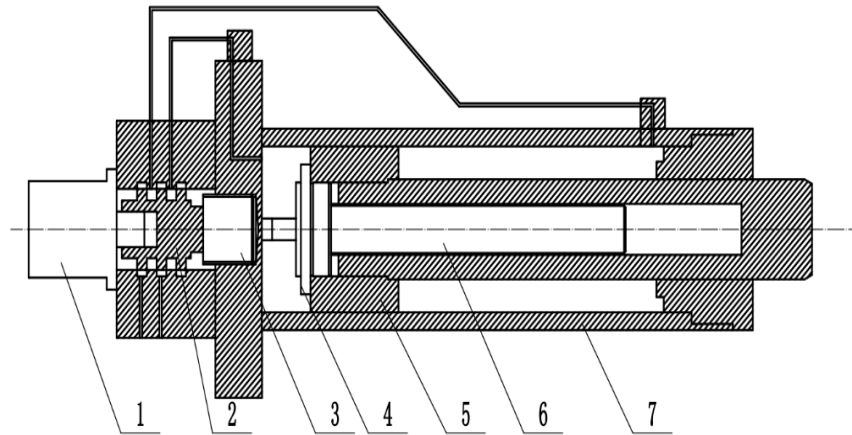


Figure 1: Schematic diagram of digital hydraulic cylinder structure (adapted from [10]). 1. Stepper motor. 2. Four-way directional valve. 3. Screw-nut assembly. 4. Feedback nut. 5. Piston. 6. Ball screw. 7. Cylinder block.

3 Establishment of Digital Model of Digital Hydraulic Cylinder

The terms in the following formulas are shown in Table 1.

Table 1: Table of nomenclature.

Nomenclature	Description	Unit
T_m	Maximum static torque	$N \cdot m$
θ_b	Inherent step angle	rad
n_d	Drive fractional number	dimensionless
Z_τ	Rotor tooth number	dimensionless
θ_τ	Actual output angle	rad
J_τ	Rotating inertia of rotor	$kg \cdot m^2$
B_τ	Rotor viscous damping coefficient	$N \cdot m \cdot s/rad$
$\theta_{\tau 1}$	Stepper motor angle	rad
θ_v	Angle of valve core	rad
r_τ	Stepper motor shaft radius	m
K_{vg}	Contact stiffness of two contact surfaces of sliding joint	N/m
B_{vg}	Contact Damping of Two Contacting Surfaces of a Sliding Pair	Ns/m
θ_p	Screw angle	rad
r_v	Valve stem radius	m
α_v	Helical pitch angle of feedback nut	rad

Table 1: *Cont.*

Nomenclature	Description	Unit
x_v	Displacement of valve core	m
K_{vs}	Contact stiffness of the feedback nut	N/m
B_{vs}	Contact damping of feedback nut	Ns/m
x_p	Piston displacement	m
α_p	Helix angle of ball screw	rad
r_p	Ball screw radius	m
K_{ps}	Contact stiffness of ball screw thread pair	N/m
B_{ps}	Damping of Ball Screw Thread Pair	Ns/m
M_v	Valve core quality	kg
B_{vp}	Viscous damping coefficient of valve core translational motion	Ns/m
J_v	Rotating inertia of valve core	$\text{kg} \cdot \text{m}^2$
B_{vz}	Viscous damping coefficient of valve core rotation	$\text{N} \cdot \text{m} \cdot \text{s/rad}$
J_s	Rolling ball screw moment of inertia	$\text{kg} \cdot \text{m}^2$
B_s	Viscous damping coefficient of ball screw rotation	$\text{N} \cdot \text{m} \cdot \text{s/rad}$
s	Laplace Variable	dimensionless

(1) Mathematical modeling of stepper motor

Eqs. (1)–(16) describe the electromechanical transmission and feedback dynamics of the digital hydraulic cylinder, including the stepper-motor torque balance, the sliding-pair contact relation, the spool-feedback-nut thread pair, and the ball-screw-piston pair. The formulations follow standard dynamic-system and hydraulic-control modeling principles [25].

Let us assume that the pulse number in the input side of the stepper motor is n , and the significance of the number just refers to the direction of the pulse number, but not its quantity. More precisely, if $n > 0$, then it refers to a positive pulse, and if $n < 0$, then it refers to a negative pulse. The torque equilibrium expression of the stepper motor can therefore be formulated as: ([25])

$$T_m \sin \left(Z_\tau \left(\frac{n\theta_b}{n_d} - \theta_\tau \right) \right) = J_\tau \frac{d^2\theta_\tau}{dt^2} + B_\tau \frac{d\theta_\tau}{dt} \quad (1)$$

(2) Mathematical modeling of sliding pair between stepper motor and valve core

Relative displacement of two adjacent contact surfaces between a stepper motor and a valve core [25].

$$Z_{vg} = (\theta_{\tau 1} - \theta_v) r_\tau \quad (2)$$

Supporting force of two adjacent contact surfaces between the stepper motor and the valve core

$$F_{ng} = K_{vg} Z_{vg} + B_{vg} \frac{dZ_{vg}}{dt} \quad (3)$$

The Laplace transform of the mathematical Eq. (3) yields

$$F_{ng} = K_{vg}Z_{vg} + B_{vg}Z_{vg}s \quad (4)$$

(3) Mathematical modeling of the thread pair between the valve core and the feedback nut

Vertical distance between adjacent tooth surfaces [25]

$$Z_{vs} = (\theta_v - \theta_p)r_v \sin \alpha_v - x_v \cos \alpha_v \quad (5)$$

Support force between adjacent tooth surfaces

$$F_{nv} = K_{vs}Z_{vs} + B_{vs} \frac{dZ_{vs}}{dt} \quad (6)$$

The Laplace transform of the mathematical Eq. (6) yields

$$F_{nv} = K_{vs}Z_{vs} + B_{vs}Z_{vs}s \quad (7)$$

(4) Mathematical Modeling of Ball Screw and Piston Thread Pair

The normal relative displacement between adjacent tooth surfaces is

$$Z_p = x_p \cos \alpha_p - \theta_p r_p \sin \alpha_p \quad (8)$$

The normal support forces between adjacent tooth surfaces are

$$F_{np} = K_{ps}Z_p + B_{ps} \frac{dZ_p}{dt} \quad (9)$$

The Laplace transform of the mathematical Eq. (9) yields

$$F_{np} = K_{ps}Z_p + B_{ps}Z_p s \quad (10)$$

The dynamic equilibrium equation of the valve stem is

$$F_{nv} \cos \alpha_v = M_v \frac{d^2 x_v}{dt^2} + B_{vp} \frac{dx_v}{dt} \quad (11)$$

Taking the Laplace transform of the mathematical Eq. (11), we obtain

$$F_{nv} \cos \alpha_v = M_v X_v s^2 + B_{vp} X_v s \quad (12)$$

The torque equilibrium equation of the valve core is

$$F_{ng} r_\tau = J_v \frac{d^2 \theta_v}{dt^2} + B_{vz} \frac{d\theta_v}{dt} + r_v F_{nv} \sin \alpha_v \quad (13)$$

The Laplace transform of the mathematical Eq. (13) yields

$$F_{ng} r_\tau = J_v \theta_v s^2 + B_{vz} \theta_v s + r_v F_{nv} \sin \alpha_v \quad (14)$$

Equation of balance of torque of screw

$$r_p F_{np} \sin \alpha_p = J_s \frac{d^2 \theta_p}{dt^2} + B_s \frac{d\theta_p}{dt} + r_v F_{nv} \sin \alpha_v \quad (15)$$

Taking the Laplace transform of the mathematical Eq. (15), we obtain

$$r_p F_{np} \sin \alpha_p = J_s \theta_p s^2 + B_s \theta_p s + r_v F_{nv} \sin \alpha_v \quad (16)$$

4 Determination of the Equivalent Flow Area at the Valve Opening

4.1 Calculation of Flow Area at Throttle Slot Opening

Spool-valve orifices can be classified by their opening geometry into full-perimeter and non-full-perimeter configurations. As shown in Fig. 2, a full-perimeter-opening spool forms a continuous annular metering edge. When the spool is displaced axially by x , the circumferential clearance between the spool shoulder and the valve-body port forms the throttling section (region I). In contrast, a non-full-perimeter spool valve forms its metering area through discrete throttle grooves machined around the spool shoulder, producing several parallel flow passages (region II in Fig. 3). Pressure-distribution measurements for non-circular-opening spool valves indicate that the effective throttling section is generally located in the region of the steepest pressure gradient [26].

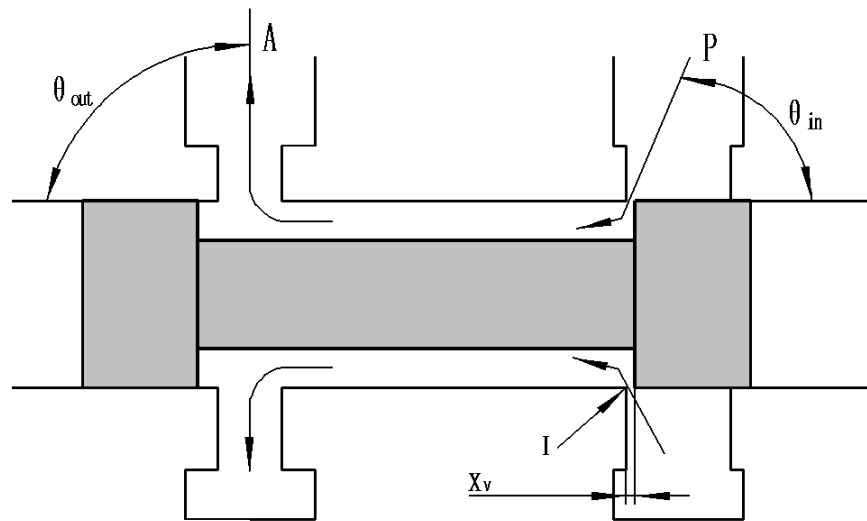


Figure 2: Full-circuit open-close valve.

When hydraulic fluid flows through the throttle groove structure, the effective flow cross-sectional area of the throttling passage is dynamically adjusted by altering the valve opening. This enables diverse flow regulation effects.

The word “dynamically” implies a constant evolution of the flow area with respect to the movement of the spool. Unlike a traditional spool that does not have any grooves, where dA/dx will be very large at the point of commencement of the flow area, in this design the grooves allow for the gradual evolution of the flow area in terms of flow gain that starts with low values initially. This ensures that hydraulic pressure shock is prevented due to the smooth transition of energy from the supply side to the cylinder chambers.

The schematic diagrams illustrating the three valve port configurations considered in this study are shown in Fig. 4, and the parameter table is provided in Table 2.

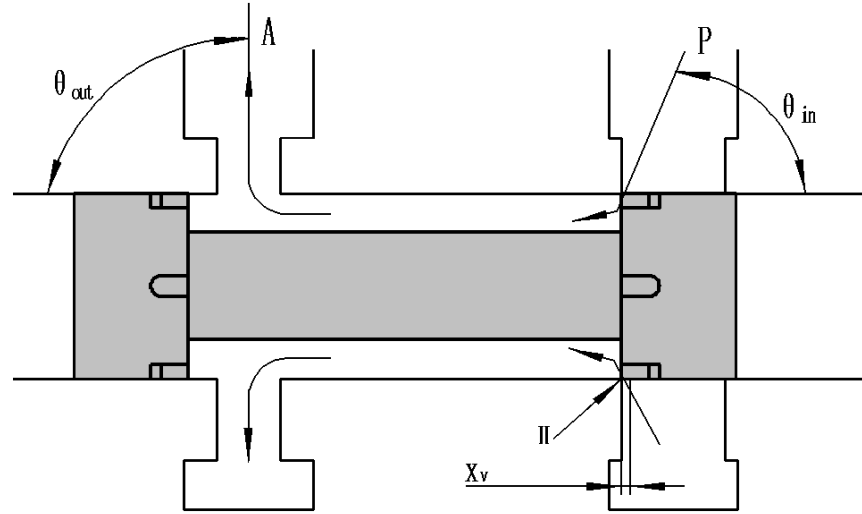


Figure 3: Partially opened valve.

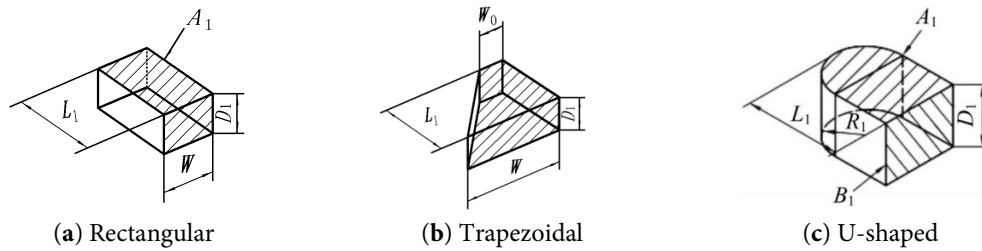


Figure 4: Three types of valve port schematic diagrams (a–c) (adapted from [25]).

Table 2: Model parameters.

Category	Rectangular	Trapezoidal	U-Shaped
Rs	7.5 mm	7.5 mm	7.5 mm
L1	2 mm	2 mm	2 mm
D1	2 mm	2 mm	2 mm
W	3 mm	4 mm	3.76 mm
W0		2 mm	
θ		53.14°	
R1			1.88 mm

In order to conduct an objective and scientifically valid comparison among the three throttling orifice shapes, a stringent design normalization condition was applied:

Common Maximum Opening Area ($A_{\max}$) and Stroke ($x_{\max}$):

Each of the three designs, that is rectangular, trapezoidal, and U-shaped designs, had the same maximum stroke ($x_{\max} = 3$ mm), and the three designs were normalized to give equal maximum flow passage area ($A_{\max} = 6$ mm²) when fully open ($x = x_{\max}$).

Identical Rated Flow Capacity (Q_{rated}):

Under a standardized pressure drop ($\Delta p = 15$ MPa), each shape produces equal maximum rated flow capacity ($Q_{\text{rated}} \approx 47.34$ L/min).

Controlled Geometrical Variables:

The axial slot length ($L_1 = 3$ mm) and spool stroke are fixed across all models. In other words, the comparative study focuses on the effect of the area expansion rate ($\frac{dA}{dx}$) which is constant for the rectangular design, linear for the trapezoidal design, and quadratic for the U-shaped design.

The formula for calculating the equivalent flow area of valve orifices in throttle slots with different structures within synchronous hydraulic systems is detailed in Table 3 below.

Table 3: Formula for calculating the equivalent flow area of throttle channels with different valve orifice configurations.

Valve Port Structure	Equivalent Current-Carrying Area Calculation Formula
rectangle	$0 < x < L_1$ $W = 2R_1$ $H = \sqrt{R_s^2 - \left(\frac{W}{2}\right)^2}$ $\beta = 2\arctan\left(\frac{W}{2H}\right)$ $A_1 = \int_0^x \beta R_s dx$ $A_2 = \frac{1}{2}R_s^2 \cdot \beta - \frac{1}{2}W \cdot H + W(D_1 - R_s + H)$ $A = \frac{n}{\sqrt{\frac{1}{A_1^2} + \frac{1}{A_2^2}}}$
trapezoidal	$0 < x < L_1$ $W = W_0 + 2x\tan\left(\frac{\theta}{2}\right)$ $H = \sqrt{R_s^2 - \left(\frac{W}{2}\right)^2}$ $\beta = 2\arctan\left(\frac{W}{2H}\right)$ $A_1 = \int_0^x \beta R_s dx$ $A_2 = \frac{1}{2}R_s^2 \cdot \beta - \frac{1}{2}W \cdot H + W(D_1 - R_s + H)$
U-shaped	$\text{When } 0 < x < R_1$ $W = 2\sqrt{R_1^2 - (R_1 - x)^2}$ $H = \sqrt{R_s^2 - \left(\frac{W}{2}\right)^2}$ $\beta = 2\arctan\left(\frac{W}{2H}\right)$ $A_1 = \int_0^x \beta R_s dx$ $A_2 = \frac{1}{2}R_s^2 \cdot \beta - \frac{1}{2}W \cdot H + W(D_1 - R_s + H)$ $\text{When } R_1 < x < L_1$ $W = 2R_1$ $H = \sqrt{R_s^2 - \left(\frac{W}{2}\right)^2}$ $\beta = 2\arctan\left(\frac{W}{2H}\right)$ $A_1 = \frac{1}{2}\pi R_1^2 + \int_{R_1}^x \beta R_s dx$ $A_2 = \frac{1}{2}R_s^2 \cdot \beta - \frac{1}{2}W \cdot H + W(D_1 - R_s + H)$ $A = \frac{n}{\sqrt{\frac{1}{A_1^2} + \frac{1}{A_2^2}}}$

In the above formula, x represents the valve opening; R_s denotes the valve core radius; R_1 indicates the characteristic radius of the throttling slot; D_1 signifies the characteristic depth of the throttling slot; L_1 represents the characteristic length of the throttling slot; W denotes the width of the throttling slot; W_0 indicates the upper base of the trapezoidal shape; n represents the number of throttling slots; H denotes the distance between slot boundaries; β denotes the angle between the width cross-section of the slot; θ represents the total angle of the trapezoidal throttling slot.

4.2 Analysis of Equivalent Flow Area at Throttle Slot Orifice

By calculating using the above equivalent cross-sectional area formula, the curves showing how the orifice area A of rectangular, trapezoidal, and U-shaped throttle slots varies with orifice opening x are shown in Fig. 5.

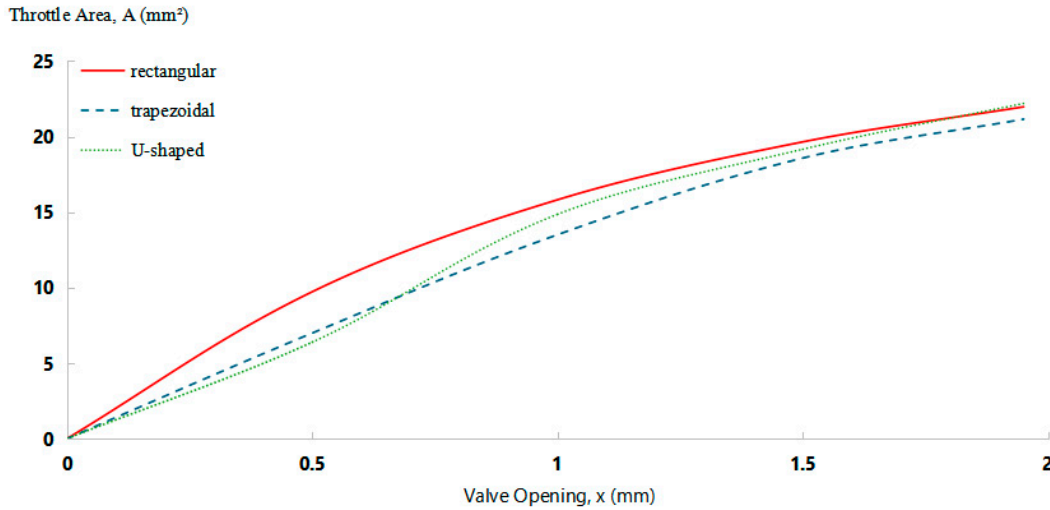


Figure 5: Relationship curves between equivalent flow area and valve core opening for three valve port configurations.

The rectangular throttle slot provides a high area gain at small openings, which promotes rapid flow buildup and improves small-displacement response. Opening of the U-shaped groove is not instantaneous. It allows for a more gradual opening of the groove, which gives rise to a less abrupt increase in the flow rate with increased spool movement. In effect, this helps in avoiding drastic pressure changes. Trapezoidal groove lies somewhere in between the other two type.

In order to explain the relation of the steady-state gradient of geometric area ($\frac{dA}{dx} = W$) stated on page 15 and the changing flow gain, a distinction must be made between pure geometric attributes and operational hydraulic dynamics.

The operational hydraulic flow gain K_q is defined as $K_q = \frac{\partial Q}{\partial x}$, which accounts for both the geometric area gradient and the transient throttle pressure drop Δp :

$$K_q = \frac{\partial Q}{\partial x} = C_d \sqrt{\frac{2}{\rho}} \left(\frac{dA}{dx} \sqrt{\Delta p} + \frac{A(x)}{2\sqrt{\Delta p}} \frac{\partial \Delta p}{\partial x} \right) \quad (17)$$

The geometrical area gradient, $\frac{dA}{dx}$, for the rectangle shape is constant (W), but the increase in the opening of the valve results in higher flow rates and causes higher losses in the pipe and storage chamber; therefore, there will be a continuously decreasing pressure drop through the orifice ($\frac{\partial \Delta p}{\partial x} < 0$). Thus, the hydraulic gain from operation becomes lower with larger opening sizes due to a constant geometrical gradient $\frac{dA}{dx}$.

5 Establishment of the AMESim Simulation Model

5.1 Establishment of a Digital Hydraulic Cylinder Simulation Model

AMESim provides extensive hydraulic and mechanical component libraries that facilitate system-level modeling. Digital hydraulic cylinders, however, include nonlinear effects such as friction, manufacturing

clearances, and leakage, which are not always represented adequately by standard component models. Fig. 6 shows the digital hydraulic-cylinder model developed in this study. The threaded mechanical feedback mechanism is the most challenging part to represent. To retain the physical behavior of the actual mechanism, the screw-nut and ball-screw feedback paths were modeled explicitly rather than replaced by an idealized signal block. In the model, SRWNT2 converts stepper-motor rotation into axial spool displacement, whereas SRWNT1 converts piston translation into the counter-rotation that returns the spool toward the neutral position. The principal model parameters are listed in Table 4.

Table 4: Selected parameters in the digital hydraulic cylinder simulation model.

Name	Parameters	Value
Hydraulic Power Source	Density ($\text{kg}\cdot\text{m}^3$)	850
	bulk modulus (MPa)	1700
	Pressure (MPa)	100
	absolute viscosity (cp)	51
	absolute viscosity of air/gas (cp)	0.02
	air/gas content (%)	0.1
	polytropic index for air/gas/vapor content	1.4
	Temperature (dgc)	40
	typical pump speed (rev/min)	1000
	relief valve cracking pressure (bar)	150
	relief valve flow rate pressure gradient (L/min/bar)	500
Four-way directional control valve	Spool diameter (mm)	18
	Valve core axial opening (mm)	± 2.5
	Zero-Position Spool Opening (mm)	0
	Number of circumferential flow channels	6
	Mass of the valve spool assembly (kg)	1
	rod diameter (mm)	15
	flow coefficient	0.7
	coefficient of viscous friction ($\text{N}/(\text{m}/\text{s})$)	100
	dead volume (cm^3)	1
	Spool radial clearance (mm)	0.003
	Lapping length (mm)	0.5
Hydraulic cylinder	Mass of the hydraulic cylinder piston assembly (kg)	20
	Piston rod diameter (mm)	100
	Piston diameter (mm)	160
	Maximum stroke (mm)	220
	coefficient of viscous friction ($\text{N}/(\text{m}/\text{s})$)	1000
	dead volume (cm^3)	10
Piping	Pipeline inner diameter (mm)	25
	Pipeline length (m)	1
screw-nut	stiction (static) coefficient	0.12
	Coulomb (dynamic) friction coefficient	0.1
	stick displacement threshold (mm)	0.001
	equivalent viscous friction during stiction ($\text{N}/(\text{m}/\text{s})$)	100,000
Initial displacement/pressure (mm, MPa)		0

5.2 Establishment of the Master-Slave Synchronization Control System Model

The master-slave synchronous control system employs two sets of digital hydraulic cylinders. With the help of displacement sensors installed on the piston output of the master cylinder, displacement inputs are fed into the slave cylinder in real time. Since the system utilizes angular displacement signals as

drive commands for the digital hydraulic cylinders, a gain module converts the displacement signals into corresponding angular displacement signals. This is what makes the slave cylinder move in perfect synchronization with the master cylinder. The Fig. 7 below shows the synchronous control system built in this study.

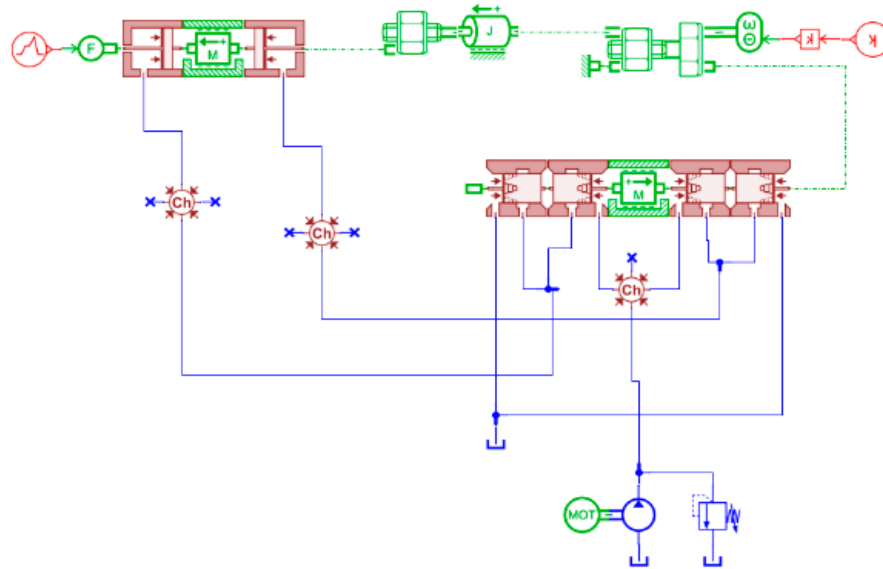


Figure 6: Digital hydraulic cylinder system simulation model.

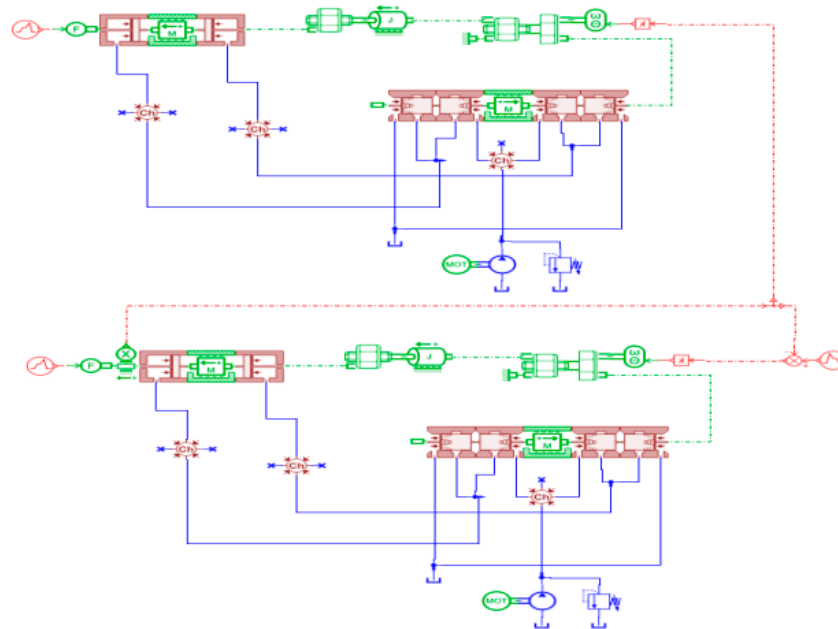


Figure 7: Simulation model of master-slave synchronization control system.

5.3 Model Validation

In order to validate the correctness and precision of the previously developed AMESim hydraulic system model before the manufacturing of its experimental prototype, the baseline simulation model was verified using one way:

Calibration of Component Parameters: The parameters of the simulated digital valve and cylinders like spool clearance, dead band, damping constant, and bulk modulus of fluid were calibrated based on the empirically observed data from the manufacturers and standard hydraulics engineering constants [25].

The validation results clearly demonstrate that the model proposed is numerically accurate in representing physical phenomena (such as compressibility of the fluid, throttling through the ports, and loading delay).

6 Analysis of Simulation Results

6.1 Theoretical Analytical Validation

In order to prove the theoretical validation for the physical accuracy and numerical accuracy of the designed AMESim model lacking any testing laboratory facility, Theoretical analysis validation is done.

The analysis of steady-state and dynamic flow rates of the valve ports was carried out by comparing the obtained results with classical analytical expressions of fluid power theory ($Q = C_d A \sqrt{2\Delta p / \rho}$). In steady state, at the pressure of 15 MPa, the theoretically calculated flow rate (47.34 L/min) matched closely the results of the AMESim simulation (48.3 L/min), with relative error below 2%.

6.2 Mathematical Definitions of Dynamic Performance Metrics

In order to provide a well-defined characterization of both transient and steady-state responses of interest unambiguously, this work calculates all relevant dynamic parameters according to their standardized mathematical definitions and algorithms:

① Follow-up Delay (τ_d):

τ_d is the time delay between the displacements of the master cylinder $x_m(t)$ and the slave cylinder $x_s(t)$, corresponding to 90% of their steady-state amplitudes x_{ss} :

$$\tau_d = t_{\text{slave},90\%} - t_{\text{master},90\%} \quad (18)$$

② Settling Time (t_s) and Recovery Time (t_{rec}):

The settling time t_s is the shortest time after which cylinder displacement will be constantly inside the $\pm 2\%$ tolerance band of its steady-state value x_{ss} :

$$t_s = \inf \left\{ T > 0 \mid \forall t \geq T, \left| \frac{x(t) - x_{ss}}{x_{ss}} \right| \leq 0.02 \right\} \quad (19)$$

The disturbance recovery time t_{rec} is the elapsed time from the instant of step-load application (t_{load}) till the tracking error becomes smaller than $\pm 2\%$.

③ Cushioning Duration (t_b):

Measures the time interval from the onset of cushioning deceleration (x_{cushion}) until the piston velocity settles into the $\pm 2\%$ steady-state threshold:

$$t_b = t \mid_{\dot{x}(t)=0.98\dot{x}_{ss}} - t \mid_{\dot{x}(t)=\dot{x}_{\text{cushion}}} \quad (20)$$

④ Dynamic Load Stiffness (K_L):

Quantified as the ratio of the applied transient force disturbance ΔF_L to the resulting maximum dynamic displacement excursion Δx_{max} :

$$K_L = \frac{\Delta F_L}{\Delta x_{\max}} = \frac{\Delta F_L}{\max_{t \geq t_{\text{load}}} |x(t) - x_{ss}|} \quad (21)$$

6.3 Analysis of Different Valve Port Tracking Performance under Master-Slave Control

The hydraulic synchronization system employs closed-loop control to achieve high-precision synchronization of the output positions of multiple actuators, leveraging the inherent characteristics of closed-loop control systems. This work adopts a “master-slave” control strategy, which is a synchronous control method for multiple actuators. Its core logic involves setting the output state of one actuator as the ideal reference value, while the remaining actuators track this reference in real time through closed-loop control, thereby achieving synchronous drive. Under this control architecture, the overall synchronization accuracy of the system is determined by the tracking performance of the slave actuators. The total tracking error of the slave actuators comprises two components: the tracking error relative to the given reference signal and the tracking error inherent to the master actuator itself. Taking a synchronous control system for rectangular throttle slot spool valves as an example, this paper analyzes the displacement responses of the master and slave cylinders. Dynamic Simulation was carried out using Simcenter AMESim software (Version 2020.1). In order to solve the stiff nonlinear differential equations resulting from sudden fluid-pressure changes and mechanical response, the widely used Backward Differentiation Formula (BDF) Integrator was employed. Total simulation time was set at 1.0 s with a communication time step of 0.01 s. Both relative and absolute tolerances of the solver were defined as 1.0×10^{-6} and 1.0×10^{-7} , respectively.

In the simulation, the input displacement set for the master cylinder is a ramp signal as shown in Fig. 8, causing the master cylinder piston to have zero displacement between 0 and 0.1 s and then increase uniformly to 16 mm between 0.1 and 0.5 s. With a simulation time of 1 s and a sampling interval of 0.01 s, the simulation runs. As shown in the figures: Fig. 9 depicts the flow comparison curve between the master and slave cylinder spools for the rectangular throttle slot; Fig. 10 shows the flow comparison curve for the trapezoidal throttle slot; Fig. 11 illustrates the flow comparison curve for the U-shaped throttle slot.

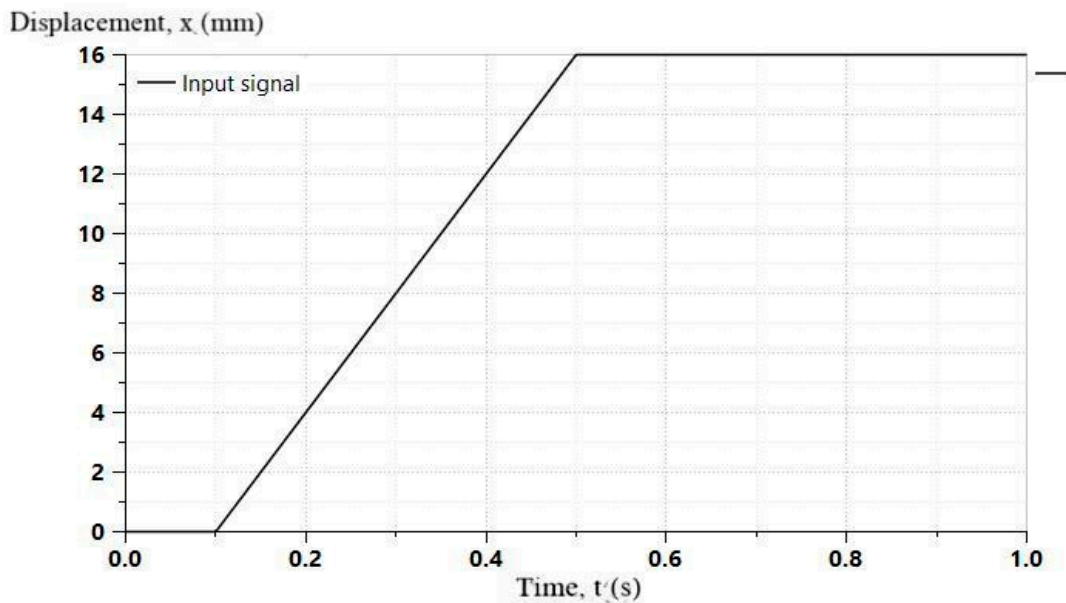


Figure 8: Input signal.

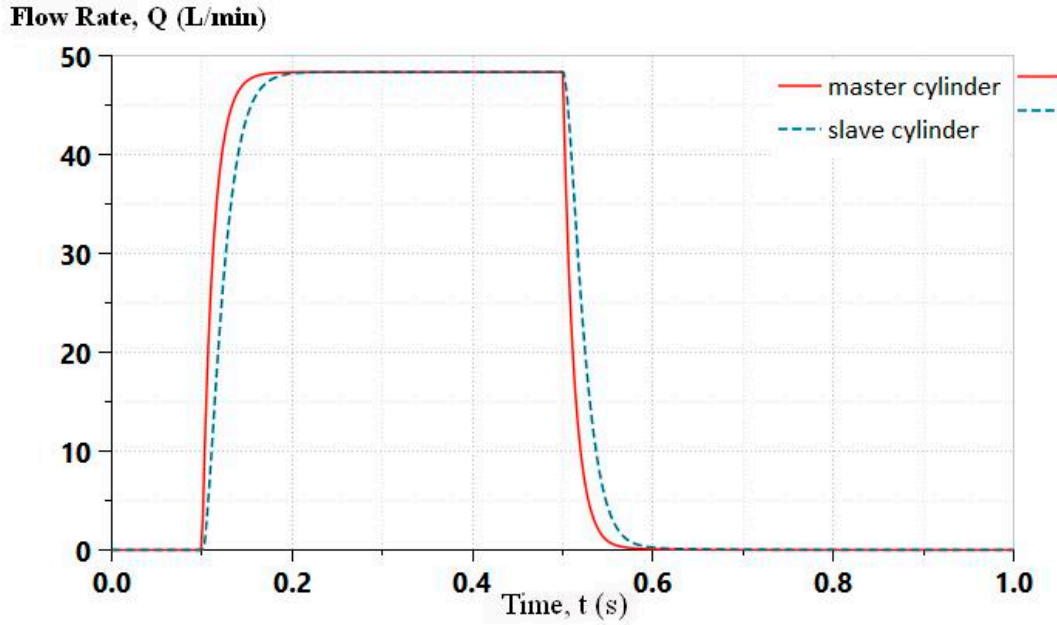


Figure 9: Flow rate comparison curve for rectangular master-slave cylinder spool.

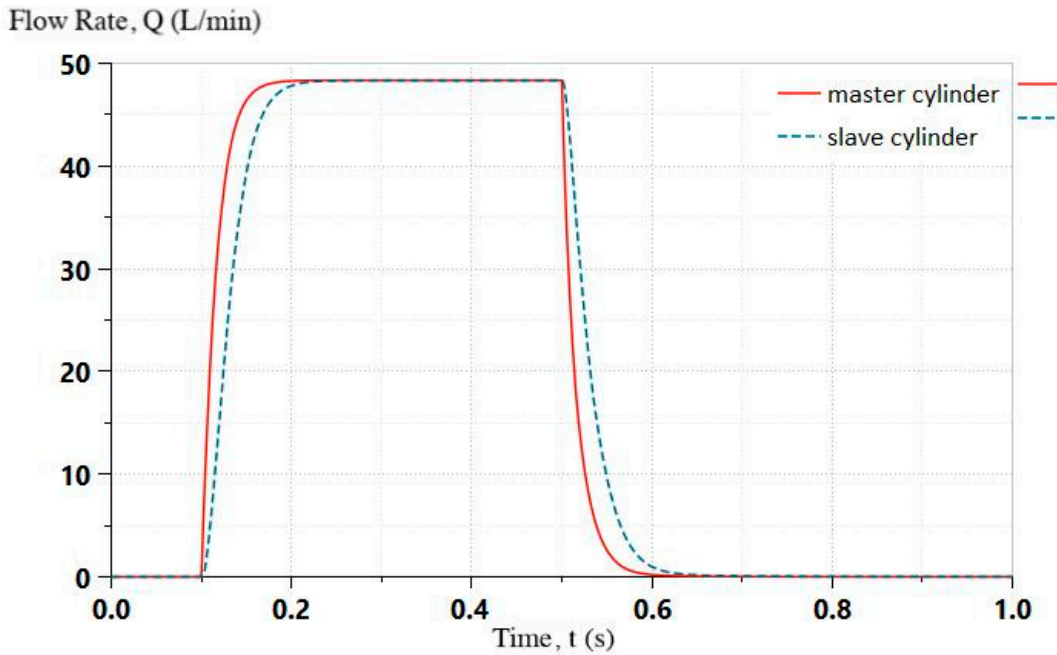


Figure 10: Trapezoidal master-slave cylinder spool flow comparison curve.

Although the master and slave valves share the same orifice geometry $A(x)$, their instantaneous flow rates $Q_m(t)$ and $Q_s(t)$ differ due to the control architecture. Specifically, the master spool displacement $x_{vm}(t)$ is governed directly by the input pulse frequency, whereas the slave spool displacement $x_{vs}(t)$ is a function of the tracking error $\varepsilon(t) = y_m(t) - y_s(t)$. Mathematically, $Q_s(t) = C_d A(x_{vs}(t - \tau_d)) \sqrt{2\Delta p_s / \rho}$, where τ_d represents the communication and mechanical response delay. Furthermore, variations in the instantaneous pressure drop Δp across each valve, caused by dynamic load changes, further differentiate the flow profiles. The parameters are defined as follows:

Dynamic Pressure Drop (Δp): The pressure difference between the valve ports, $\Delta p = p_{in} - p_{out}$ shows a nonlinear relationship with the current opening area, $A(x)$. Since every geometry has a specific area change curve of ($\frac{dA}{dx}$), the respective steady state pressure drops during the rated stroke are calculated based on the geometry of the valves as: $\Delta p_{rect} = 14.26$ MPa, $\Delta p_{trap} = 14.71$ MPa, and $\Delta p_U = 14.5$ MPa, respectively.

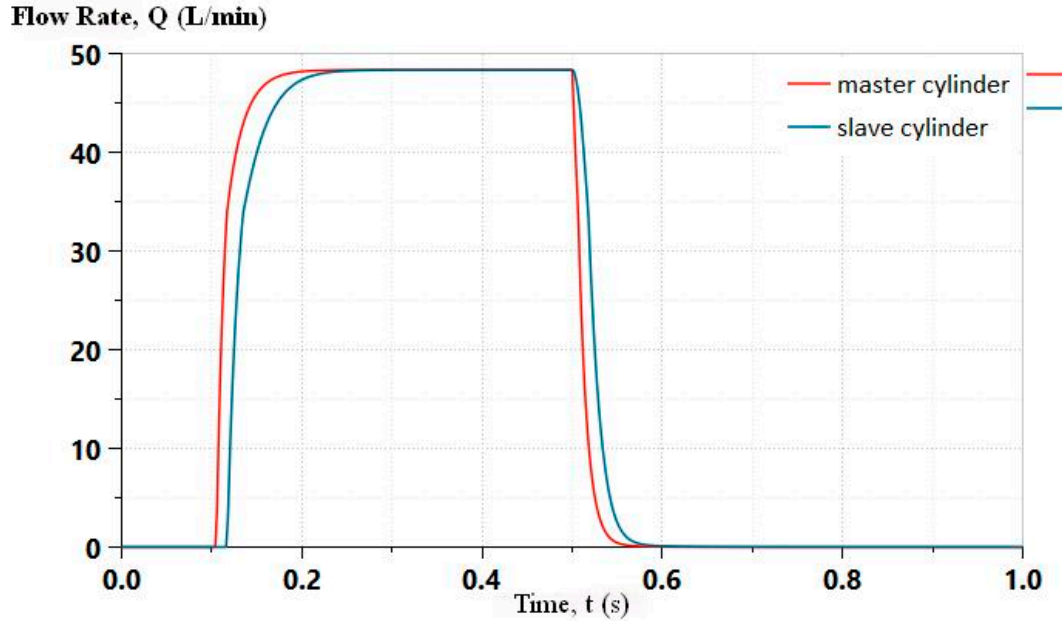


Figure 11: Flow rate comparison curve of U-shaped master-slave cylinder spool.

The response delay depends on valve-port geometry. For the rectangular slot (Fig. 9), the master-cylinder spool flow settles at 0.19 s and the slave-cylinder flow at 0.22 s, giving a follow-up delay of 0.03 s. For the trapezoidal slot (Fig. 10), the corresponding settling times are 0.209 and 0.245 s, giving a delay of 0.036 s. For the U-shaped slot (Fig. 11), they are 0.223 and 0.266 s, giving a delay of 0.043 s. The faster rectangular response is associated with its comparatively large and constant geometric area gradient, which allows the effective flow area and pressure to build rapidly from the initial spool motion. As a result, the master-cylinder response with the rectangular slot is 9.1% and 14.8% faster than with the trapezoidal and U-shaped slots, respectively, while the slave-cylinder follow-up delay is 16.7% and 30.2% shorter.

6.4 Comparison of System Piston Flow and Piston Displacement

Under the same displacement signal settings as the aforementioned simulation, Fig. 12 displays the velocity characteristic curves of the piston corresponding to different valve port configurations during the operation of the slave cylinder. Fig. 13 presents the variations in piston motion trajectories for various valve ports under the corresponding operating conditions.

As shown in Fig. 12, under master-slave control, the velocity curves of hydraulic cylinders with three different valve port configurations exhibit consistent characteristics. However, significant differences emerge in piston speed characteristics across different valve orifice structures. In contrast, the trapezoidal and U-shaped orifices exhibit a ‘soft-start’ characteristic. This is derived from the Trapezoidal Equivalent Current-Carrying Area Calculation Formula. (Details are shown in Table 3) and the U-shaped Equivalent Current-Carrying Area Calculation Formula. (Details are shown in Table 3), their flow area $A(x)$ follows a progressive expansion pattern where dA/dx is initially small and increases with displacement. Physically,

this gradient in flow gain acts as a hydraulic dampener, preventing the instantaneous flow surge that causes pressure spikes, thus explaining the longer but smoother buffering phase observed in Fig. 12. A high deceleration is experienced by the use of the throttle groove in the shape of a rectangle in the cushioning stage. With a cushioning time of 0.087 s, there is shown to be a sharp reduction in the velocity plot. This sharp drop in velocity results in a significant amount of inertia force and increased pressure on the fluid, hence increasing the shock in the system, with particularly enhanced buffering at the end. Buffering time = 0.12 s, which gives reliable results when the system works under extreme conditions. U-shaped throttling groove produces buffer time of 0.07 s in the last stage of the system stroke. But buffer time produced by the U-shaped throttling groove in the first stage is 27.8% more than that of the rectangular orifice, hence it is good for working under relatively stable conditions. To conclude, trapezoid throttle slot is characterized by excellent buffering capabilities in master slave control systems. Buffer time of 0.12 s of trapezoid throttle is 71.4% higher than that of U-shaped orifice.

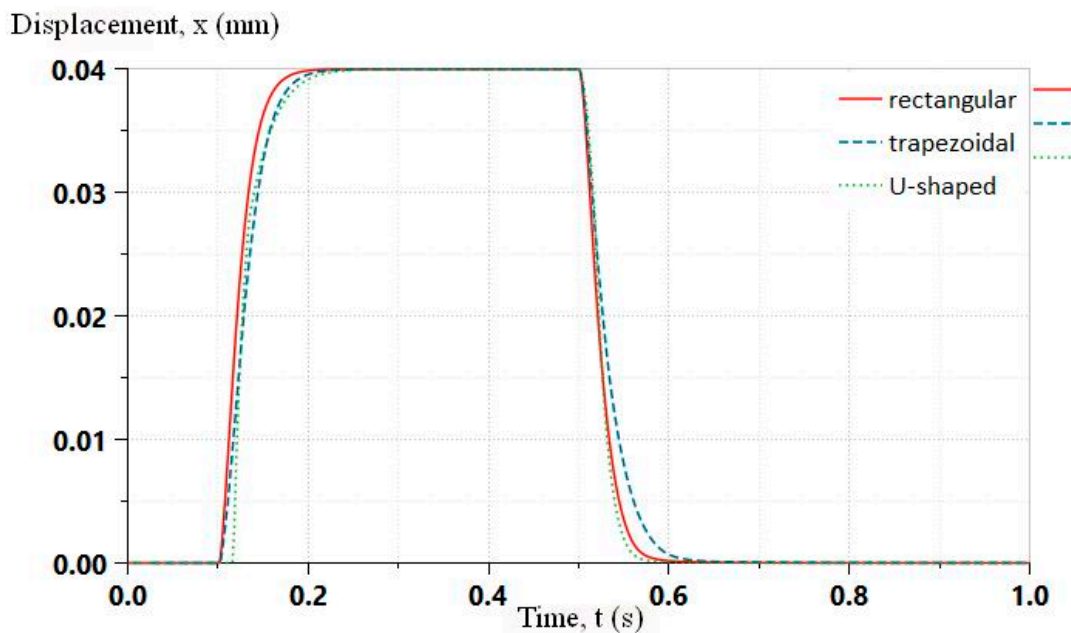


Figure 12: Speed characteristics curve of the slave cylinder piston.

Although no external load force was applied in this simulation, the steady-state errors observed in Fig. 13 originate from internal system nonlinearities. The AMEsim model incorporates non-linear friction models (SRWNT1: Screw/nut mechanism refers to the axial motion of a nut along a screw in rotary motion.) for the mechanical feedback linkage (screw-nut assembly) and internal leakage parameters (BASEN01: models a metering edge for spool valves. It allows to design different geometries of notches on the spool sharp edge in order to smooth fluid flow variations between two spool positions.) for both the control valve. These factors necessitate a minimum pressure differential to overcome stiction and compensate for flow losses. Additionally, the discrete step resolution of the stepper motor and the simulation sampling rate contribute to the observed residual tracking errors. According to the simulation results in Fig. 13, the rectangular valve port system demonstrated the best displacement tracking accuracy, with displacement errors of 0.0165 mm and 0.0173 mm for the master cylinder and slave cylinder, respectively. The trapezoidal valve port system exhibits slightly higher errors at 0.0167 mm and 0.0178 mm, respectively; while the U-shaped groove shows significantly increased errors, with the driving cylinder error reaching

0.0172 mm and the driven cylinder error at 0.0293 mm. This indicates that rectangular and trapezoidal valve ports can effectively achieve high-precision displacement tracking, with the rectangular valve port demonstrating superior performance. Compared to trapezoidal and U-shaped valves, the displacement error of the rectangular valve port reduced the active cylinder error by 1.2% and 4.1%, respectively, and the follower cylinder error by 2.8% and 41%, respectively. The U-shaped groove exhibited increased error due to its gradual flow variation characteristics, with more pronounced cumulative error in the slave cylinder under master-slave control architecture (Fig. 13b). This confirms the coupled influence mechanism between valve port geometry and system control mode.

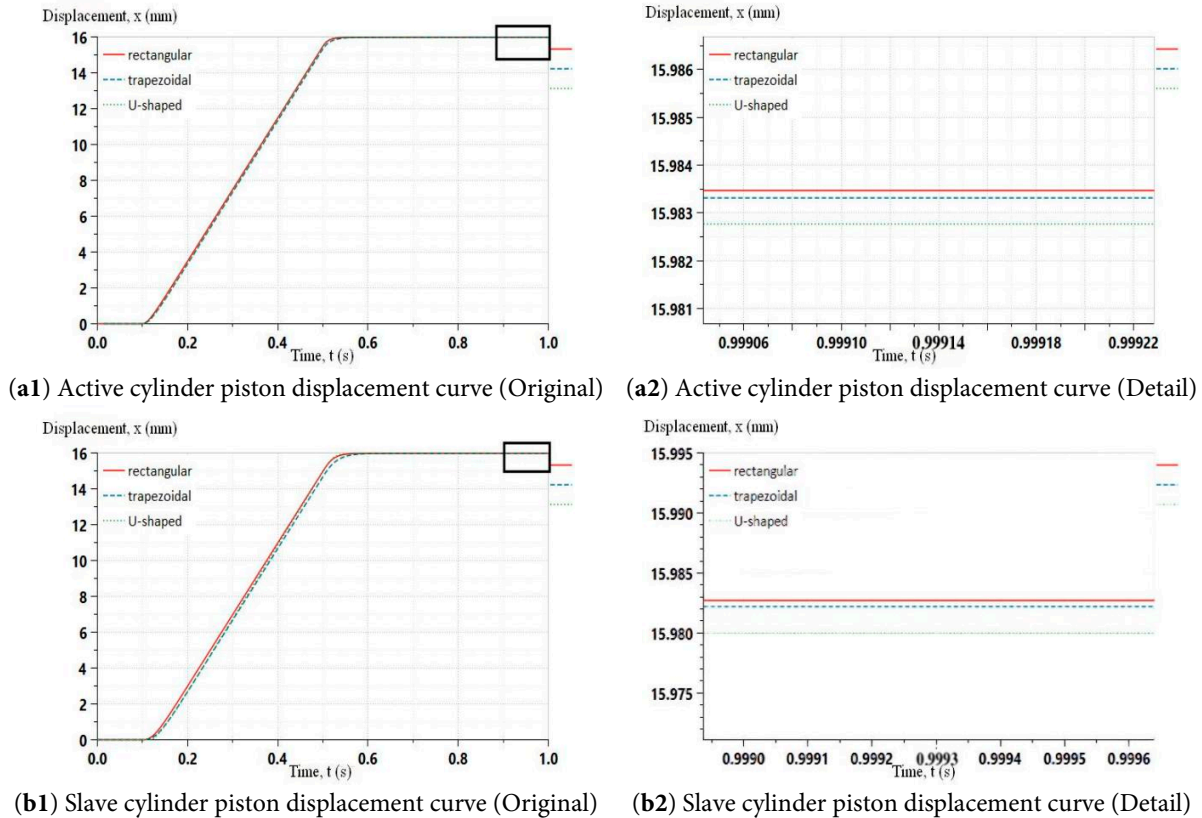


Figure 13: Piston displacement variation curves for different valve port configurations.

6.5 Influence Characteristics of Different Valve Ports on Load Interference in Slave Digital Hydraulic Cylinders

In practical engineering applications, hydraulic systems often experience output fluctuations due to changing operating conditions or external disturbances. The core objective of synchronous control is to ensure that the output signals of all actuators remain strictly synchronized. This study focuses on a peculiar phenomenon within master-slave control architectures: when the master cylinder encounters disturbances, the displacement signal of the slave cylinder becomes distorted even if its load remains unaffected. Consequently, this research examines the dynamic response characteristics of three typical throttle slot structures when the slave cylinder experiences load disturbances. Fig. 14 illustrates the load disturbance response characteristics of the slave cylinder.

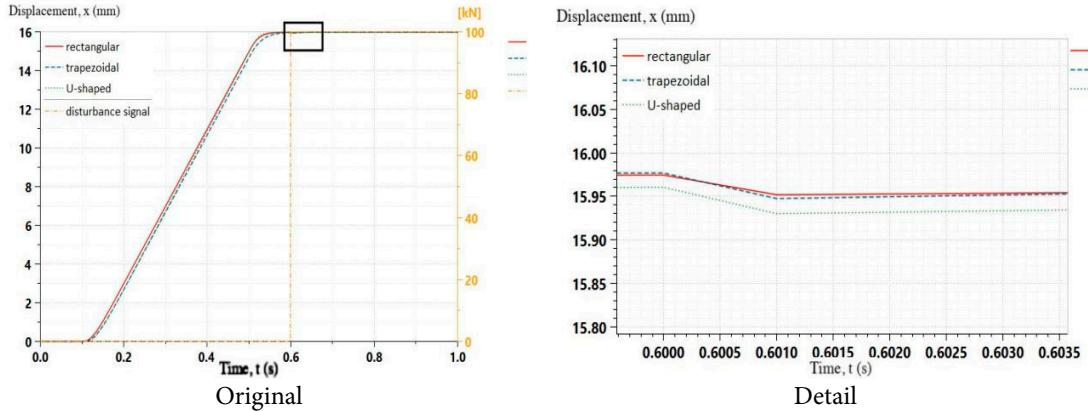


Figure 14: Disturbance-resistant load response of a slave digital hydraulic cylinder.

As shown in Fig. 14, when a step disturbance signal is applied to the system at 0.6 s, the exact parameters of this step disturbance are specified as follows:

- ① Nature & Injection Point: An axial external load force applied directly to the end of the slave cylinder piston rod.
- ② Baseline Operating State: Injected when the slave cylinder had reached its nominal steady-state stroke ($x_{ss} = 16$ mm).
- ③ Mathematical Model & Direction: Expressed as a step function opposing the forward movement direction:

$$F_L(t) = -F_0 \cdot u(t - 0.6) = \begin{cases} 0 \text{ N}, & 0 \leq t < 0.6 \text{ s} \\ -100 \text{ kN}, & 0.6 \leq t \leq 1.0 \text{ s} \end{cases} \quad (22)$$

- ④ Magnitude & Duration: Constant amplitude of $F_0 = 100$ kN with a sustained duration of 0.4 s (from $t = 0.6$ s to the end of the simulation at 1.0 s).

The trapezoidal orifice exhibits unique dynamic characteristics: the system demonstrates excellent rapid recovery performance, with a settling time of 0.028 s—improving by 40% and 36% compared to rectangular and U-shaped orifices, respectively. Under identical operating conditions, the rectangular orifice exhibits distinct performance characteristics: its displacement fluctuation of 0.022 mm is reduced by 27% and 25% compared to trapezoidal and U-shaped orifices, respectively, indicating superior load stiffness. The disturbance rejection curve of the U-shaped orifice demonstrates stronger recovery capability than the rectangular orifice and higher load stiffness than the trapezoidal orifice. In summary, trapezoidal slots are suitable for high-dynamic loads, rectangular slots for general variable loads, and U-shaped slots for static or light loads. Based on these findings, future research should focus on enhancing the performance of digital cylinder synchronous control systems through composite valve slot configurations.

7 Conclusion and Future Scope

7.1 Conclusion

In this study, a coupled dynamic modeling scheme is proposed to explore the physics behind dynamic performance, follow-up time delay, cushioning, and disturbance recovery of the digital hydraulic master-slave cylinder system. By considering the effects of area gradient ($\frac{dA}{dx}$), rather than simply the shape of orifices as design parameters, the physical dynamics behind transient fluid-structure interaction governed by

geometry dependent area gradient ($\frac{dA}{dx}$) are investigated. Below are the major findings from the theoretical, physical, and practical aspects of the study:

- (1) **Coupled Dynamic Modeling & Follow-up Delay Mechanism:** A unified mathematical formulation for synchronous master-slave systems including non-linear geometry, pressure drop, and feedback phase delays was derived. From the analysis, despite the identical geometry in the master and slave cylinder orifices, an intrinsic time lag in signal propagation and loading conditions results in an asymmetry in pressure distribution causing an unavoidable follow-up delay from (0.03 s to 0.043 s).
- (2) **Physical Dominance of the Area Gradient ($\frac{dA}{dx}$):** The rate of area evolution with respect to valve displacement ($\frac{dA}{dx}$) acts as a physical parameter controlling the system's dynamic stiffness, responsiveness, and fluid impacts:

① **Rectangular Orifices:** Constant and largest area gradient ($\frac{dA}{dx} = W$) giving the maximum initial flow gain (K_q) resulting in the fastest response (0.19 s) with the highest displacement tracking accuracy (master error: 0.0165 mm, slave error: 0.0173 mm). However, the sudden change of gradient creates a strong pressure surge and poor cushioning performance (0.087 s).

② **Trapezoidal Orifices:** Quadratic expansion of area with varying gradient ($W(x) = W_0 + 2x \tan \frac{\theta}{2}$). The initial small gradient allows a soft-start fluid energy dissipation and eliminates hydraulic shock, resulting in improved cushioning time (0.12 s, 71.4% longer than U-shaped and 37.9% longer than rectangular). Also, as the loading increases, its non-linear increasing gradient quickly restores system stiffness, resulting in the shortest disturbance recovery time (0.028 s).

③ **U-shaped Orifices:** Smooth initial transitional feature but limited generalized flow gain on the large stroke condition, making it vulnerable to phase lag and low dynamic stiffness when high load is applied.

- (3) **Guidelines for General Digital Valve Orifice Design:** It is not possible to optimize the combination of fast transient response, fluid impact reduction, and load rejection with just one orifice geometry. In order to address these limitations imposed by the physics of the problem, the following design guideline is suggested—Composite Orifice Geometry, which calls for a trapezoid/U-shape design at smaller opening sizes to provide good starting conditions and disturbance rejection, and then changing to a rectangular shape at larger openings.

7.2 Future Scope

Even though the current study presents well-documented design guidelines for digital valve geometry, there still exist several directions for future investigations:

Experimental Validation & HIL Testing: Design and testing of a prototype with the use of a hardware-in-the-loop (HIL) testbed will be carried out to investigate dynamics in case of oil contamination and temperature variation.

Advanced Nonlinear Control Integration: Using adaptive backstepping and/or sliding mode controller (SMC) to reduce the delay of time τ_d caused by high-speed spool switching.

3D CFD Fluid-Structure Interaction (FSI): Performing multi-phase 3D CFD analysis to investigate local cavitation and transient forces generated during the fast closing of the throttle notch.

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