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A Model for Valve Flow Taking Non-Steady Flow into Account, Part II

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A MODEL FOR VALVE FLOW TAKING NON STEADY FLOW INTO ACCOUNT

Part II

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5.FLOW FORCE ON VALVE PLATE

For a derivation of flow force on valve plate under non steady flow conditions we use the momentum theorem and a control volume as indicated in fig.8

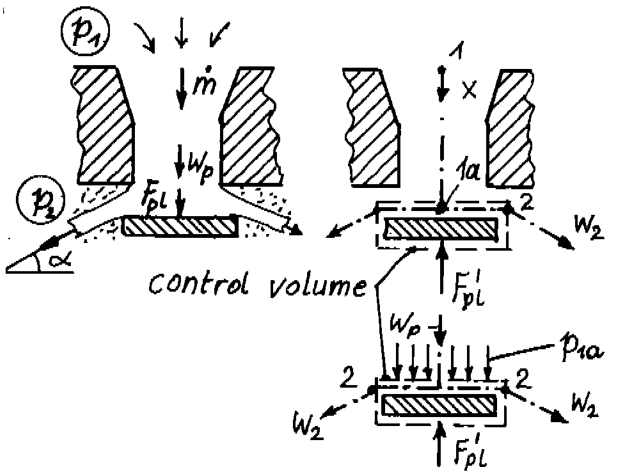


Fig.8 Control volume for calculation of force on valve plate

To simplify calculations we put $p_2=0$ ($p_1=\Delta p$). The flow force on valve plate is caused by:

- force from(non steady) pressure in point 1a: $F_p=p_{1a}A_p$
- deflection flow according to momentum change

The non steady character of the phenomenon is expressed by the fact that we calculate p_{1a} and w_2 with non steady flow theory.

The momentum theorem yields:

$$F'_{pl} - p_{1a}A_p = \dot m(-w_2 \sin\alpha + w_p)$$

F'_{pl} is an external force which can keep the plate in equilibrium.The force F_{pl} -exerted from the gas upon the plate- has the same value but is opposite in direction:

$$F_{pl} = A_p p_{1a} + \dot m(w_p - w_2 \sin\alpha)$$

Using the continuity equation and equ(4.21) results in

$$w_p = w_2 A_2/A_p \qquad \dot m = w_2 A_2 \rho$$

$$\frac{p_1 - p_{1a}}{\rho} = \frac{w_p^2}{2} + \frac{dw_2}{dt}\underbrace{\int_{x_1}^{x_{1a}} \frac{A_2}{A(x)}dx}_{I_2(s) = I(s) - l_2} + w_2 w_v I_1(s)$$

$$\boxed{F_{pl} = A_p\left\{\Delta p + \frac{1}{2}\rho w_2^2\left[\left(\frac{A_2}{A_p}\right)^2 - 2\frac{A_2}{A_p}\sin\alpha\right] - \rho I_2(s)\cdot\frac{dw_2}{dt} - w_2 w_v \rho I_1(s)\right\}} \qquad (5.1)$$

Contrary to this the steady state flow force is

$$\bar F_{pl} = A_p c_p \Delta p = A_p \cdot c_p \cdot \frac{1}{2}\rho \bar w_2^2 \qquad (5.2)$$

$\bar w_2$ is the steady state velocity($\bar w_2 = \sqrt{2\cdot\Delta p(t)/\rho}$) and c_p is the force coefficient derived from steady state experiments. For an evaluation of equ(5.1) one needs effective flow area A_2 and α from steady state tests, furthermore $I(s)$, $I_2(s)$, calculated from valve dimensions and valve plate velocity w_v(from simulation process). Instead of these quantities we may use c_p-values from steady state experiments. Comparing equ(5.1) and (5.2) one obtains the following equation

$$\boxed{F_{pl} = A_p\left\{\Delta p(t) + \tfrac{1}{2}\rho w_2^2(t)\left[c_p(s)-1\right] - \rho I_2(s)\cdot\frac{dw_2(t)}{dt} - w_2(t)w_v(t)\rho I_1(s)\right.} \tag{5.3}$$

Using a computer one can get $\Delta w_2/\Delta t$ and w_2 directly from equ(4.22).

6.REMARKS ON FRICTION FORCE ON VALVE PLATE

Frequently a friction force F_f is used in the equation for valve plate motion. This force is assumed proportional to valve plate velocity $\dot{s}=w_v$

$$F_f = c_f \cdot \dot{s} \tag{6.1}$$

Usually c_f is adjusted to match experimental results.From the point of view of non steady fluidmechanics as it has been pointed out by the author so far,there is no reason to introduce such a frictional force.

In equ(5.1) and (5.3) we can, however, find a term which is proportional to valve velocity $w_v=\dot{s}$. But this term is also proportional to $w_2(t)$ and for this reason not exactly comparable with the "friction force" of equ(6.1).Moreover equ(5.1),(5.3) also include other terms with non steady velocity $w_2(t)$.

So one may say that in the traditional model the frictional force equ(6.1) plays the role of a rough consideration of gas inertia effects. In the model presented in this paper there is no need for introducing a frictional force because the inertia effects have already been taken into account in the flow process. While there is only one constant(c_f) in the traditional model to consider gas inertia we find 3 parameters in equ(5.1),(5.3) to account for inertia: $I_1(s)$,$I_2(s)$,$w_2(t)$. These parameters are not freely available for adjustment but calculated from valve data.

Thus it can be expected that the new model gives a better basis for an approach to the calculation of valve lift-time histories.

7.SOME COMPUTER RESULTS

To become familiar with the gas inertia effect let us now use the new equations and find solutions for some exemplary cases. For all calculations the data of the valve of fig.4 were used. and the density assumed to be ρ=5.38kg/m. Force on valve plate has been calculated with equ(5.1) assuming $\alpha=\Omega$ for reasons of simplicity.

Case 1: Closing of valve with a constant plate velocity of 4m/s was assumed and Δp beeing 17500N/m²(constant). The plot,Table 3, ofthe calculated exit velocity $w_2(t)$ shows an increase from 80.7 to 98m/s,while the steady state result is w_2=80.7m/s=constant. The closing plate slows down the gas in the port area and this deceleration increases the pressure in point 1a. This in turn leads to an acceleration of the fluid in the exit area to the plotted value $w_2(t)$. The non steady force on the valve plate is greater by some 10% than the steady state result. This is mainly due to an increase of pressure in point 1a as a consequence of deceleration of the gas in port area.

Case 2: The valve is closed twice as fast as in case 1, whereas all other conditions have remained unchanged. The exit velocity increases from 80.7m/s at the beginning of the closing process to 118m/s at the end of the process.The difference of steady state and non steady valve plate force F_{pl} is more accentuated than in case 1.

Case 3: Conditions are identical to these in case 1 except the pressure difference Δp. While Δp remains constant in case 1 it drops to zero at the beginning of the closing process in case 3. Contrary to case 1 w_2 and F_{pl} decrease due to gas inertia but there is a great difference between steady state and non steady solution.

Case 4: Conditions are identical to those in case 1 except valve plate closing: this is assumed to be parabolic.Calculations show that the increase of exit velocity w_2 is more restricted to the final closing period.

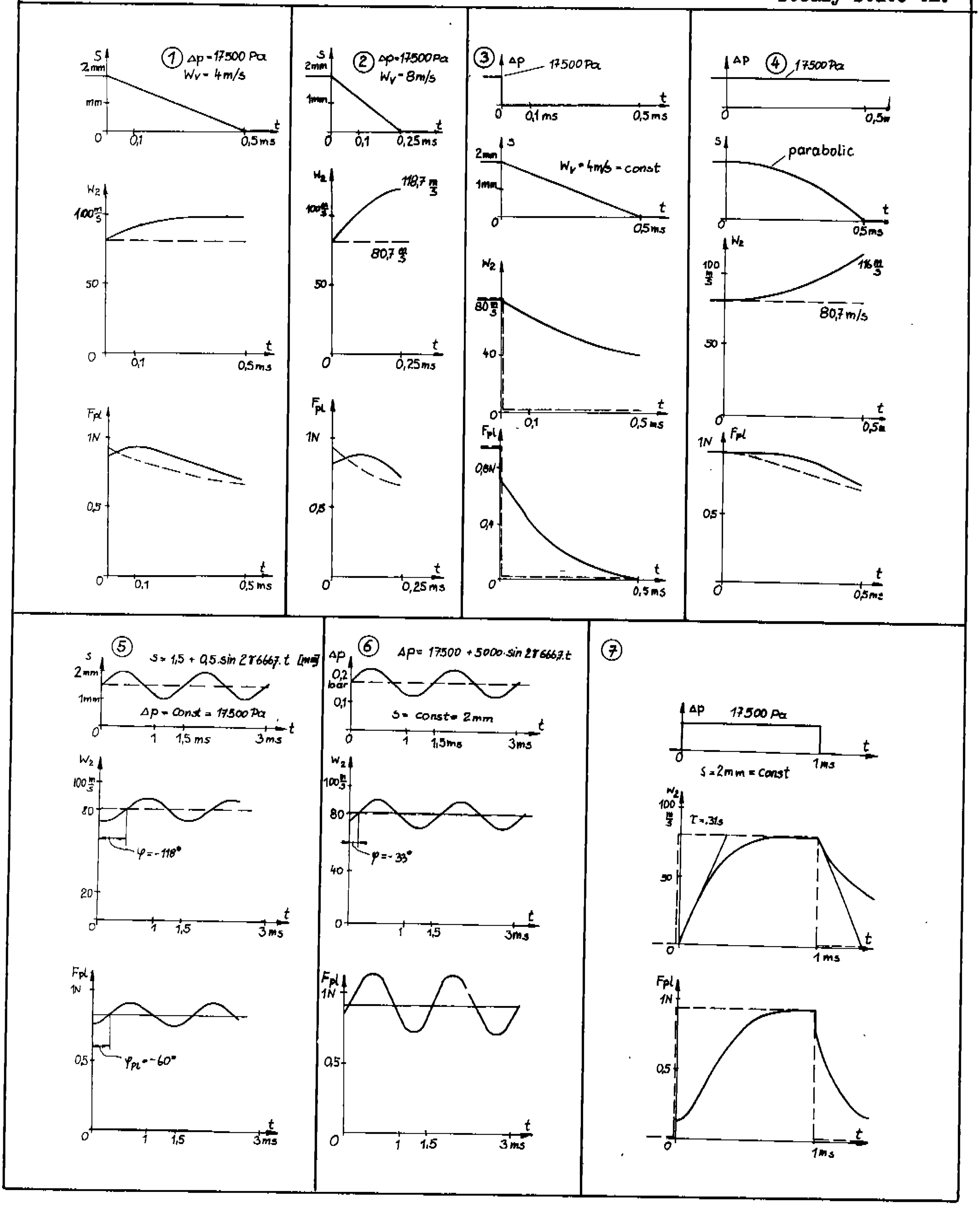
Case 5: A forced sinusoidal movement of the valve plate and constant pressure difference Δp are assumed. Though constant pressure difference Δp is applied, valve plate movement induces a sinusoidal velocity variation of exit velocity w_2 with a certain phase shift φ .This example has already been discussed in section 4.3 in connection with an approximate solution. The difference between numerical(computer) solution and approximate solution was too small to be represented in the plot. F_{pl} shows a smaller phase shift than the exit velocity.

Case 6: At constant lift s a sinusoidal variation of pressure difference Δp was assumed. This example has already been discussed in section 4.2. In addition the valve plate force is plotted.

CASE 7: The response to a short rectangular pressure pulse with a duration of 1ms is demonstrated. The lift s is assumed to be s=2mm=constant. The results $w_2(t)$ and F_{pl} indicate considerable deviations when compared with the rectangular steady state

TABLE 3 Some Computer Results

— non steady theory
--- steady state th.



results. A similiar example has already been discussed in Table 2.

8. INFLUENCE OF VALVE DIMENSIONS ON INERTIA PARAMETERS

The consequences of gas inertia may be summarized as follows:

- Mass flow in valves show a certain time delay when rapid changes in Δp and s take place. When a valve closes, the decelerating gas in port area exerts an additional force on valve plate and delays its closing point. In some cases the gas inertia force may be strong enough for causing a rebound of the valve plate(after dissipation of kinetic energy of valve plate during impact process). This does not necessarily mean that back flow takes place, because the gas inertia may continue in pushing through gas against a negative pressure difference!

- Due to time delay of mass flow -immediately after valve opening- the pressure peak of the gas in the cylinder is increased. Other reasons for this peak are: valve plate mass inertia and oil sticktion.

- Rapid sinusoidal variations in Δp or s cause sinusoidal variations of the velocity field with a certain phase shift. Roughly speaking this may be classified as "damping" but the phenomenon is grasped in detail only by non steady flow equations

A valve in which high velocity regions are restricted to areas near seat edge is definitely a valve with low inertia effects. Quantitatively speaking: a valve with low inertia effects is a valve with low values of inertia parameters I, I_1, I_2. The influence of valve dimensions on these parameters is principally expressed by the integral

$$i = \int_{x_1}^{x_{1a} \text{ or } x_2} \frac{A_2}{A(x)}\, dx \qquad (8.1)$$

High values "i" lead to great inertia effects. Table 4 gives an impression on the influence of valve dimensions on inertia parameters.

Often valve configurations are used with circular holes and long rectangular plates, fig.9. The long ways of gas at relatively high velocities in the gap between valve plate and seat plate result in high inertia effects.

9. EQUATIONS FOR MORE COMPLEX VALVE CONFIGURATIONS

Up to now we have discussed only the simplest valve configuration. Let us now outline the application of the model presented to more complex valve geometries (which may include internal friction

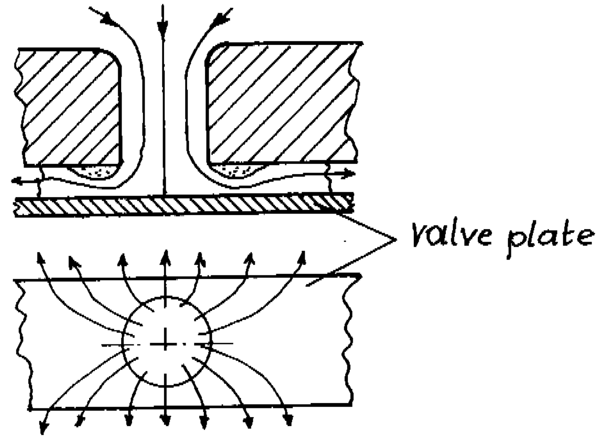


Fig.9 To inertia effect

losses),fig.10. Equ(4.21) has now to be modified to include a loss term Δp_f. If we proceed along the simplified streamline, velocity develops as indicated in fig.10b. Between points 2 and 3 pressure recovery takes place but the gain in pressure is not as high as the Bernoulli equation predicts.Hence there is a pressure loss Δp_f(which may be calculated by momentum theorem or found by experiments. We may modify equ(4.21) to account for this fact. With p(x,t) being the non steady pressure along the idealized mean streamline

$$\frac{p_1(t)}{\rho} = \frac{p(x,t)}{\rho} + \frac{\Delta p_f(x,t)}{\rho} + \frac{w^2(x,t)}{2} + \int_{x_1}^{x_3} \frac{\partial w(x,t)}{\partial t}\, dx \qquad (9.1)$$

Δp_f is only the pressure loss <u>within</u> the valve channel. The total pressure loss is greater according to loss of kinetic energy at the exit. For steady state condition there is

$$\Delta p_f = p_1 - p_3 - \tfrac{1}{2}\rho w_3^2 \quad (\hat{=}\, p_3 - p_{3'}) \qquad (9.2)$$

Δp_f may be expressed by exit velocity w_3 and a loss factor ζ :

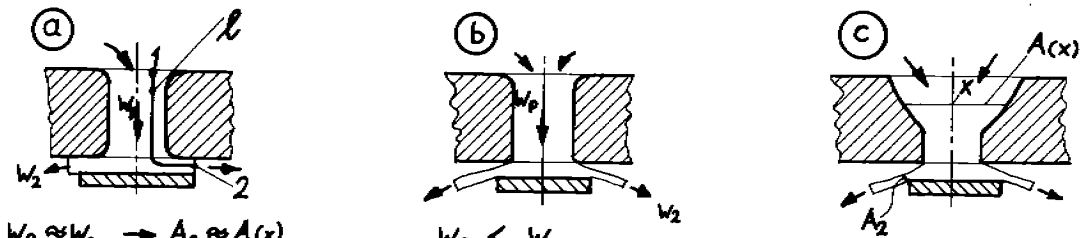
$$\Delta p_f = \zeta \tfrac{1}{2}\rho w_3^2 \qquad (9.3)$$

Putting $x=x_3$ and $[p_1(t)-p_3(t)]/\rho = P(t)$ the following differential equation for exit velocity $w_3(t)$ can be obtained:

$$P(t) - \frac{w_3^2(t)}{2}\cdot(1+\zeta) - I(s)\frac{dw_3(t)}{dt} - w_3(t)\, w_v(t)\, I_1(s) = 0 \qquad (9.4)$$

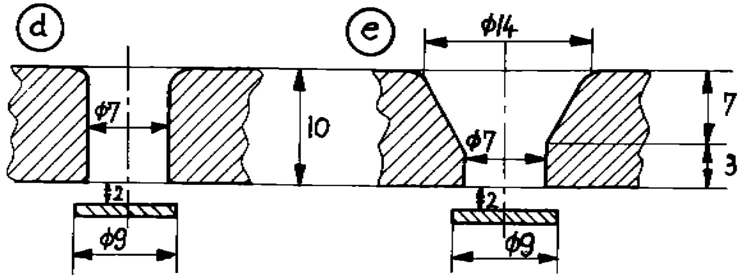
$$I = \int_{x_1}^{x_3} \frac{A_3}{A(x)}\, dx \qquad I_1 = \int_{x_1}^{x_{2a}} \frac{\partial}{\partial s}\left[\frac{A_3}{A(x,s)}\right] dx$$

TABLE 4 Influence of valve dimensions on inertia parameters



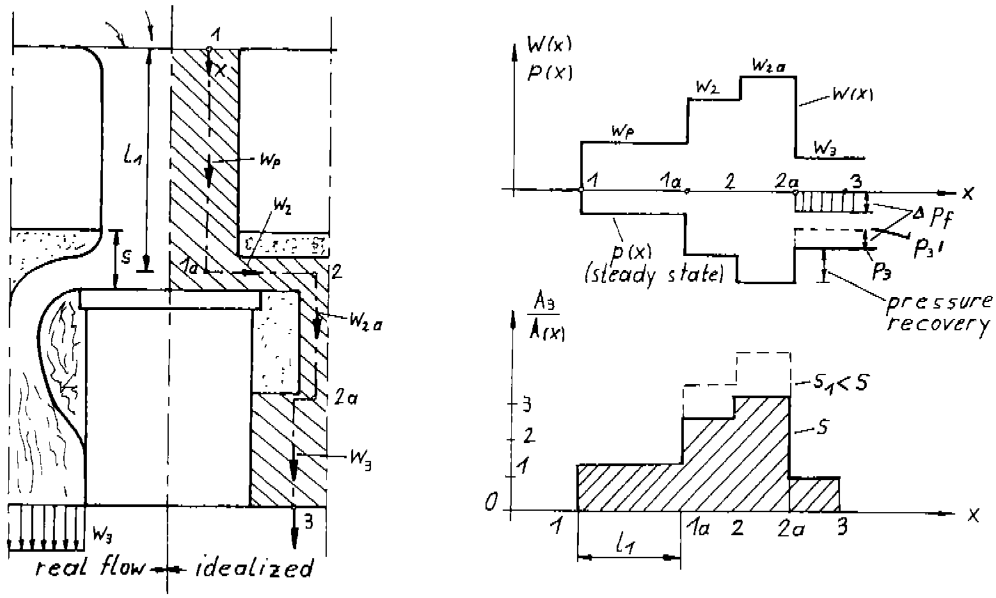
a	b	c
$W_p \approx W_2 \rightarrow A_2 \approx A(x)$	$W_p < W_2$	$I_c < I_b$
$I \approx \ell$ (length of simplified streamline)	$\rightarrow I_b < I_a$, but also $C_{D_b} < C_{D_a}$!	

Calculated values of inertia parameters



	d	e
I	9,18 mm	7 mm
I_1	I_{1d}	$0{,}56 \cdot I_{1d}$
I_2	7,18 mm	5,0 mm

Fig.10 Channel of a ring plate valve



All other velocities may be calculated from w_3 by continuity equation.

The calculation of plate force is somewhat more complex. Using the same guidelines and principles as demonstrated for the simple configuration the reader himself may derive formulas for the plate force for various configurations.

Experiments have been designed to back up the theoretical treatment. The nozzle of an educational wind tunnel is used as an enlarged model of a valve channel and hot wire anemometrie for velocity measurements. This will be reported later.

CONCLUSION

With a relatively moderate effort the gas inertia effect is accessible to mathematical computation.

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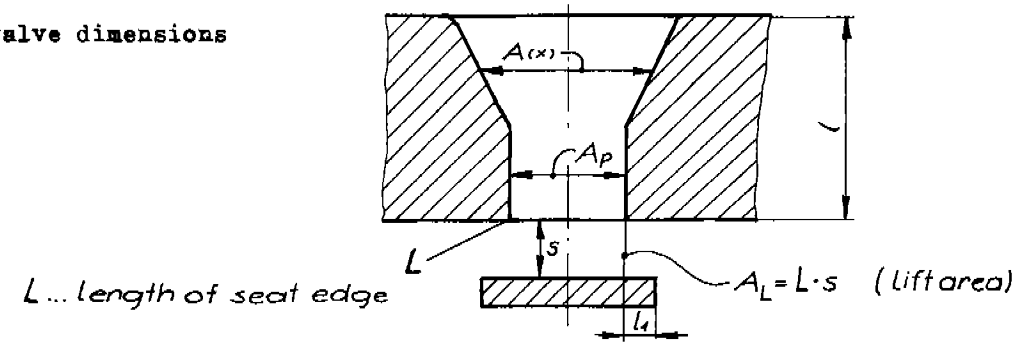
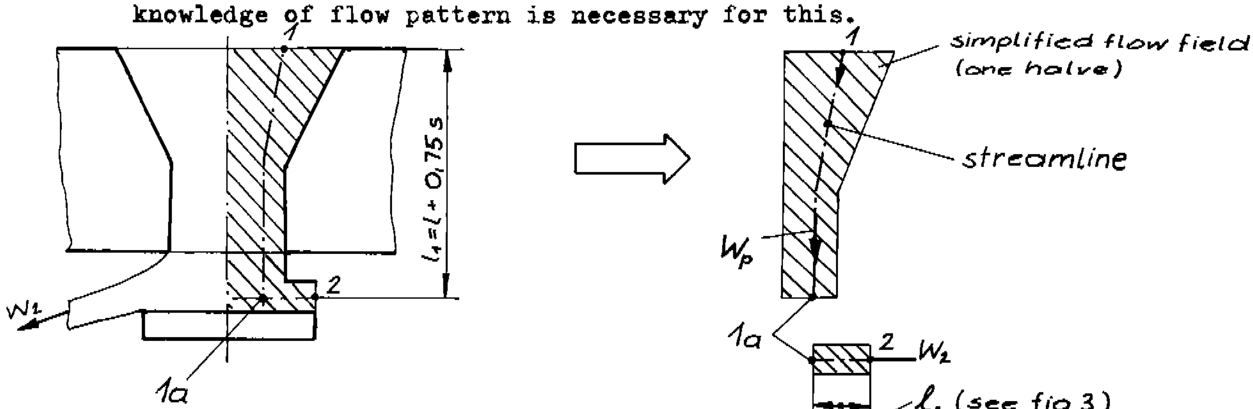
APPENDIX: Calculation of inertia parameters from valve dimensions

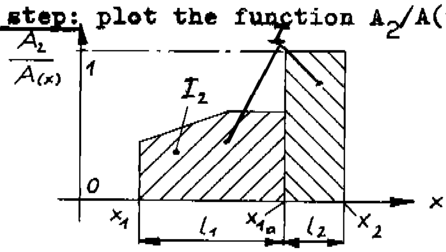
valve dimensions



1st step: simplify flow field and find representativ mean streamline. Basic knowledge of flow pattern is necessary for this.



2nd step: plot the function $A_2/A(x)$



$A_2 = A_L C_D$ ...actual flow area

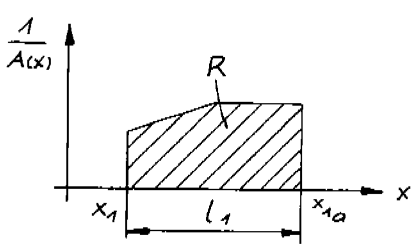
C_D from literature or from experiments

$A(x)$....cross section of simplified flow field

3rd step: Calculate aera under plotted curves as indicated above. This procedure has to be done for various lifts to get curves $I(s), I_2(s)$.

$$\boxed{I = \int_{x_1}^{x_2} \frac{A_2}{A(x)} \cdot dx \qquad I_2 = \int_{x_1}^{x_{1a}} \frac{A_2}{A(x)} \cdot dx = I - l_2 \cdot 1}$$

4th step: Calculation of I_1:
a) plot the function $1/A(x)$ and find area R under the curve as indicated in figure
b) plot a curve $sC_D(s)$ from about 5 values of s and find by graphical differentiation $d/ds(sC_D)$
c) calculate $I_1(s)$ as indicated below



$$\boxed{I_1(s) = L \frac{d}{ds}(s \cdot C_D) \cdot \int_{x_1}^{x_{1a}} \frac{dx}{A(x)} = L \frac{d}{ds}(s C_D) \cdot R}$$

I_1 is a non dimensional quantity, I, I_2 have the dimension of a length (m)