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THE CUSHIONING GROOVE FOR SOLENOID SWITCHING VALVES – CONCEPT AND THEORETICAL ANALYSYS

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ABSTRACT

A proper cushioning of the spool and armature of a switching valve at the end stops is important for a high lifetime and a low impact sound. Squeeze gap concepts are often employed. Oil between two basically parallel surfaces, which come into contact and which form the mechanical end stop, provides an excellent damping mechanism. But this oil film may exhibit high stiction forces, which cause a high response delay or even a total failure of the valve, unless cavitation in the oil film limits the stiction force. A new cushioning concept based on a notched groove is presented in this paper. Its basic functional principle is studied by compact mathematical models. The basic differences to squeeze gap cushioning are worked out and basic dimensioning rules are derived. This concept overcomes the stiction force problem fully.

KEY WORDS

Switching valve, Fluid stiction, Cushioning system

NOMENCLATURE

A_g	:	Notch flow area
b	:	Radial extension of groove
F_C	:	Cushioning/stiction force of cushioning groove
F_S	:	Stiction force of annular squeeze gap
d	:	Diameter of groove
e	:	Base of natural logarithm
F_{Meff}	:	Effective actuation force at switch on, i.e., magnet force minus spring force
F_{Spr}	:	Spring force; is the effective actuation force at switch off
H	:	Nondimensional gap height
H_0	:	Initial nondimensional gap height
H_1	:	Final value of H attained in the valve's switched off rest position immediately before valve switch on
H_2	:	Final value of H attained in stiction after valve switch on time equals switch off time in rest position
h	:	Gap height
h_0	:	Initial gap height
m	:	Moved mass
p	:	Pressure in cushioning groove
p_a	:	Pressure in ambience of the cushioning gap or groove
p_{min}	:	Minimal pressure in squeeze gap

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Q_1, Q_2	:	Flow rates at outer and inner groove edges
Q_g	:	Flow rate over notches
Q_{tot}	:	Total flow out of groove
R_i, R_o	:	inner and outer radius of squeeze gap
t	:	Physical time
T	:	Reference time
v, v_{End}	:	Nondimensional speed, its value at landing
α	:	Nondimensional squeeze gap system parameter
α_C	:	Nondimensional cushioning groove system parameter
α_E, α_g	:	Flow coefficients of edges and notches
β_C	:	Nondimensional cushioning groove system parameter
δ	:	Slenderness factor of annular gap
η	:	Viscosity of fluid/oil
ρ	:	Fluid (oil) density
τ	:	Nondimensional time

INTRODUCTION

Solenoids counteracted by an elastic spring are the dominating actuation technology of today's hydraulic switching valves. In the context of digital hydraulics, in particular of hydraulic switching control, high lifetime of the solenoids means at least a billion operations without a significant performance loss. Such valves switch within a millisecond or faster and this leads to a high speed of the valve spool or poppet and, in turn, to high impact speed at the end stops. In order to keep this speed in tolerable limits some cushioning is necessary. A powerful means are squeeze gaps, since the oil film provides an excellent damping behavior. But such gaps always cause some oil stiction force [1], which might deteriorate a proper functioning when the valve is switched into the opposite state. The fundamental problem of such squeeze gaps becomes evident by the following, simple mathematical model of an annular, parallel squeeze gap. A pot core type valve, as shown in Figure 1, is used as example.

It is assumed that the flushing channels provide sufficient flow to avoid noticeable stiction in the coil domain. Stiction is dominated by gaps of the iron core poles and at the opposite side by the landing ring. Oil stiction or squeeze film damping in an annular domain with inner and outer radii R_i, R_o, actual gap height h, fluid viscosity η, and surrounding pressure p_0, is described by the following set of equations (see [1]).

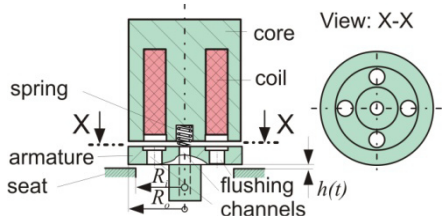


Figure 1 Pot core solenoid with flat armature plate; oil stiction can occur between iron core and armature and the lower seat area.

$$F_S = 2\pi\,\eta\,R_i^4\delta^3\frac{\dot h}{h^3}\left(1+\frac{\delta}{2}+O\left(\delta^2\right)\right)$$
$$p_{\min} = p_a - \frac{3\dot h\,R_i^{\,2}\eta}{2h^3}\delta^2 + O\left(\delta^3\right);\ \ \delta = \frac{R_o - R_i}{R_i} \tag{1}$$

F_S is the stiction ($\dot h > 0$) or damping ($\dot h < 0$) force, p_{min} the lowest pressure in the gap, which has to stay above vapor pressure to avoid cavitation. δ is the gap slenderness. Since this is just an introductory example, which should create awareness for the fundamental problem of stiction with parallel gaps, the case landing at the lower end stop, when the valve is switched off, and the subsequent lift off after some idle time are considered. The equation of motion of the spool

$$m\ddot h = -F_S + \begin{cases} -F_{Spr} & \text{switch off} \\ F_{Meff} & \text{switch on} \end{cases} \tag{2}$$

in combination with (1) and the following scales

$$h = h_0 H(\tau);\ \ t = \tau T;\ \ T = \sqrt{2\,m\,h_0 / F_{Spr,Meff}}$$
$$\alpha^3 = \frac{\pi\eta R_i^{\,4}\delta^3\sqrt{2}(2+\delta)}{h_0^{\,5/2}\sqrt{m\,F_{Spr,Mef}}};\ \ \frac{d}{d\tau} = (\,)' \tag{3}$$

brings the problem into the following dimensionless form.

$$H'' = -\alpha^3\frac{H'}{H^3} + \begin{cases} -2 & \text{switch off} \\ +2 & \text{switch on} \end{cases} \tag{4}$$

F_{Spr} is the constant spring force and F_{Mef} the magnetic force minus the spring force. h_0 is chosen to be the spool stroke. Figure 2 shows the evolution of the dimensionless gap height and gap speed for a specific value of α. For the data of a valve according to Table 1 it computes to $\alpha = 0.036$. The cushioning process in the switch off case leads to a late braking of the motion. A contact between armature and seat does never happen but the spool approaches its final position only asymptoti-

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cally. In this final process and also initially, when the valve is switched on, inertia forces are negligible.

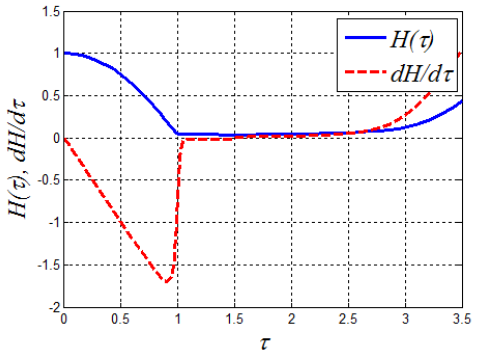


Figure 2 Solution of Equ. (2) for $\alpha = 0.2$; magnetic force is switched on at $\tau = 1.5$; initial stiction (at $\tau = 0$) is disregarded; solution is obtained numerically with Matlab routine ode23s.

Table 1 Data of spool, fluid, seat area, return spring.

m	30 g	F_{Spr}	25 N	η	0.02 Pa s
h_0	0.5 mm	R_o	12 mm	R_i	11.5 mm

In these cases the evolution is guided by the right hand side of (4) with the solutions

$$H(\tau) = H_0 \frac{\alpha^{3/2}}{\sqrt{\pm 4\tau H_0^{\,2} + \alpha^3}} \qquad (5)$$

H_0 is the initial value in these phases. The plus sign is valid for switch off, the minus for switch on. (4) shows that (5) is only valid, if $H = O(\alpha)$. A more profound asymptotic analysis tells that after cushioning the solution of the full problem (4) transits to (5) at values of order $H = O(\alpha^{3/2})$. Thus, even if the system does not stay long in the open position but is switched on short time after switch off, it takes an $O(1)$ nondimensional time until the strong stiction force is overcome. This means that the denominator term of (5) (minus sign is valid) becomes small to enable a substantial growth of $H(\tau)$. If the system stays in a rest position for longer time, exactly that time is needed in addition to leave the strong stiction force situation. This can be checked if the final gap H_1 for an initial value H_0 ($H_1 = H(H_0, \tau_1)$ for valve switch off cushioning (plus sign in (5)) is taken as initial value for the switch on (minus sign in (5)) which gives H_0, ($H_2 = H(H_1, \tau_1) = H_0$). Thus, liberation from stiction takes approximately the switching time plus the time at rest position. All this, of course, is only true if cavitation does not occur.

CUSHING GROOVE SYSTEM

Proposed solution concept

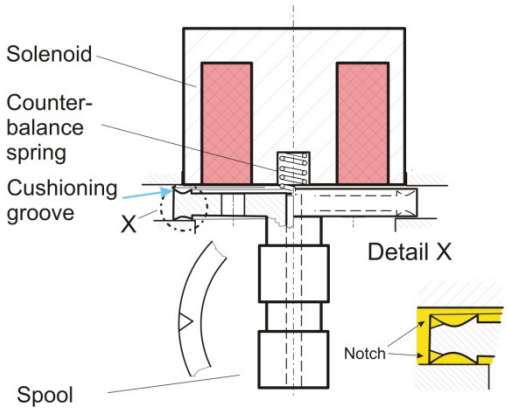


Figure 3 Solenoid type switching valve with solenoid, armature plate, counterbalance spring, valve spool, and cushioning grooves.

In order to overcome the fundamental problems of the squeeze gap cushioning system an alternative cushioning concept based on a cushioning groove (see Figure 3) has been developed. The groove is bounded by two edges and forms one side of two abutting surfaces normal to the moving direction of the spool or poppet. The gap between this groove and its opposing surface is filled with fluid which is pressed out when the valve is switched and comes close to the limiting contact. The fluid flow over the two edges causes a quadratic pressure loss leading to a higher pressure in the groove. The outflow resistance increases when the edges approach the opposing surface and generates in this way the cushioning effect. Finally, the edges might get into contact with the opposing surface and the fluid in the groove would be fully sealed from the environment. This would cause fluid stiction as well. To prevent this, a small orifice, for instance realized by some notches as shown in Figure 3, is connecting the groove with the environment.

A mathematical model of the cushioning groove system

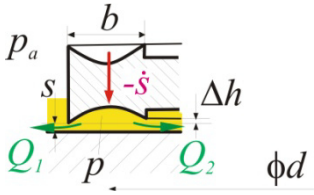


Figure 4 Cushioning groove: flow rates and dimensions.

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In order to assess this cushioning system's behavior and to compare it with the squeeze gap concept, an analog model is formed. The equation of motion (2) is combined with the following cushioning force model for F_C, which replaces F_S in (2); (see also Figure 4).

$$
\begin{aligned}
&Q_{tot} = Q_1 + Q_2 + Q_g = -\dot{h}\, b\, d\, \pi \\
&Q_{1,2} = \alpha_E (d \pm b) \pi\, h \sqrt[\wedge]{\frac{2(p - p_a)}{\rho}} \\
&Q_g = \alpha_g A_g \sqrt[\wedge]{\frac{2(p - p_a)}{\rho}} \\
&F_C = (p - p_a) b\, d\, \pi = \tfrac{1}{2} \frac{\rho \dot{h}^{\hat{2}}\, (b\, d\, \pi)^3}{(2\alpha_E \pi\, h\, d + \alpha_g A_g)^2} \\
&\sqrt[\wedge]{u} = \sqrt{|u|}\ sign(u); \qquad (u)^{\hat{2}} = u\,|u|
\end{aligned}
\tag{6}
$$

Using the following scales

$$
\begin{aligned}
&h = h_0 H(\tau);\ \ t = \tau T;\ \ T = \sqrt{2 m h_0 / F_{Spr,Meff}} \\
&\alpha_C{}^2 = \frac{\rho\, h_0{}^2\, (b\, d\, \pi)^3}{F_{Spr,Meff}\, (2\alpha_E \pi\, h_0\, d)^2};\ \ \beta_C = \frac{\alpha_g A_g}{2\alpha_E \pi\, h_0\, d}
\end{aligned}
\tag{7}
$$

a similar nondimensional equation for the motion of the system is obtained.

$$
H'' = -\frac{\alpha_C{}^2 (H')^{\hat{2}}}{(H + \beta_C)^2} \pm 2 \tag{8}
$$

This cushioning system has significant differences to the squeeze type system, which are listed in Table 2.

Table 2 Characteristics of the squeeze gap and groove cushioning systems.

Effect	Squeeze gap	Groove
Depends on viscosity η	Yes	No
Influence of speed $\dot{h}$	Linear	Quadratic
Landing speed	Zero	Finite
Lift off speed's dependency on history	Large	Zero
Influence of cavitation, hence of ambient pressure on lift off response	High	Marginal

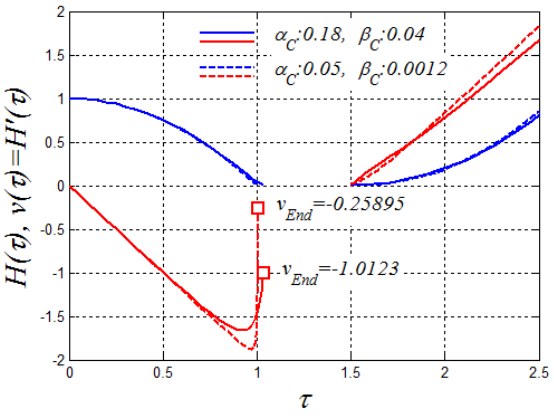


Figure 5 Trajectories of the solution of (8); solutions for $H(\tau)$ (blue lines) and $v(\tau)$ (red lines) are obtained by a numerical integration in Matlab using the solver ode23s.

Figure 5 shows the behavior of the system according to the state equation (8): It starts at $\tau = 0$ from the valve shut on state ($H = 1$) when the magnetic force is shut off and the spring brings the armature and spool back to the shut off position ($H = 0$). Position H and speed H' or v, respectively, of this phase are only shown till landing at the end stop. The landing speed v_{End} is a central cushioning system performance property. To avoid excessive wear the speed in physical scales must be typically in the order of 0.1 m/s. The nondimensional force in (8) in this valve shut off phase is -2. At $\tau = 1.5$, the valve is shut on again, a very fast lift off takes place, since stiction forces of this cushioning system are marginal.

Dimensioning rules for the cushioning system obtained from an asymptotic analysis

The cushioning system is intended to exhibit its damping force only during the late phases of the motion in order not to brake the motion too early and reduce the switching time significantly.
The solution of Equ. (8) has in part similar properties as Equ. (4). When the motion starts with zero speed ($H'=0$) at $H=1$, the cushioning term $\frac{-\alpha_C{}^2 (H')^{\hat{2}}}{(H + \beta_C)^2}$ does not play a significant role, since $\alpha_C{}^2$ is typically very small compared to one and the speed square H'^2 is also quite small. In first order, the evolution is guided by

$$
H'' = -2\,, \tag{9}
$$

which in combination with the initial conditions $H(0) = 1$, $H'(0) = 0$, has the following phase space graph representation

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$$H' = 2\sqrt{1-H}\ . \tag{10}$$

However, close to landing (at $H=0$) and as long as the speed is not too small the cushioning force dominates the motion. The solution is characterized by

$$H'' = -\frac{\alpha_C{}^2 (H')^{\hat{2}}}{(H+\beta_C)^2} \tag{11}$$

Equ. (11) can be brought into the following first order differential equation for the nondimensional velocity v as function of the nondimensional position H

$$\frac{d\,v(H)}{dH} v(H) = -\frac{\alpha_C{}^2 (v(H))^{\hat{2}}}{(H+\beta_C)^2}, \quad v(H) = H' \tag{12}$$

which can be easily integrated to obtain a phase space representation of the solution by the following graph.

$$v(H) = C \exp\left(-\frac{\alpha_C{}^2}{H+\beta_C}\right). \tag{13}$$

Equ. (8) is a singularly perturbed boundary equation in the parameter α and exhibits a sharp boundary layer in its speed trajectory when it comes to landing. This is clearly seen in Figure 5. The smaller α_C, the sharper is this layer. For $\alpha_C \to 0$ a strict mechanical impact occurs.

Approximate analytical solutions could be obtained in a systematic manner with the method of matched asymptotic expansions of the problem using α_C as perturbation parameter. See [2, 3] for the mathematical background of singular perturbation theory and the asymptotic expansion method. [4] employs such matched asymptotic expansions to derive a boundary layer theory for 3D viscid wave propagation in frequency domain. This, however is not done in this paper for the given space limitations. Instead, simple assessments of some technically important characteristics of the solutions of (8) are given.

The cushioning process is guided by the solution (13). However, the integration constant C is not determined yet. With C the landing speed $v_{End} = v(H=0) = C \exp(-\alpha_C{}^2/\beta_C)$ can be computed immediately. From Figure 5 and from the limit case $\alpha_C \to 0$ one can conclude that the transition from the solution of (9) to the solution of (13) takes place at speeds $v = 2+O(\alpha)$. Therefore, the following order relation is valid

$$C = 2 + O(\alpha_C)\ . \tag{14}$$

From this very rough estimate a correspondingly rough estimate for v_{End} is

$$v_{End} \approx 2\exp\left(-\frac{\alpha_C{}^2}{\beta_C}\right). \tag{15}$$

In Table 3 the estimates according to (15) are compared with the numerical results shown in Figure 5. For small values of α_C the agreement is good.

Table 3 v_{End} estimates according to (15) for the parameter values of Figure 5; for comparison the corresponding numerical results are listed too.

Parameter values		v_{End}	
α_C	β_C	accord. to (15)	numeri-cal
0.18	0.04	-0.89	-1.01
0.05	0.0012	-0.25	-0.26

A further technically relevant property of the solution is the maximum acceleration in the cushioning phase, since this acceleration corresponds to the maximum cushioning force. This force constitutes a peak load of the involved components and a short pulse excitation of the whole mechanical system with mechanical shock waves and impact sound.

The nondimensional acceleration is related to the phase space quantities as follows

$$H'' = v' = \frac{dv}{dH} v\ . \tag{16}$$

Evaluation of (16) with the help of (13) yields

$$H'' = \frac{C^2 \alpha_C{}^2}{(H+\beta_C)^2} \exp\left(-\frac{2\alpha_C{}^2}{H+\beta_C}\right). \tag{17}$$

The maximum value attains H'' at the point $H = \alpha_C{}^2 - \beta_C$. It is

$$H''_{\max} = \frac{C^2}{e^2 \alpha_C{}^2}; \quad \ln(e) = 1\,. \tag{18}$$

For C the estimate (14) can be used as a first guess. Doing so, and evaluating Equ. (18) for the two parameter combinations as of Figure 6 (which are identical to those of Figure 5) gives the results according to Table 4.

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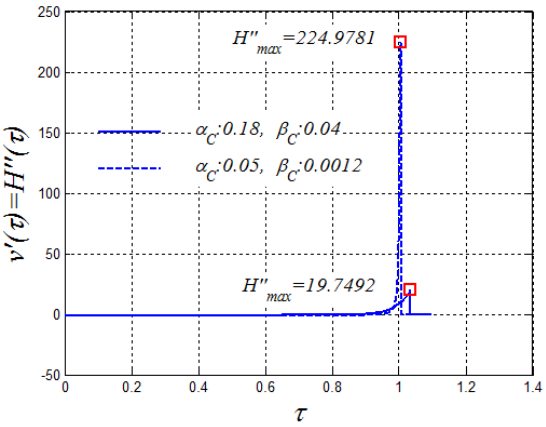


Figure 6 Nondimensional acceleration H'' of the solution of (8) obtained by a numerical integration in Matlab using the solver ode23s.

Table 4 H''_{max} estimates according to (15) for the parameter values of Figure 6 results and for comparison the corresponding numerical results.

Parameter values		H''_{max}	
α_C	β_C	accord. to (18)	numeri-cal
0.18	0.04	16.7	19.7
0.05	0.0012	216	225

With the results (15), (18) in combination with the estimate (14) and the scales (7), a dimensioning of the cushioning system can be made. A recommended sequence for determining α_C and β_C is to use the H''_{max} criterion (18) first to determine α_C and, second, the v_{End} criterion to fix β_C.
There is a third technically relevant criterion, namely a lift off speed criterion which can be obtained from a solution of the following modification of Equ. (8).

$$\frac{\alpha_C{}^2 (H')^{\hat{2}}}{(H+\beta_C)^2} = 2 \qquad (19)$$

which has the solution

$$H(\tau) = \beta_C \left(\exp\left(\frac{\sqrt{2}\tau}{\alpha_C} \right) - 1 \right) . \qquad (20)$$

Of course, this solution disregards inertia, but it is a good indicator of the lift off limitations of the cushioning system. Equ. (20) tells clearly that the cushioning groove does not have the lift off limitations of the squeeze gap concept.

CONCLUSIONS AND OUTLOOK

The cushioning groove concept overcomes a basic potential problem of squeeze gap cushioning systems of wet armature magnets in fast switching valves, namely the response delay due to oil stiction forces. Most of the valves in practice have in some way such a cushioning system. They often only work because of cavitation which limits the stiction force to such values which can be overcome by the valve actuators. But, this stiction force limitation effect of cavitation depends on a low ambient pressure in the cushioning area. If such low pressure cannot be guaranteed valve failure is very likely.

The cushioning groove system can provide good damping without significant stiction at any ambient pressure. Simple design rules have been derived in this paper on the basis of asymptotic considerations of a nondimensional state equation for the valve movement.

A first version of the cushioning groove concept, which employs an elevated groove edge instead of the notches, hast been experimentally tested and turned out to work as expected. The accurate manufacturing of the two edges with a small height difference, however, might pose high manufacturing quality demands, since the tight tolerances are required. Problems resulting from an out of tolerance groove edge heights are reported in [5].

The theoretical analysis of this paper is based on two simplifying assumptions, the validity of which will be investigated in the future:

1. Perfect parallel arrangement of cushioning groove edges and the contact surface: A perfect parallelism is impossible; therefore, at landing the groove edges will not be in uniform contact with their opposite contact plane but will have a gap which allow outflow of fluid in addition to the notches; notch area in relation to remaining gap area, elastic deformations, and impedances of the circumferential flow in the groove are expected to influence the system behavior in this final phase.
2. Validity of the orifice equation, as used in (6). It strongly determines the system properties as given by equation (8) and most simplifications derived from it. The findings presented in [5] indicate that this is not fully valid when the gap height h becomes very small (µm range), since a perfectly sharp edge is neither feasible nor desirable. Thus there is a further motion regime in that range, the influence of which is not yet mathematically modeled and understood. Even though the experimental findings did not show significant impact of this regime a thorough analysis

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can provide clarity about its role and rules for its control.

This cushioning groove concept can be used also for other applications than for switching valves, whenever a compact, simple and cheap system which can generate short end stop damping without stiction problems is required. Authors will investigate the application of this cushioning groove for fast check valves, where oil stiction has also been observed.

ACKNOWLEDGEMENT

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