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Non Steady Flow in Valves

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ABSTRACT

This paper continues and brings to a close previous efforts of the author in the field of non steadY valve flow. Extensive experiments with enlarged models of valves led to a set of manageable equations describing non steady flow in valves. It becomes apparent that besides gas inertia two other effects had to be considered: an effect taking into account the non steady work exchange between flow and valve plate and a socalled "gas spring effect" which considers the interaction between valve plate and gas elasticity. This paper as a whole is an extract of parts of a comprehensive investigation on the subject [1].It continues and improves previous activities of the author [2][3][4].

1.INTRODUCTION

When dealing with the problem of modelling flow phenomena in compressor valves, one could differentiate 3 different approaches:

- one dimensional quasisteady flow model: non steady flow is calculated by using steady state flow equations. Current practice.
- simplified one dimensional non steady flow model: the model makes allowance for non steady flow effects in a simplified way.This model gives a much better approach to the real compressor process than the quasisteady flow model at the expense of a slight increase in computer time. This paper deals with such an approach assuming an incompressible fluid.
- Three dimensional non steady flow as described by Navier-Stokes equations. Solutions for these equations taking into account the moving valve plate are far beyond the possibilities of presentday computers.

It is understood that the higher level approach includes the lower one as a special case.

To find out the correct equations for a simplified one dimensional non steady flow model experiments with enlarged models have been carried out. These experiments concern basically valve flutter and are presented elsewhere in these Proceedings[5].A small educational wind tunnel has been adapted for this purpose,Fig.1. In this set up a valve model 50 times larger than a compressor valve is used. The cabin volume models the cylinder volume. Time is reduced by a factor of about 0.01 when compared with real compressor conditions. For a meaningful evaluation of experiments with this set up a correct similarity theory has been worked out. This is presented elsewhere in these Proceedings[5]. The basic equations presented in this paper are the essential results of these experiments.

2.SURVEY ON STEADY STATE FLOW EQUATIONS

The quasisteady flow model uses the steady state flow equations and assumes that the formulas give reasonable results not only for constant but also for time varying pressure differences ΔP_{12}. This approach gives acceptable results if ΔP_{12}-variatios are "slow enough". A quantitative limit for the quasisteady flow model can be derived from

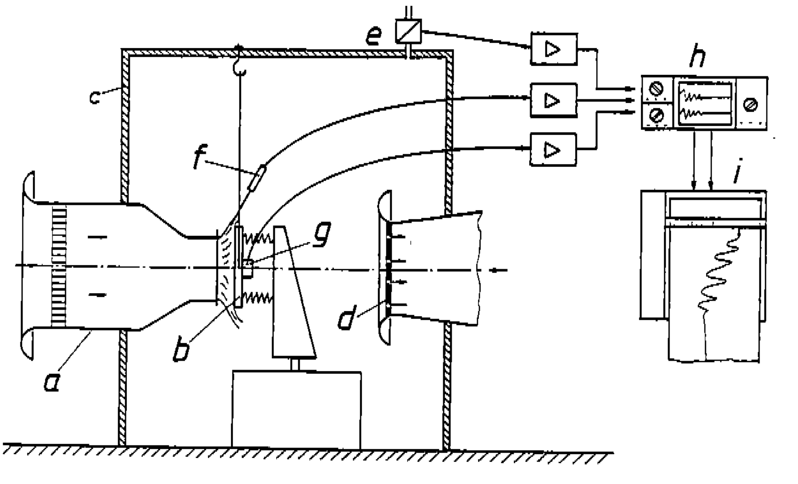


Fig.1 Experimental set up
a wind tunnel nozzle; b spring loaded valve plate; c box(V=17,9m³); d flow resistance; e pressure difference transducer; f hot wire probe; g valve lift transducer; h digital storage oscilloscope; i recorder.

the position of a non steady flow model only(see section 3).

Table 1 summarizes formulas and notation for this flow concept. Valve configurations with simple 90°-deflection flow and incompressible fluid is assumed throughout in this paper. Furthermore it is assumed that the bulk flow leaves the valve with a more or less constant exit velocity W_2. Contrary to configuration "a", configuration "b" allows for frictional effects at the entrance(β).

3.NON STEADY FLOW EQUATIONS FOR VALVE FLOW

Following the argumentations in a previous paper [3], the equation for a one dimensional non steady flow reads,Fig.2,

$$\underbrace{\frac{P_1(t)-P_2(t)}{\rho}=\frac{\Delta P_{12}(t)}{\rho}=\frac{W_2^2(t)}{2(1+\beta)}}_{a}+\underbrace{\int_{s_1}^{s_2}\frac{\partial W(s,t)}{\partial t}}_{b}-\underbrace{\frac{P_M(t)}{\dot{m}_2(t)}}_{c} \qquad (1)$$

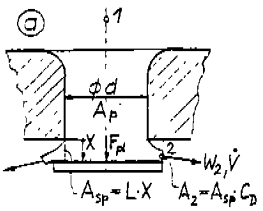
a...quasisteady flow equation, in addition to [3] this equation allows also for frictional losses at the entrance(β).
b,c..non steady flow effects; b gas inertia term
c...term which accounts for non steady work exchange between flow and plate. When the plate is moving, some work is transfered from the flow to the plate,resulting in a reduction of exit velocity W_2. This is true for the opening movement. In case of closing W_2 is increased. This term was not included in [3] . It is backed by experiments and becomes especially important for small values W_2.

Now let us discuss non steady effects in more detail.

Gas inertia effect

Using continuity equation the integral term may be evaluated and results in

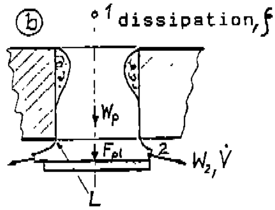
TABLE 1 SURVEY ON STEADY STATE VALVE FLOW EQUATIONS. NOTATION



$$W_2 = \sqrt{\frac{2\Delta P_{12}}{\rho}}$$

$$\dot{V} = A_{sp} \cdot C_D(X) \cdot W_2$$

$$F_{pl} = A_p \cdot c_p(X) \cdot \Delta P_{12}\ ;\ c_p \approx 1$$

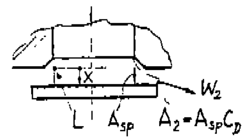


$$W_2 = \sqrt{\frac{2\Delta P_{12}}{\rho(1+\beta)}}\ ;\quad \beta \approx \zeta\left(\frac{A_{sp}C_D}{A_p}\right)^2$$

$$\dot{V} = A_{sp} \cdot C_D(X) \cdot W_2$$

$$F_{pl} = A_p \cdot c_p(X)\, \Delta P_{12}\ ;\ c_p \approx \frac{1}{1+\beta}$$

(c) notation for seat edge with chamfer



Notation:

- A_p...port area
- A_2...effective flow area
- A_{sp}..opening area
- C_D...discharge coefficient
- c_p...force coefficient
- F_{pl}..force on valve plate
- L....length of seat edge
- ΔP_{12}.pressure diff.across valve
- $\dot{V}$...volume flow rate
- W_2..velocity of emerging jet
- W_p..port velocity
- X...valve plate lift
- β...coeff. concerning W_2
- ζ...entrance friction coeff. with respect to W_p

$$\int_{s_1}^{s_2} \frac{\partial W(s,t)}{\partial t}\, ds = J \cdot X(t) \cdot \dot{W}_2(t) + J \cdot \dot{X}(t) \cdot W_2(t) \qquad (2)$$

"J" is a parameter depending on valve geometry only

$$J = L\, C_D \int_{s_1}^{s_2} \frac{ds}{A(s)} \qquad (3)$$

A(s) is the varying cross section along streamline 1-2.

Instead of using 3 parameters accounting for gas inertia in [3] (I, I_1, I_2) here we use only one, namely J. J is a non dimensional quantity. This simplification was made possible by the following assumptions:

a) C_D is independent of valve lift

b) C_D is not subject to inertia effects

c) the contribution of gas masses after 90°-deflection to the gas inertia effect is small as compared to gas masses in the seat plate channel.

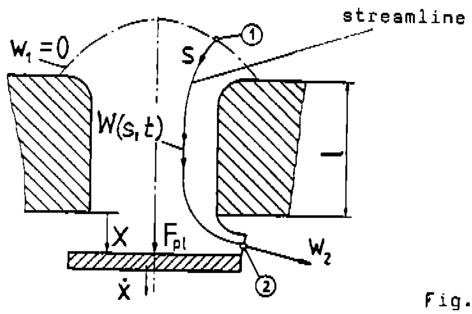


Fig.2 Notation to equation (1)

Experiments and theoretical reasoning back these assumptions [1], which result in a considerable simplification in the mathematical treatment.

For valve channels with constant cross section $A(s)=A_p$ (see Fig.2) the integral in (3) may be evaluated and reads

$$\boxed{J = \frac{LC_D(1+\varkappa d)}{A_p}} \qquad (4)$$

$\varkappa$ is an end correction coefficient which accounts for gas masses accelerated outside the valve channel. Experiments and theoretical considerations indicate for a channel with one free end and the other end with a plate in some distance ($X=0{,}04d$ to $0{,}14d$)

$$\varkappa \approx 1{,}4$$

For an existing valve J may be found by a simple acoustic experiment.

Non steady work exchange effect

The work (per second) transfered to gas becomes

$$P_M(t) = -F_{pl}\cdot\dot{X}(t) \qquad (5)$$

The non steady plate force $F_{pl}(t)$ may be calculated with stagnation pressure of non steady exit velocity W_2

$$F_{pl}(t) = A_p \frac{1}{2}\rho W_2(t) \qquad (6)$$

This formula is much simpler than the corresponding expression in [3] (equ(5.3)). This is mainly due to the assumption c) mentioned above. Finally we get for the specific work exchange

$$\frac{P_M}{\dot{m}_2} = \frac{A_p\frac{1}{2}\rho W_2^2\dot{X}}{A_2\rho W_2} = \frac{A_p W_2\dot{X}}{2LC_D X} \qquad (7)$$

So we may summarize non steady flow equations as presented in Table 2.

Contrary to steady flow it is not possible to calculate W_2 directly from ΔP_{12}, because $W_2(t)$ is a variable in the differential equation (8). With a computer equ(8) may be integrated step by step. Using $\dot{W}_2=dW_2/dt\approx\Delta W_2/\Delta t$ results in

TABLE 2 NON STEADY VALVE FLOW EQUATIONS	
$\frac{\Delta P_{12}(t)}{\rho} = \frac{W_2^2(t)}{2\cdot(1+\beta)} + J\left[\dot{W}_2(t)\cdot X(t) + W_2(t)\cdot\dot{X}(t)\right] + \frac{A_p W_2(t)\cdot\dot{X}(t)}{2LC_D\cdot X(t)}$	(8)
$F_{pl}(t) = A_p\cdot\frac{1}{2}\rho W_2^2(t)$	(9)
$\dot{V}(t) = LC_D\cdot W_2(t)\cdot X(t)$	(10)

$$\left.\begin{aligned}\Delta W_2 &= \frac{1}{J\cdot X(t)}\left[\frac{\Delta P_{12}(t)}{\rho} - \frac{W_2^2(t)}{2\cdot(1+\beta)} - \frac{A_p W_2(t)\cdot\dot{X}(t)}{2LC_D\cdot X(t)} - J\cdot W_2(t)\cdot\dot{X}(t)\right]\cdot\Delta t\\ W_2(t+\Delta t) &= W_2(t) + \Delta W_2\end{aligned}\right\} \quad (11)$$

As a first application of these equations the limits up to which steady state flow equations give a reasonable approach to reality may be estimated [1]. Adopting a deviation of 10% leads to the following estimates

$$\frac{JX4\pi f_o}{W_2} < 0.10 \quad \text{gas inertia effect negligible}$$

$$\frac{A_p 2\pi f_o}{LC_D W_2} < 0.10 \quad \text{non steady work exchange effect negligible} \qquad (12)$$

with f_o...frequency of plate

For fluttering valves f_o is the flutter frequency. For valves without flutter, f_o may be estimated: $f_o \approx 1/2T_K$ with T_K as opening period of valve.

Calculations indicate that usually steady state flow equations give a good approach in the absence of valve flutter. With flutter, gas inertia usually is of importance. The effect of non steady work exchange is important in fluttering valves with low velocity level.

4. DETERMINATION OF THE INERTIA PARAMETER J BY AN ACOUSTIC EXPERIMENT

Theoretical reasoning lead to a simple method to determine J by an acoustic experiment, Fig.3. From a measurement of the resonance frequency $f_{g,o}$ of the valve-volume arrangement one gets

$$\boxed{J = \frac{kP}{\rho}\cdot\frac{LC_D}{V4\pi^2 f_{g,o}^2}} \qquad (13)$$

k,P,ρ...isentropic exponent,pressure,density of ambient air

$kP/\rho = a^2$, a...velocity of sound

In an arrangement according to Fig.3 the resonance amplitude maybe weak. This difficulty could be removed by the following procedure: replacement of reed by the(1/2") microphon diaphragm of the sound level meter, held perpendicular to the hole axis in a distance of about 1mm(thus simulating the open reed). If the reed is removed and the microphon in great

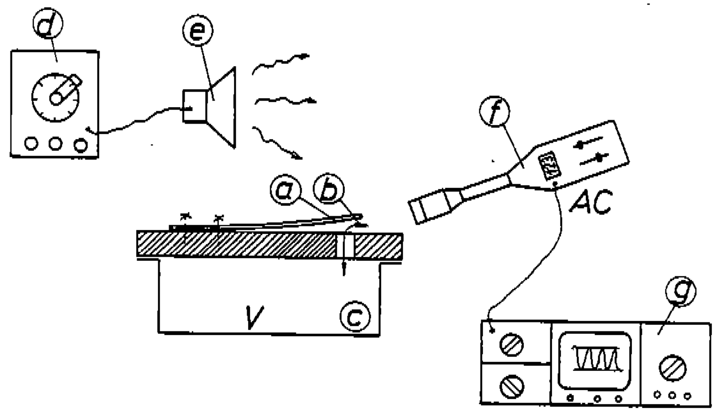


Fig.3 Set up for an acoustic resonance experiment to determine the gas inertia parameter J of the valve. a valve reed(fixed), b acoustic resonance vibration, c resonator volume, d oscillator, e loudspeaker, f sound level meter, g digital storage oscilloskope.

distance(Helmholtz-resonator with 2 free ends) the resonancefrequency is increased by about 5% only.

The method is based on the identification of the "acoustic mass" in a Helmholtz-resonator type experiment with the gas mass responsible for the inertia effect in non steady valve flow. This identification may cause a small error but generally will give good results, even for complicated arrangements. J for the model in Fig.1 has been determined by such an experiment and by calculation according to equ(3).The diference was about 6%.Experiments with reed valves also gave reasonable results.

5.THE GAS SPRING EFFECT IN COMPRESSOR ARRANGEMENTS

Let us consider an arrangement as sketched in Fig.4a,b. From thermodynamics we know that a volume displacement dV and/or a gas mass transfer dm produces a pressure increase dP as given in Fig.4a,b. To meet the conditions in a compressor we superimpose a steady (or quasisteady) flow as sketched in Fig.4c. Under subsonic conditions as usually found in compressors the boundary of the bulk exit flow behaves like a rigid body.This boundary together with the valve plate defines the exact demarcation line between volume V_1 and V_2 and so replaces the function of the piston in a configuration like Fig.4b. In a configuration like Fig.4c we have to distinguish between a pressure difference ΔP_{12} due to the steady flow and an additional pressure difference dP according to deviations of the valve plate from equilibrium position. Such deviations produce a combined displacement effect(dV) and mass transfer effect(dm).The gas spring effect is associated with the displacement dV.

Current practice of compressor process simulation accounts for the mass transfer effect but usually does not consider the gas spring effect(i.e. a fixed demarcation line between V_1 and V_2 is assumed).

To calculate the pressure change due to volume displacement alone it is assumed that the superimposed flow is not influenced by the pressure changes(i.e. $\dot{m}_{zu}=\dot{m}_{ab}$=const.). This is true exactly for flow to or from the cylinder caused by piston displacement. The flow to or from plenum chamber may be influenced to some extent by the pressure changes.

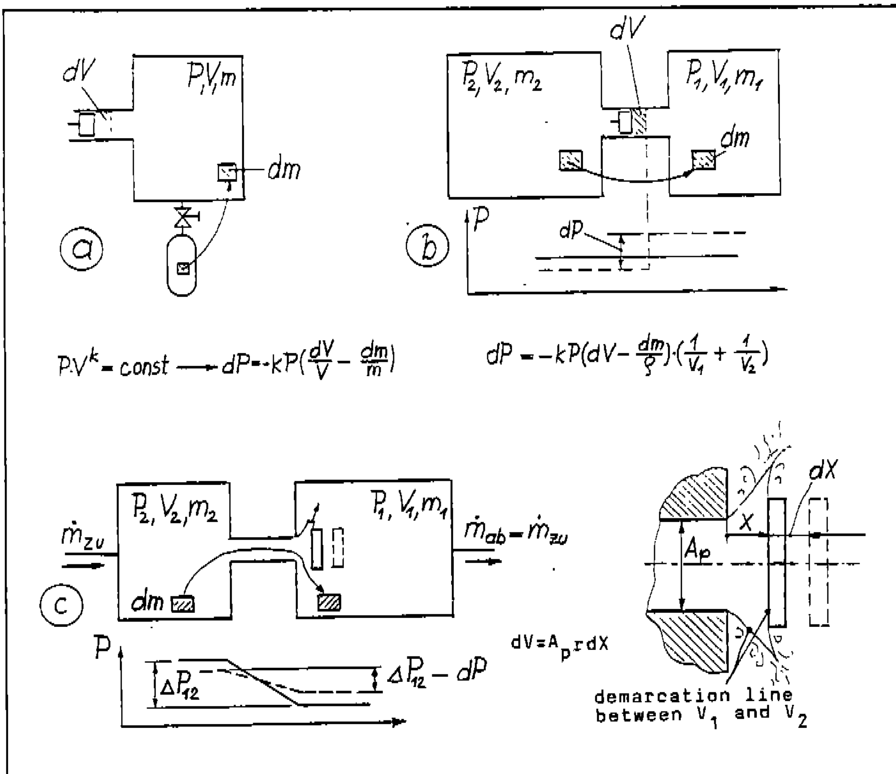


Fig.4 Isentropic pressure changes in some volume configurations.
a,b configurations with piston displacement dV and mass transfer dm;
c configuration with superimposed steady flow and with valve plate.

The pressure change due to the gas spring effect becomes

$$dP = -kP_{m12}\cdot dV\cdot\left(\frac{1}{V_1} + \frac{1}{V_2}\right) = -kP_{m12}A_p r dX\cdot\left(\frac{1}{V_1} + \frac{1}{V_2}\right) \quad \text{or} \quad \frac{dP}{dt} = -kP_{m12}A_p r\dot{X}\left(\frac{1}{V_1} + \frac{1}{V_2}\right) \qquad (13)$$

with $p_{m12} = \frac{1}{2}(p_{m1} + P_{m2})$ mean pressure

This pressure change causes an additional valve plate force which may be attributed to a "gas spring" in addition to the mechanical spring of the valve:

$$dP\cdot A_p\cdot c_p = kP_{m12}A_p r\left(\frac{1}{V_1} + \frac{1}{V_2}\right)A_p c_p dX = \Delta c . dX \quad \text{or}$$

$$\boxed{\Delta c = kP_{m12}A_p^2 r\left(\frac{1}{V_1} + \frac{1}{V_2}\right)} \quad \text{gas spring constant} \qquad (13a)$$

The correction coefficient $r>1$ may be estimated by formulas given in Fig.5, which is backed by theoretical reasoning and by experiments [1]. In dynamic processes like flutter the gas spring acts like a real spring. Near the top dead center Δc often exceeds c(=spring constant of mechanical spring).

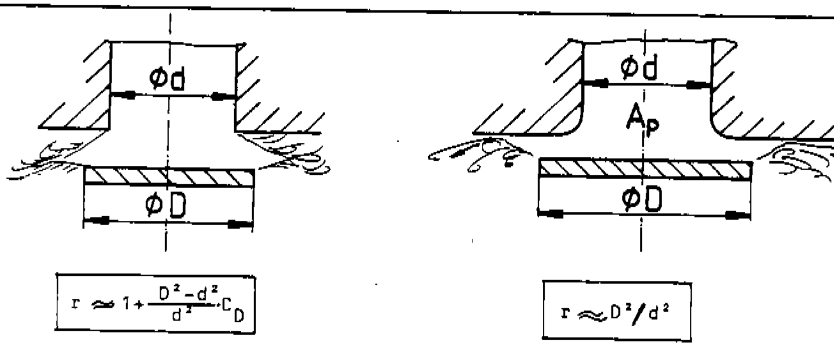


Fig.5 Estimation of correction coefficient r

It was found that the gas spring constant Δc is one of the essential quantities which govern valve flutter phenomena. If computer simulation of compressor process shall include flutter phenomena, the gas spring effect has to be included into the basic simulation equations. This can be done easily as indicated in Fig.6: when calculating pressures P_1 and P_2, instead of the traditionally defined values V_1, V_2, modified values according to the displacement effect($\pm A_p rX$) have to be used.

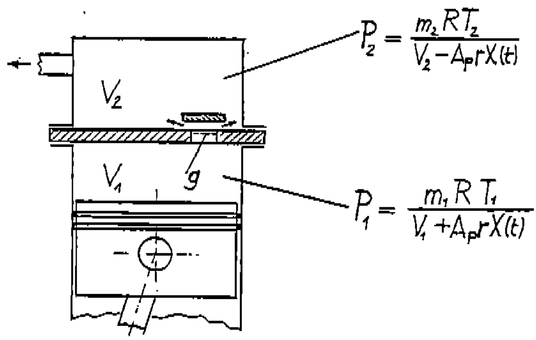


Fig.6 Taking into account the gas spring effect in computer simulation. V_1, V_2 values of cylinder volume and plenum chamber volume with respect to traditional demarcation line g.

In the first instance one may be astonished about the effects of such a small correction, but one has to bear in mind that this correction produces force pulses in the same rhythm as the plate moves. thus they may act like a resonance force or damp out oscillations depending on phase shift.

Finally let us discuss why traditional computer simulation, using quasi steady flow concepts and ignoring gas spring effect, is relatively successful, though this approach is adequate for compressors with moderate speed and without valve flutter only. Computer calculations and theoretical reasoning lead to the following explanation: the dm-transfer effect, mentioned in connection with Fig.4c, is in principle included in the traditional computer simulation model. While this effect stimulates flutterlike plate movement, non steady flow effects and gas spring effect normally act in a way which calms down this flutter. The traditional damping force($\sim\dot{X}$) acts in the same way. Now if one compares computer results and experimental results there is an easy way to harmonize them: to adjust the damping constant. This substitutes the non steady effects and the gas spring effect, but only under very specific conditions (i.e. locally). The gas spring effect e.g. depends largely on piston position and therefore its damping effect on flutter phenomena varies also largely. A damping force proportionate to valve plate velocity $\dot{X}$ only could not catch this effect adequately. Only local and rough aproximations are possible by this method.

6. DAMPING FORCE ON VALVE REED DUE TO GAS SQUEEZING EFFECT

There is another non steady flow phenomenon with valves: the gas in the gap between reed and seat plate is squeezed and sucked in periodically. This is a dissipative effect and results in valve plate damping. Fig.7 shows a basic configuration and the pressure distribution during squeezing period. To estimate squeezing pressures and forces, results from tribology may be used. Results show that squeezing forces are proportionate to b^3, b being the width of the squeezing strip. Hence the squeezing pressure along the sealing ring with a width of about 1mm is negligible when compared with the pressure under the broad reed towards the clamping end.

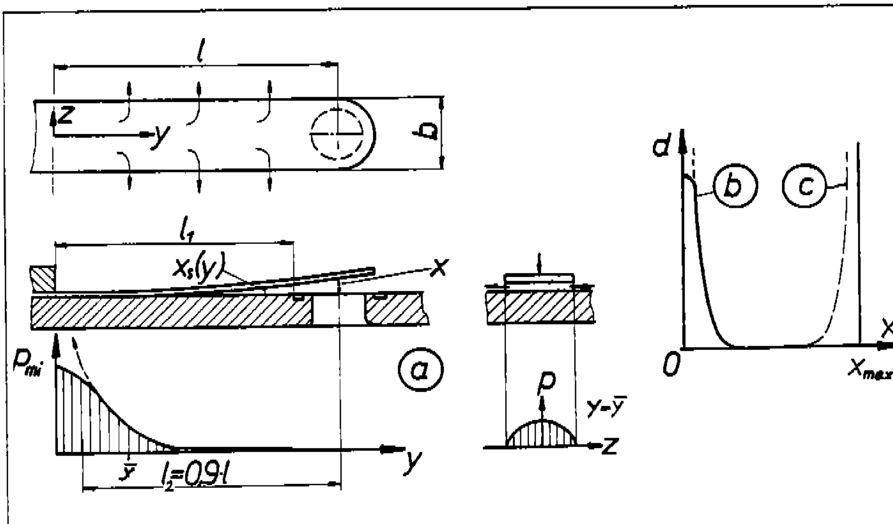


Fig.7 Gas squeezing effect. a squeezing pressure along y(left) resp. along z=const(right) at a certain time; b damping constant d due to gap reed-seat plate; c d due to gap reed-backing plate(if existing).

Theoretical considerations supposing a bending line $\sim y^3$ lead to the following formula for an estimation of the reed damping force F_d, having the same dissipative power as the squeezing effect:

$$\boxed{F_d = d\dot{X} \qquad d = k_d\left(\frac{b}{X}\right)^3 \eta l_1} \tag{14}$$

d...damping constant — η...dynamic viscosity
b...reed width, Fig.7 — l_1..length, Fig.7
X...lift, Fig.7
k_d..non dimensional empirical constant ($k_d \approx 15$)

The dynamic viscosity η is almost independent of gas pressure. It comes out that the width b of the reed is of extrem importance for this effect. Such a squeezing effect exists also along a correctly curved backing plate, if used. If one calculates the damping force according to equ(14) it ppears that the values obtained are relevant for small lifts only, Fig.7b. As indicated in Fig.7a the squeezing pressure p_{mi} theoretically tends to infinity for $y \to 0$(dotted line). In reality the reed experiences elastic deformations and the pressure tends to a final value(full line). The pressure distribution near the clamping end acts in the same way as a shortening of the free reed length($l \to l_2$) and causes an increase of the natural frequency of the reed. By measurements this increase was found to be about 25% corresponding to a reduction of the free reed length l by some 10%(l_2=0,9.l). Obviously the damping force stimulates higher modes of bending vibrations.

7.CONCLUSIONS

- At the expense of a slightly increased effort it is possible to replace quasisteady flow models for valve flow by a simplified non steady flow model. This model accounts for gas inertia and non steady work exchange between flow and valve plate.
- A socalled "gas spring effect" was found to be essential for flutter phenomena and this effect should also be included in simulation models.
- The calculation of the damping force on a reed valve may be based on the squeezing flow in the gap between reed and seat plate.
- The proposed improvements are found to be essential for high speed compressor simulation and for a basic understanding of flutter phenomena.

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