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Flow Forces and the Tilting of Spring Loaded Valve Plates - Part I

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Part I

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ABSTRACT

Up to the present, so far as the author is aware, it has been considered selfevident that spring loaded valve plates remain parallel to the seat during valve lift when nominally symmetrical conditions of flow and spring force apply. In this paper it will be shown, that this in general is not the case. In the author's opinion the problem of instability in valve plate motion has not been studied because of insufficient knowledge of forces resulting from flow in valve channels. A complete theory for flow forces is complex but a simplified treatment makes clear the fundamentals of the phenomenon.

Forces acting on a valve plate during opening and closing are discussed. Flow forces resulting from deflection of the gas flow coupled with spring forces govern valve dynamics, except within small region near seat and guard. Flow forces increase considerably (by some 25%) with increasing lift. This is shown for the case of a simple slot with 90^{o} deflection of the flow by potential flow solution, which gives a close approximation to the real flow.

If increase in flow force with lift exceeds the increase in spring force, valve plate motion becomes unstable and degenerates to tilted motion. Conditions for stability are given in terms of valve parameters and discussed in detail.

INTRODUCTION

Seat parallel motion of the valve plate is very important for valve life time. In the opinion of the author, failure of valve plates is connected closely with tilted motion and consequent impact. A hypothesis of the cause of these failures is presented elsewhere in these Proceedings.

Before looking closer at stability we have to discuss the forces acting on a valve plate. These forces are flow forces and spring force, fig.1. Flow force on valve plate arises solely as a consequence of deflection in the gas flow, except small regions near seat and guard. We shall call this force the impulsive force F_i. Near the guard (when opening) there is an additional flow effect causing a "squeezing force" F_{squ}. This effect is normally important only for distances less than 0.2mm between plate and guard (in the absence of valve plate tilt) [1]. The squeezing force is especially important for high pressure compressors. It does not occur when steady state flow force measurements are performed.

When the valve plate is relatively near to the seat, reattachment of flow to seat wall occurs and causes pressure recovery and hence increases impulsive force F_i. According to [2] reattachment up to $y/(e-b) \approx 0.5$ is to be expected.

In computer calculations of valve dynamics a viscous damping force, proportional to plate velocity often is introduced. There is little physical basis in the flow process for postulating such a force. The above mentioned squeezing force becomes only important in the vicinity of the guard. Mechanical friction associated with guides or in the bending arms of the springs may cause some damping, the magnitude of which is difficult to estimate.

We may conclude that the impulsive force governs motion in the main part of valve lift together with the spring force.

THE IMPULSIVE FORCE

For a basic investigation of the stability phenomenon it is helpful to begin with a simple situation, accessible to theoretical treatment. We start with flow through a parallel entrance slot of infinite

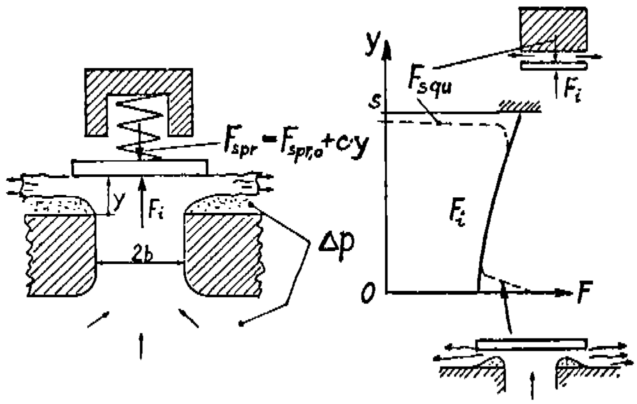


FIGURE 1 Forces acting an valve plate

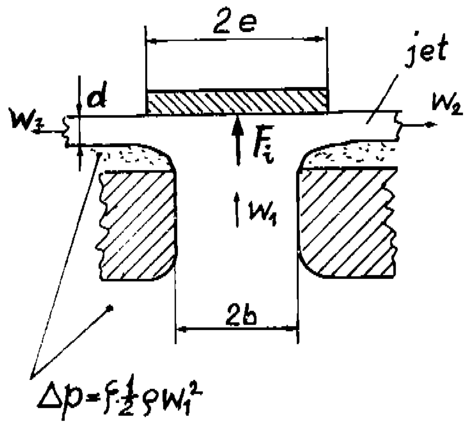


FIGURE 2 Jet from infinite slot

length, deflected by a valve plate normal to the slot, fig.2. The plate is assumed wide enough to ensure deflection of effectively 90° (this means e.g. $e \geqslant 1.5b$, which corresponds to real conditions). Quantities such as impulsive force, spring force, valve plate mass etc. are related to unit length of slot and given the suffix "1".

For this flow problem the theory of jets of an ideal fluid allows a very good approach to real fluid flow. Real flow has a separation line along the seat edge and forms a wake of approximately constant pressure, which corresponds to the boundary condition of ideal jet flow. The jet is concentrating from b to d. Kinetic energy of the leaving jet (velocity w_2) $\frac{1}{2}\rho w_2^2$ is lost. The pressure loss Δp (=pressure difference across the valve) is therefore

$$\Delta p = \frac{1}{2} \cdot \rho \cdot w_2^2 = \zeta \cdot \frac{1}{2} \cdot \rho \cdot w_1^2 \qquad (1)$$

From continuity:

$$w_2 = \frac{b}{d} \cdot w_1 \longrightarrow \boxed{\zeta = (b/d)^2} \qquad (2)$$

Frequently a quantity "flow area" is used instead of ζ to characterise losses. It is easily seen that flow area is 2d in our notation.

The concentration of the jet -and hence ζ- can be calculated from jet potential flow theory, see e.g. [3],[4]. Table 1 in appendix gives some numerical data. Detailed data on pressure distribution, jet boundary etc. are given in [1].

The momentum theorem than offers an easy way of calculating impulsive force on the valve plate. For a control volume as indicated in fig.3 we get for the y coordinate

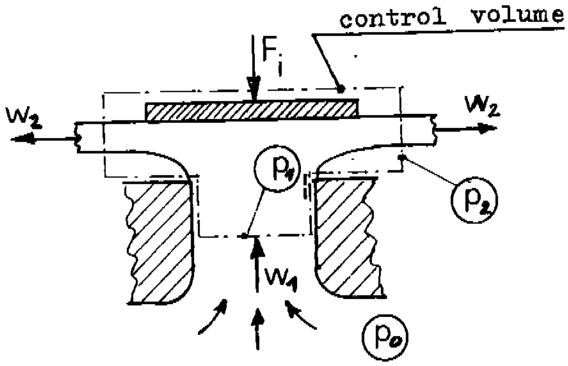


FIGURE 3 Control volume

putting $p_2=0$ for simplicity:

$$p_1 \cdot 2b - F_{i,1} = \dot{m}(w_{2y} - w_{1y}) \qquad (3)$$

$$F_{i,1} = p_1 \cdot 2b + \dot{m} w_1 = 2b(p_1 + \rho w_1^2)$$

From Bernoulli's equation we get

$$p_1 = \frac{1}{2}\rho(w_2^2 - w_1^2)$$

$$F_{i,1} = b\rho(w_2^2 + w_1^2) \qquad (4)$$

The use of Bernoulli's equation is justified, if boundary layers remain thin compared with b, which holds for practically all valve channel flows under consideration (see e.g. [1]). Introducing pressure loss Δp and its coefficient ζ, see fig.2, we get finally

$$\boxed{F_{i,1} = 2b.\Delta p.(1+\frac{1}{\zeta})} \qquad (5)$$

In words:

$$\boxed{\text{Impulsive force } F_i = \text{port area } A \cdot \text{pressure difference } \Delta p \cdot (1 + 1/\zeta)} \qquad (6)$$

In this general form equation (6) holds also for ports of arbitrary form provided that

- flow deflection is 90^o
- boundary layers remain thin
- ζ is a loss coefficient associated with port velocity w_1

Experimental results indicate good agreement with eq(6).

As ζ varies between 1 ($y \gg b$) and ∞ ($y \ll b$) the theoretical limits of F_i are

$$A \cdot \Delta p < F_i < 2A\Delta p \qquad (7)$$

The most important result for us is that F_i increases with valve lift y for const. Δp. The reason is evident: a greater valve lift y permits higher mass flow and this -according to the momentum theorem- increases the impulsive force F_i.

The simple model of fig.2 idealizes somewhat real flow conditions in valve channels. Nevertheless it is helpful, to understand this simple case in detail, before investigating more complicated devices empirically.

Now let us consider channel devices with 2 x 90^o deflection of gas flow. Here we cannot calculate F_i from ζ due to lack of jet flow solutions. So we use the following analogous equation, incorporating a dimensionless force coefficient c_p, to be determined empirically

$$\boxed{F_i = A.\Delta p.c_p} \qquad (8)$$

On the contrary to some other authors "A" stands for the seat port area, not for the valve plate area ($A=\pi\cdot\Sigma(R_a^2-R_i^2)$). Frequently a so called "force area A_f" is used instead of A. Evidently it is $A_f=A.c_p$. The author prefers to use A and c_p as most appropriate because these quantities are coherent with the above given theoretical background.

Fig.4 gives values c_p for a 3-ring plate valve with 2 x 90^o flow deflection, adapted from measurements published by Frenkel [5]. Reinisch [6] has published experimental results for a 2-ring plate valve which show smaller increase in flow force than fig.4. In this paper we use the values of fig.4 for multi-ring valves.

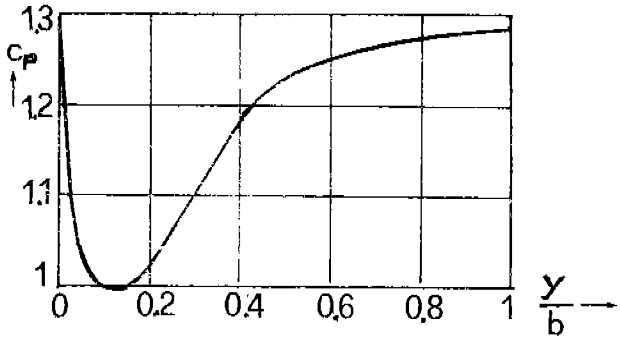


FIGURE 4 c_p for multi-ring plate valves [5]

The author has estimated Mach number influence by comparison of simple compressible and incompressible solutions of jet flows and finds, that this influence is small, even under sonic outflow condition.

VALVE PLATE AS MASS POINT

Let us first consider the simple configuration as given in Fig.1. The equation for the motion of the valve plate, idealised as a mass point, gives

$$m\ddot{y} + cy + F_{spr,o} - \Delta p.A(0.9+0.39\frac{y}{b})=0 \quad (9)$$

In this eq. the linear approximation for F_i is used as given in appendix. From eq(9) follows

$$m\ddot{y} + (c-\frac{A}{b}\Delta p 0.39)y + F_{spr,o} = 0$$

Using A/b=21 and dividing by "l" results in

$$m_1\ddot{y} + (c_1-0.78\Delta p)y + F_{spr,o,1} = 0 \quad (10)$$

$F_{spr,o,1}$ stands for the spring preload per unit length. For constant pressure difference Δp across the valve the general solution of eq(10) is listed in Table 1, next page. The constants A,B,C can be calculated, if initial conditions of plate motion are given. If the solution leads to a motion which is not completely within the allowed lift y=0 to s, repeated reflections may occur with frequencies higher than natural frequency, fig.5.

The effect of the impulsive force F_i is twofold:

- lift of steady state equilibrium position y_{equ} of valve plate

TABLE 1 Solutions of equation (10) for constant pressure difference Δp			
Periodic case	$y = A + B.\sin\bar{\omega}t + C.\cos\bar{\omega}t$	symbol: P	(11)
Aperiodic case	$y = A + B.e^{-\bar{\omega}t} + C.e^{+\bar{\omega}t}$	symbol: A	
Indifferent case	$y = A + B.t + C.t^2$	symbol: I	
$\omega = \sqrt{\frac{c}{m}} = \sqrt{c_1/m_1}$; $\bar{\omega} = \sqrt{\frac{\lvert c_1 - 0.78\cdot\Delta p\rvert}{m_1}} = \sqrt{\frac{\lvert c_1 - c_{F_{i,1}}\rvert}{m_1}}$			$c_{F_{i,1}} = \lvert \partial F_{i,1}/\partial y \rvert$

- lowering of natural frequency $\omega/2\pi$ to $\bar{\omega}/2\pi$ of valve plate or inverting periodic to aperiodic case.

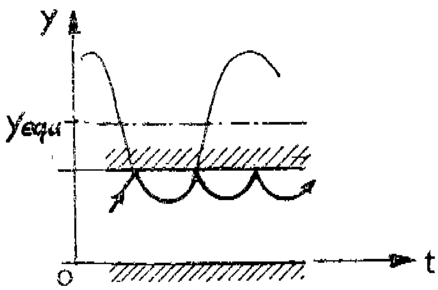


FIGURE 5 Solution with reflections

F_i acts like a spring with negative stiffness ($c_{F_{i,1}} = -0.78\Delta p$).

Now let us consider the case, when $\Delta p = \Delta p(t)$. Eq(9) could be solved numerically, again with the plate considered as a mass point.

SPRING FORCE AND IMPULSIVE FORCE AS LINE-LOADS

Let us leave the mass point idealisation and regard a simple strip as a valve plate fig.6. For this we use distributed loads for spring force and impulsive force(per unit length of channel). If we superimpose a small longitudinal tilting disturbance on the lift of the strip, the lines representing load distributions diverge from parallel. For small inclinations we can neglect threedimensional effects on flow and calculate $F_{i,1}$ according to eq(5) with the local lift y, see fig.6.

Considering the moment on the tilted plate we can see from fig.6, that there are two possibilities: the resulting moment acts against the tilting disturbance(and is stabilising) or it amplifies the tilting (i.e.motion is unstable). This is expresed by

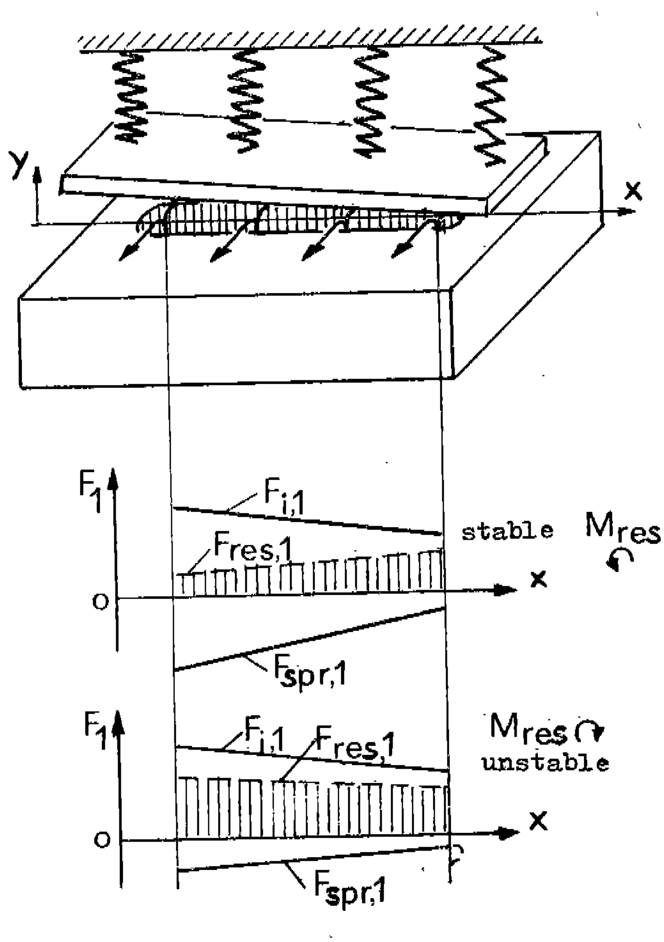


FIGURE 6 Forces on inclined strip

motion stable	$\left(\frac{\partial F_{res,1}}{\partial y}\right)_{\Delta p=const} > 0$ or $\left(\frac{\partial F_{res}}{\partial y}\right)_{\Delta p=const} > 0$ or $c_1(y) > c_{F_i}(y,\Delta p)$	$\frac{c_1^*}{\Delta p} > 0.78$
motion unstable	—‖— < 0 or —‖— < 0 or —‖— $<$ —‖—	$\frac{c_1}{\Delta p} < 0.78$
*estimation for simple configuration according to fig.1; F_i approximated linearly		

(12)

In these formulas c_1 and $c_{F_{i,1}}$ in the case of nonlinear spring and impulsive force stand for

$$c_1 = \left|\frac{\partial F_{spr,1}}{\partial y}\right| = c_1(y);\quad c_{F_{i,1}} = \frac{\partial F_{i,1}}{\partial y} = c_{F_{i,1}}(y,\Delta p) \qquad (13)$$

The same conditions for stability apply evidently for ring and multi-ring plate valves. <u>The essential criterion is:</u>

does rate of increase in spring force with valve lift exceed rate of increase in impulsive force or not.

Now let as make a closer look at stability during the opening and closing motion of the valve plate.

<u>Opening</u>

Fig.7 shows a typical curve $\Delta p(t)$, when pressure pulsations in plenum are absent.

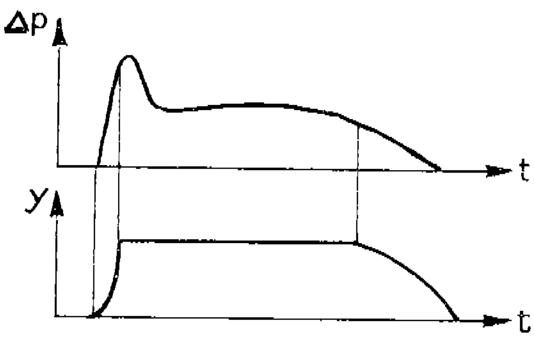


FIGURE 7 Typical curves $\Delta p(t)$. $y(t)$

The plate opens with rapidly increasing values $\Delta p(t)$ and closes with slowly decreasing values $\Delta p(t)$. So the plate may enter unstable conditions during the process at a certain value Δp. Fig.8 demonstrates this for the simple configuration due to fig.1 with linear approximated F_i-characteristics.

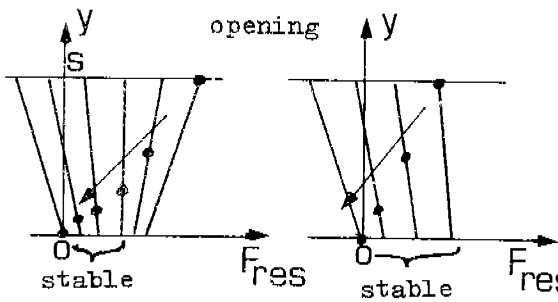


FIGURE 8 Parameter lines $(F_{res})_{\Delta p=const}$

The dots mark the instantaneous positions along the various parameter lines F_{res} (for const. values of t and hence Δp).

Let us now consider more realistic conditions. Fig. 9a shows a typical spring characteristic for a spring with bending arms. For the impulsive force we use a typical characteristic for a multi-ring plate valve as given by fig.4 and eq(8).

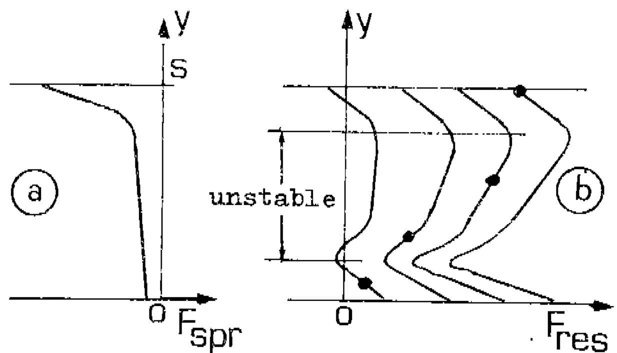


FIGURE 9 Typical parameter lines for multi-ring plate valves

Fig.9b shows typical parameter lines F_{res} for an opening process. Dots again mark instantaneous positions y of plate on the corresponding parameter lines. From fig.9b it arises that instability can develop half way during opening and be followed by an end period of stable seat parallel motion. In this period tilted positions of plate which may have been established in previous period will be reduced due to high stiffness of spring in end period.

Closing

Here Δp decreases relatively slowly when valve plate starts to close(see fig.7). Fig.10 gives typical parameter lines for linearised force characteristics as used previously.

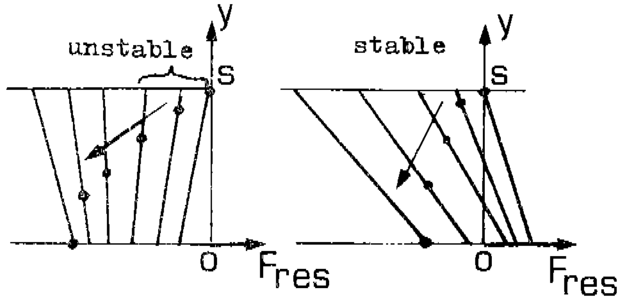


FIGURE 10 Parameter lines, closing

Fig.11 shows typical situations for multi-ring valve plates with bending arm springs. It can be seen that there is a broad instability region between seat and guard.

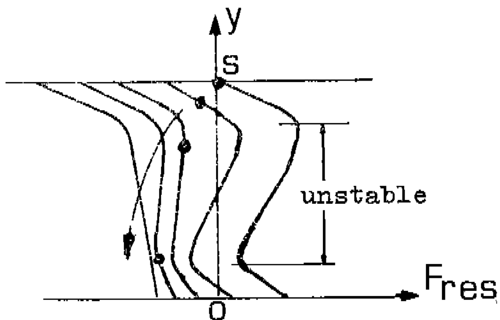


FIGURE 11 Typical parameter lines for multi-ring plate valves(closing)

In Table 1 we have introduced symbols P,A, I to characterize principal conditions of motion. We can refine this procedure by adding a second symbol, according to equilibrium position of characteristic line ($F_i = F_{spr} \rightarrow y_{equ}$). Table 2 gives this symbols.

TABLE 2 Symbols to characterize equilibrium position y_{equ} of parameter lines
s...y_{equ} within valve lift
+...y_{equ} above valve guard
−...y_{equ} below valve seat

Table 3 gives a survey of important cases.

If one wants to estimate the stability of seat parallel motion of a given valve, one can proceed as follows:

→ Find spring stiffness c from valve data; c may not be constant when spring plates with bending arms are used: $c=c(y)$

→ Calculate spring stiffness per unit channel length:

$$c_1 = c/l \quad l..\text{total length of channel}$$

For multi-ring plate valves with given seat port area A, width 2b of channels:

$$c_1 = c.2b/A$$

→ Find pressure difference across valve $\Delta p(y)$ during opening or closing period from computer simulation(with parallel motion), measurements, or general experience or loss coefficient ζ.

→ Form quotient $c_1/\Delta p = f(y/b)$ and enter diagram for estimation of stability, Table 4.

The left hand diagram is derived from fluid flow theory and merits a good deal of confidence. The right hand diagram is derived from fig.4[5]. According to other sources the curve $c_p(y/b)$ for multi-ring plate valves is more flat and resembles the curve with 1 x 90° deflection flow. As configurations in multi-ring valves differ considerably, care should be taken when drawing more than rough conclusions from the diagram at the right of Table 4.

Diagrams in Table 4 give no values for $y/b < 0.2$. Beyond this limit reattachment of flow to seat wall will certainly occur and this gives stable conditions.

Current practice in spring dimensioning is based on the requirement that the plate begins to close early enough to reach the seat even when pressure differences are low. This requirement is absolutely necessary; otherwise volumetric efficiency will decrease and plate impact velocity become excessively high. So there is only a restricted margin to take into consideration the additional requirements of stability of motion.

In existing valves one finds usually

$$c_1 = 0.05 \text{ to } 0.5 \text{ bar}$$

the higher values for high speed compressors or for high pressures. If one compares this with diagrams in Table 4, one would guess that many valves working with pressures up to, say 10 bar could avoid unstable motion. On the contrary high pressure valves are likely to work with unstable motion conditions. Limiting of valve lift to values as small as 0.5mm allows small tilting angles in these cases.

REFERENCES

See Part II.

TABLE 3 Some basic cases of plate motion with Δp = const

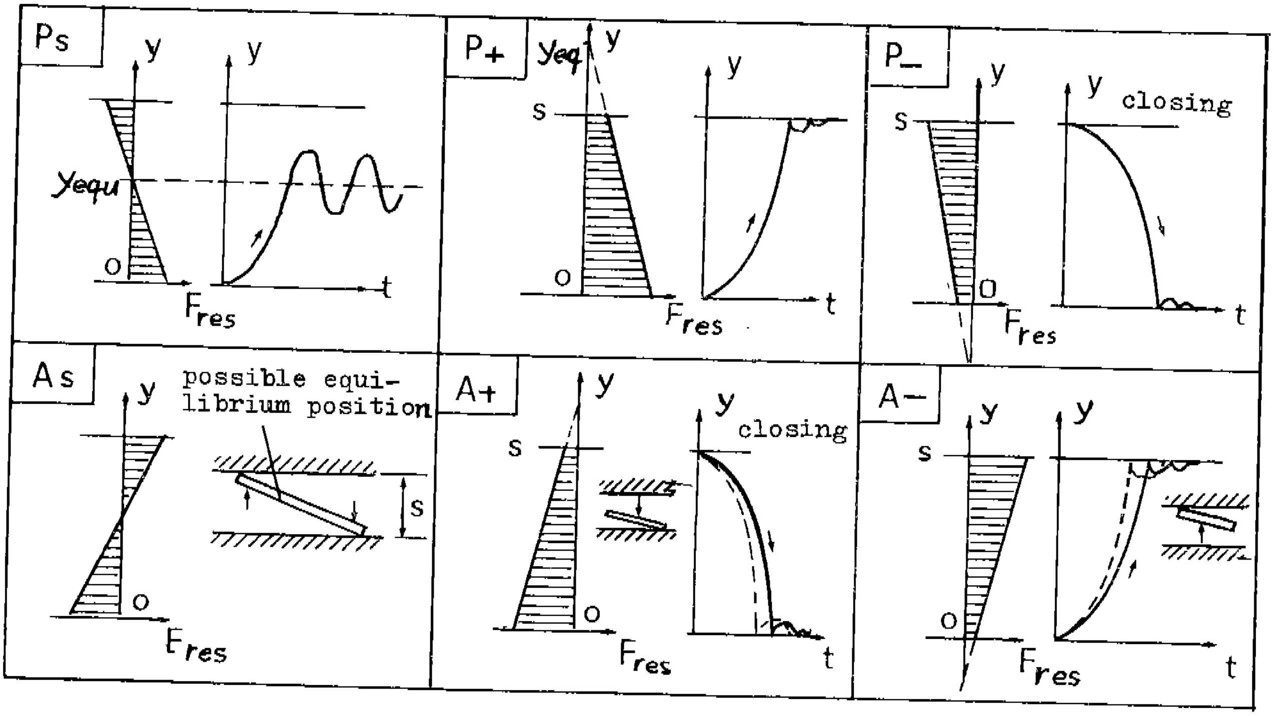
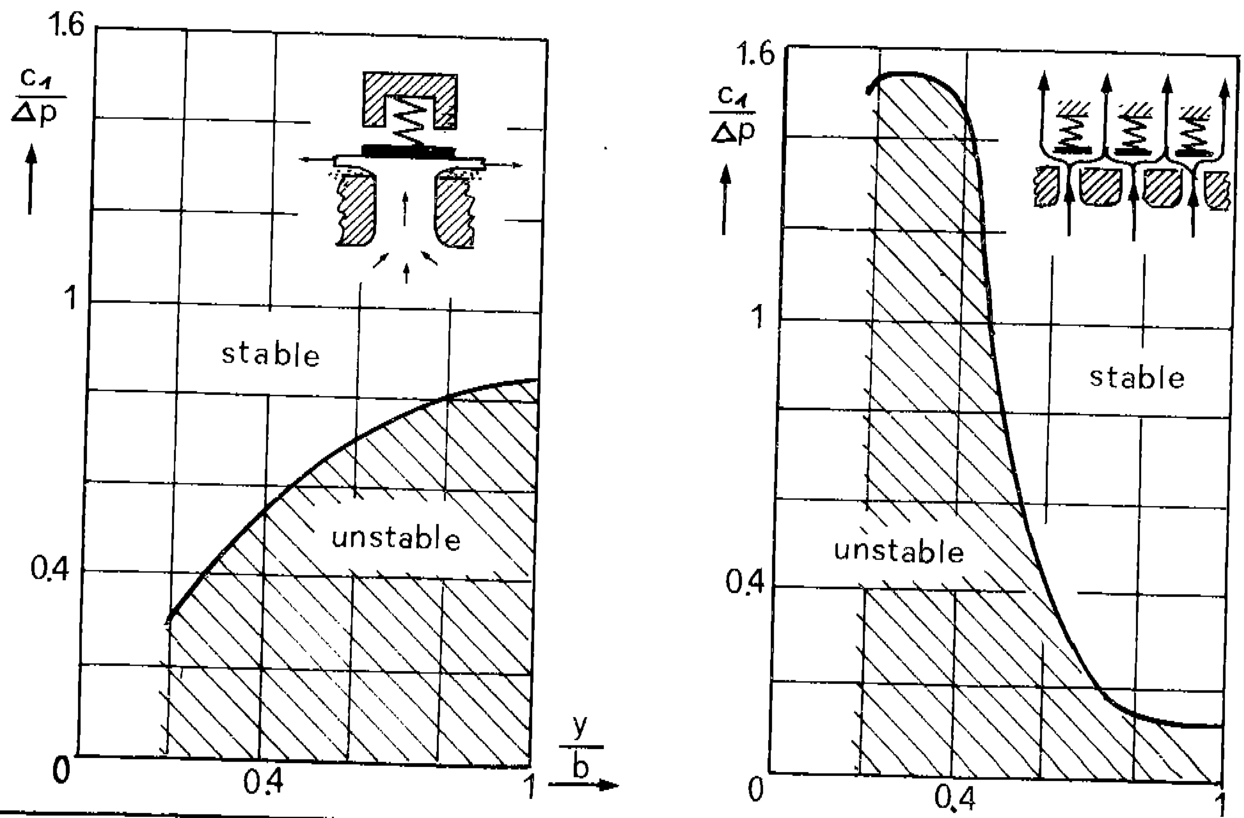


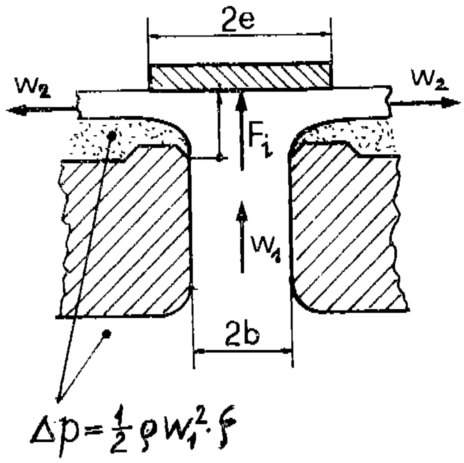
TABLE 4 Diagrams for estimating stability of seat parallel motion (spring force = line load)



APPENDIX

Infinite slot with 90° deflection flow

Pressure loss Δp and impulsive force F_i, calculated from potential flow theory and momentum theorem [1]



y....valve plate distance from seat edge(if edges chamfered, from lower edge!)
2b...width of slot
2e...width of plate
A....port area; $A=2b.l$; l...length of slot
w_1...velocity in slot; $w_1=\dot{V}/A$, $\dot{V}$..volumetric flow rate
ρ....density of gas
ζ....pressure loss coefficient
Δp..pressure difference across valve
F_i...impulsive force
$F_{i,1}$.impulsive force per unit of length; $F_{i,1}=F_i/l$
c_p...force coefficient
d....asymptotic width of leaving jet branch

Potential flow theory(jet flow) leads to the following equation, which allows to calculate d from lift y and from b:

$$\boxed{\frac{y}{b} = \frac{d}{b} + \frac{1+(d/b)^2}{\pi}\ln\frac{1+(d/b)}{1-(d/b)} \qquad \zeta = \left(\frac{b}{d}\right)^2}$$

Provided that deflection angle is $90°(e \approx\geqslant 1.5b)$, momentum theorem leads to

$$\boxed{c_p = 1 + \frac{1}{\zeta}}$$

and impulsive force is $F_i = A.\Delta p.(1+1/\zeta) = A.\frac{1}{2}\rho w_1^2(\zeta+1)$

From evaluation of above given equation follows:

y/b	ζ	c_p
0.1	269	1.004
0.2	68.3	1.015
0.3	30.9	1.032
0.4	17.8	1.056
0.5	11.7	1.085
0.6	8.4	1.12
0.7	6.44	1.16
0.8	5.14	1.19
0.9	4.25	1.24
1.0	3.62	1.28
1.2	2.78	1.36
1.4	2.26	1.44
1.6	1.92	1.52
1.8	1.69	1.59
2.0	1.52	1.66
2.2	1.40	1.71

Approximation of c_p by a linear function: