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Valve Flow Experiments with Enlarged Models

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VALVE FLOW EXPERIMENTS WITH ENLARGED MODELS - PART I

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1. ABSTRACT

Rules of similarity for valve flow experiments with enlarged models are compiled. Experiments are described which had been carried out to back theoretical insights in the field of valve flow presented to the 1982 PURDUE Compressor Technology Conference.

2. RULES OF SIMILARITY FOR VALVE FLOW

Compressor valves are usually of small size so that it is difficult to make detailed investigations about local flow patterns. In this topic -as in many other fields of fluid flow application- experiments with scaled up models can be a helpful tool. This is true for both steady state and non steady state flow experiments. When designing flow experiments with scaled models rules of similarity theory must be observed carefully. Table 1 gives a brief survey on important basic cases. For notation see also Fig.1.

Case 1: Ideal fluid. The basic approach to real valve flow is achieved by applying jet flow theory of an ideal fluid to valve geometry [1]. In this case flows are always similar irrespective of model scale and velocity level. This case is of interest only in connection with theoretical flow calculations. For $s/d<0.3$ the influence of d_1/d is weak. Data for slotlike configurations are given in [1] p.48.
Case 2: Viscous fluid. According to fluid flow theory similarity with models can be achieved if Reynolds number Re for valve flow in compressor operation is identical to Reynolds number for flow in model. Re is formed as follows

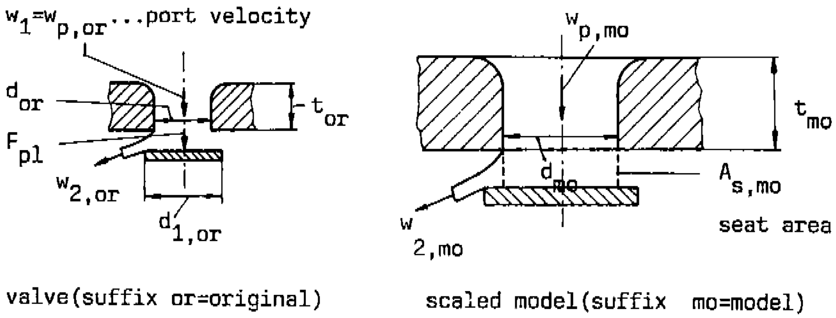


Fig.1 Similarity Of Valve And Model. Notation

$$Re = wl/\nu \qquad (1)$$

w...characteristic velocity,here preferably w_p or w_2
l...charcteristic length,here preferably t or d_p
ν...kinematic viscosity of working fluid(see Fig.2)

Having the same Re in valve and in model flow means that boundary layer thicknesses and wake shapes are in the same proportion as the model scale indicates(e.g. in a model 50 times larger than the valve,the boundary layer thickness is also 50 times larger).

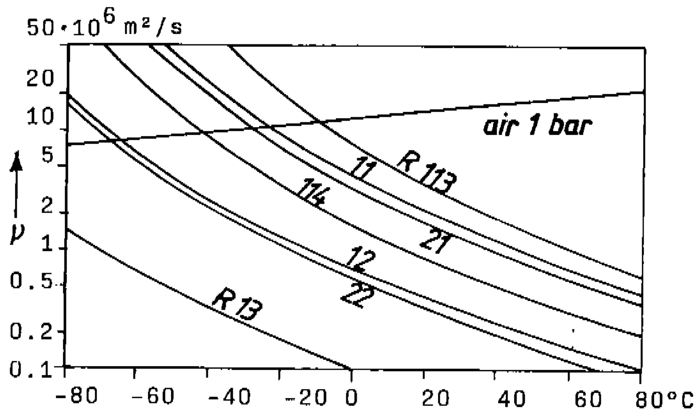


Fig.2 Kinematic Viscosity ν Of Some Refrigerants(satur. steam) and air

The correct presentation of experimental results gained with valves or with models of them should be given in non dimensional form:

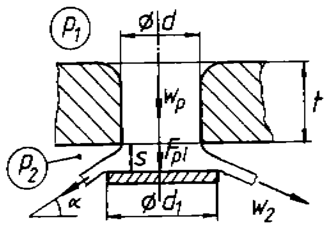
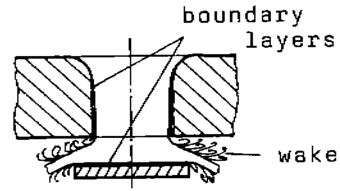
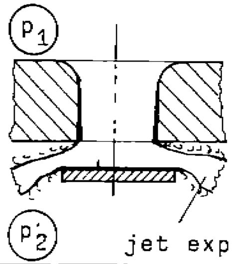
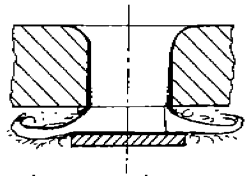
Table 1	Survey on Rules of Similarity for Valve Flow	
	Case 1	*Case 2*
flow conditions	ideal fluid, jet flow, steady state flow	fluid with friction but incompr., steady state flow
		
flow characteristics	jetlike flow, separation at sharp edges	forming of boundary layers and wakes
similarity laws	similarity for all scales and velocity levels	$Re_{or} = Re_{mo}$
nondimensional variables for discharge & force	$c_D=c_D(s/d,d_1/d)$; $c_p=c_p(s/d,d_1/d)$ relation between c_D and c_p $c_p=1+(c_D A_s/A_p)^2-2c_D A_s/A_p \sin\alpha$	$c_D=c_D(s/d,d_1/d,Re)$ $c_p=c_p(s/d,d_1/d,Re)$

Table 1 cont'd Survey on Rules of Similarity for Valve Flow		
	Case 3	*Case 4*
flow conditions	real fluid (compressible, with friction) steady state flow	real fluid non steady flow
	 jet expansion	 rapid changes in pressure difference $\Delta p=p_1-p_2$
flow characteristics	boundary layers, wakes, jet expansion, (choked flow)	delayed flow due to inertia effects, "starting vortex"
similarity laws	$Ma_{or}=Ma_{mo}$ $+(Re_{or}=Re_{mo})$ or $(\frac{p_1}{p_2})_{or}=(\frac{p_1}{p_2})_{mo}$	$Str_{or}=Str_{mo}$ $+(Re_{or}=Re_{mo})$ $+(Ma_{or}=Ma_{mo})$
nondimensional variables for discharge & force	$c_D=c_D(s/d, d_1/d, Ma, Re)$ $c_p=c_p(s/d, d_1/d, Ma, Re)$	coefficients have to be replaced by adequate equations, e.g. equ. (4.21), (5.3) in [3].

$$c_D = c_D(s/d, d_1/d, Re) \quad (2)$$ discharge coefficient

$$c_p = c_p(s/d, d_1/d, Re) \quad (3)$$ force coefficient

From c_D and c_p volume flow rate $\dot{V}$ and valve plate force F_{pl} can be calculated with the equations

$$\dot{V} = c_D \cdot A_s \cdot \sqrt{2 \cdot \Delta p / \rho} \quad (4) \qquad F_{pl} = c_p \cdot A_p \cdot \Delta p \quad (5)$$

Case 3: Viscous, compressible fluid. With very high velocities compressibility comes into play. This is quantitatively expressed by a nominal Mach number Ma formed with port velocity and a sonic speed a corresponding to stagnant upstream condition

$$Ma = w_p / a \quad (6)$$

Instead of Ma pressure ratio p_2/p_1 may be used if model fluid and compressor working fluid have the same ratio k of specific heat. Looking specially to compressibility effects, one tries to make $Ma_{or} = Ma_{mo}$. In most cases it is not possible to make at the same time $Re_{or} = Re_{mo}$. But this is of minor importance if compressibility effects are investigated.

Usually compressors must have good efficiencies which in turn means low flow velocities in valves. For this reason compressibility effects in valve flow are not of primary importance.

Case 4: Non steady valve flow. The flow process in compressor valves is inherently non steady. Non steady flow through valves is caused by rapid changes in pressure difference $\Delta p = p_1 - p_2$ across valve and/or rapid valve plate motion $x(t)$. Time enters as a new variable as well as inertia parameters [3]. Here case 4 deals with varying Δp only while changes in $x(t)$ are dealt with in case 5. According to theory of non steady fluid flow (incompr. fluid) flows in valves and in models are similar [5] if their Strouhal numbers Str are identical, Fig.3. The Strouhal number is formed as follows

$$Str = \frac{l}{w.T} \quad (7)$$

l...characteristic length, preferably inertia parameter I [3] or t

w...characteristic velocity, preferably $w_{2,max}$, Fig.3

T...characteristic time, preferably time during valve is open or valve flutter period

Large values Str correspond to highly non steady flow. Somewhat unexpected is that small velocities are connected with large non steady effects. However small velocities at the beginning and ending of the flow process in a valve -though highly non steady- do not contribute essentially to the forces acting on the valve plate and therefore it is justified to use $w_{2,max}$ to characterize non steady flow level.

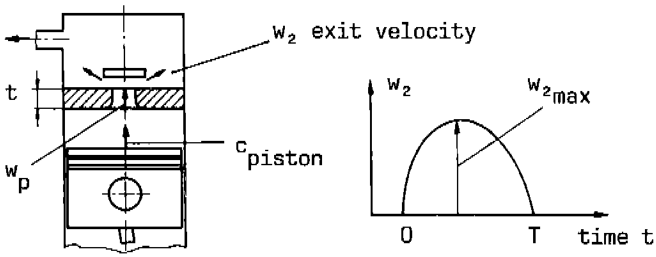


Fig.3 Strouhal Number, Notation

In most cases it is not possible to have in a model flow $Str_{mo}=Str_{or}$ and at the same time $Re_{mo}=Re_{or}$. The last condition is of minor importance if non steady effects are investigated.

Case 5: Non steady flow. Valve plate as spring-mass system. In addition to case 4 rapid changes in valve plate position x(t) take place. Similarity in this case calls for proportionality of valve period T and plate vibration period T_f. This results in

$$T_{or} : T_{mo} = (\sqrt{m/c})_{or} : (\sqrt{m/c})_{mo} \qquad (8)$$

c...spring stiffness
m...valve plate mass

Similar pressure time histories p(t) or velocity time histories $w_2(t)$ between t=0 and t=T are covered already by $Str_{or}=Str_{mo}$ (as discussed in 4).

For similarity of flow and plate dynamics the ratio of gas force to spring force for valve and model have to be equal. This results in

$$(\rho w_{2max}^2)_{or} : (cs_{max})_{or} = (\rho w_{2max}^2)_{mo} : (cs_{max})_{mo} \qquad (9)$$

ρ...density of gas
s_{max}...maximum plate lift

Usually volumes V_1 and V_2, Fig.4 are relatively small so that adiabatic compressions or/and expansions due to non steady flow take place(causing e.g. valve flutter). If a model shall cover also this phenomenon,theoretical reasoning(not presented here) lead to the following additional condition

$$\frac{\left[kp_m s_{max} T(1/V_1+1/V_2)\right]_{or}}{\left[kp_m s_{max} T(1/V_1+1/V_2)\right]_{mo}} = \frac{(\rho w_{2max})_{or}}{(\rho w_{2max})_{mo}} \qquad (10)$$

k....ratio of specific heat
p_m...mean absolute pressure in V_1 and V_2

In this model compressibility is included as far as adiabatic compressions/expansions in volumes V_1/V_2 are concerned. Compressible flow in ducts is not included.

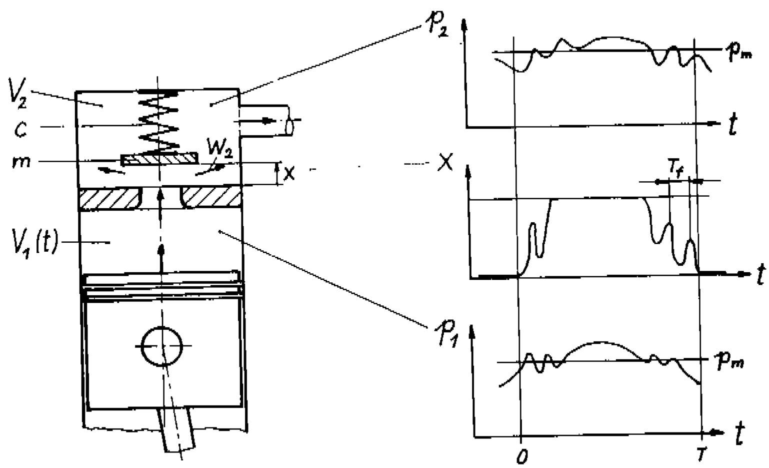


Fig.4 Case 5

Case 5 is somewhat complicated and it is not so easy to design experiments with enlarged models. This is of course due to the complexity of the process which has to be modelled. Up to now the author has carried out non steady flow experiments concerning case 4 only.

3. EXPERIMENTAL RESULTS

A small educational wind tunnel with a nozzle of circular cross section has been adapted for experiments described later. The principal set up is shown in Fig.5.

As will be discussed later it is not necessary that the nozzle entrance has the same geometrical shape as is found usually with compressor valves.- Some losses occur in a honeycomb flow straightener. To eliminate this from results stagnation pressure in point P has been used instead of pressure in point M (for steady state flow experiments).

3.1 Steady State Flow Experiments

All experiments described in Part I belong to case 2.

3.1.1 Boundary Layers

A hot wire anemometer has been used to measure boundary layer thicknesses δ. Some results are presented in Fig.6. Thickness δ of boundary layer on valve plate were in good agreement with the following formula ($Re_1 = w_p d/\nu$):

$$\delta = 0.4d/\sqrt{Re_1}$$

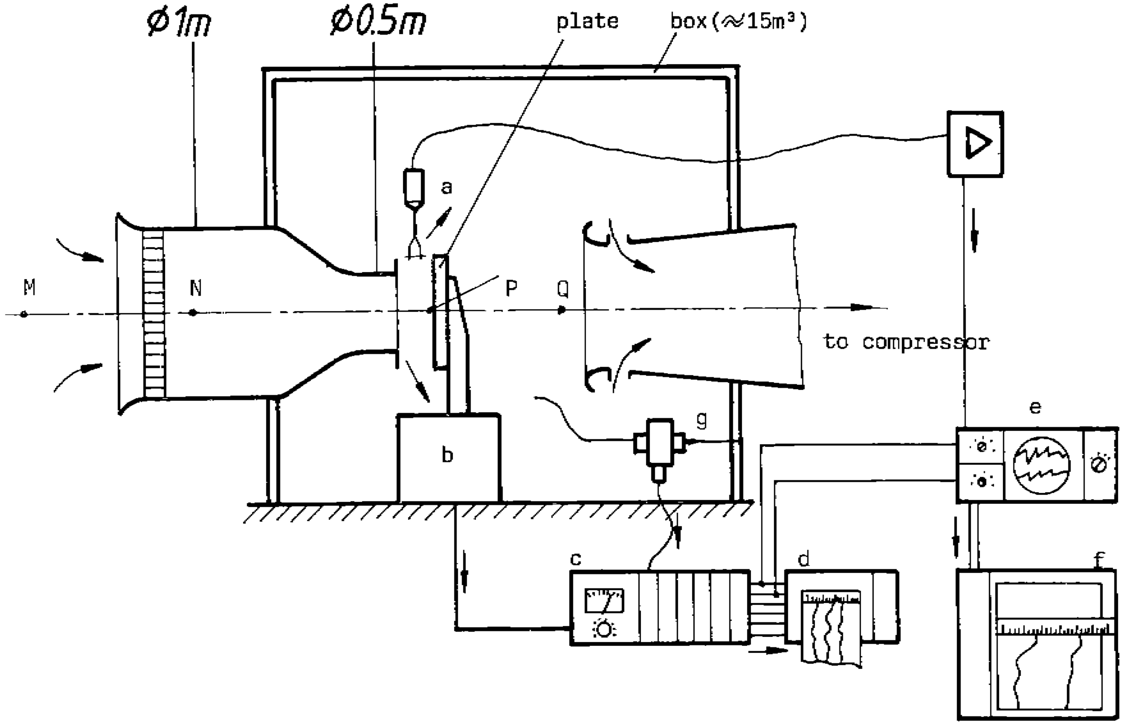
In general relative thicknesses δ/d were very small. Note that displacement thicknesses δ_1 (which characterize the real displacement effect of the layers) are about a quarter of δ only!

Reynolds numbers found in real valve operation are roughly in the same range as in this experiment. This is mainly due to extrem low velocities (compensated by large dimensions!).

From these experiments we can conclude that simple nozzle-plate configurations without internal wake regions should have very weak Re-effects.

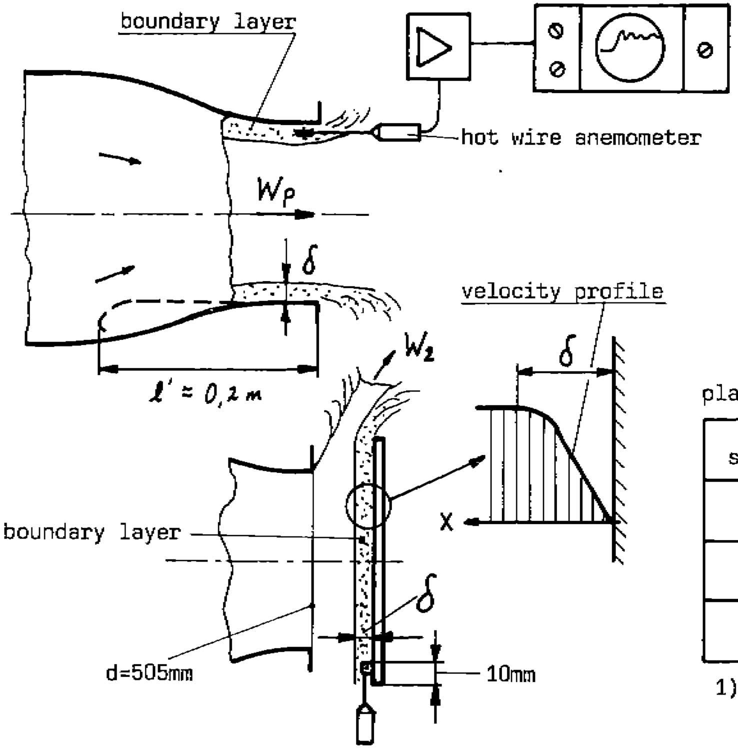
3.1.2 Pressure Distribution

Static pressure has been measured using a small static probe gliding along a radius of the plate and along the side wall of nozzle. Fig.7 gives some results including velocity profiles calculated from static pressures (confirmed by hot wire measurements in a small distance from



a hot wire probe
b electronic 6-component balance
c bridge & amplifier
d chart recorder
e digital storage oscilloscope
f x,y-recorder
g pressure difference transducer

Fig.5 Experimental Set Up



nozzle (without plate)

w_p	δ	$Re_1 = w_p d/\nu$
m/s	mm	-
0.72	8 [1)]	24.000
6.7	1.5 [2)]	225.000

plate

s/d	w_2	δ	$Re_2 = w_2 d/\nu$
-	m/s	mm	-
0.1	1.65	1.5 [1)]	23.700
0.2	0.25	3.8 [1)]	3.600

1) laminar   2) turbulent

Fig.6 Boundary Layer Thicknesses

the wall outside boundary layer). The curves show the dramatic decline of pressure(and corresponding increase of velocity) in a very small region just upstream of the seat edge. The length of this region is only about 1 times valve lift s.

Similar results for slotlike configurations based on purely theoretical calculations were presented in [1], Table 1.The fact that the increase from port velocity to exit velocity is achieved in a very small region near seat edge means also that the shape of the nozzle upstream of this region does not influence c_D and c_p considerably. Design details upstream of the exit region can influence c_D and c_p only(except for weak boundary layer effects) if losses due to sharp entrance edges with subsequent separation and reattachment of flow in the duct are present. Basic valve flow characteristics are only influenced by geometry of the exit region(which includes a space about one valve lift s upstream the seat edge,see Fig.7).

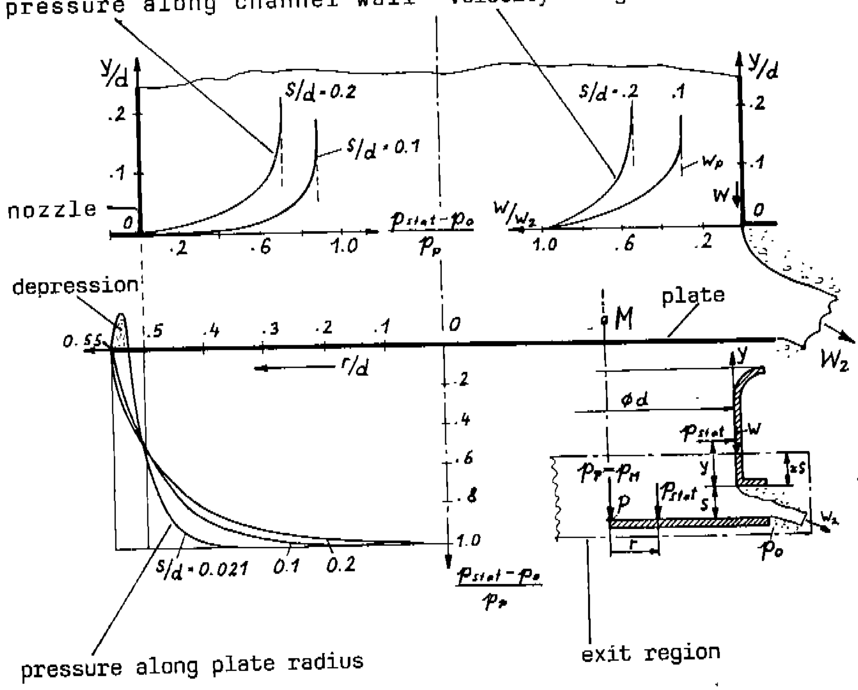


Fig.7 Distribution Of Static Pressure And Velovity

Pressure distribution along valve plate is as expected. For small lifts a depression region in the gap between plate and "land" on the opposite seat plate is developped. This is due to reattachment of flow and corresponding pressure recovery.

3.1.3 Force Coefficient c_p

The wind tunnel balance was used to measure plate force and to calculate force coefficients c_p,equ(5). To look for Reynolds number effects a curve $F_{pl}=F_{pl}(\Delta p)$ at a fixed valve lift s was plotted with an x,y-recorder,Fig. 8. After a short time of a steady run at F_{pl}=68N and Δp=342Pa the wind tunnel compressor speed was reduced steadily to zero. Transducers with excellent linear characteristics were used. Except for random oscillations of a few percent, the plot gave a straight line beginning at zero. This means that the force coefficient c_p is constant and no Reynolds number effect could be noted. From the inclination of the plotted line c_p could be calculated to 0.99.

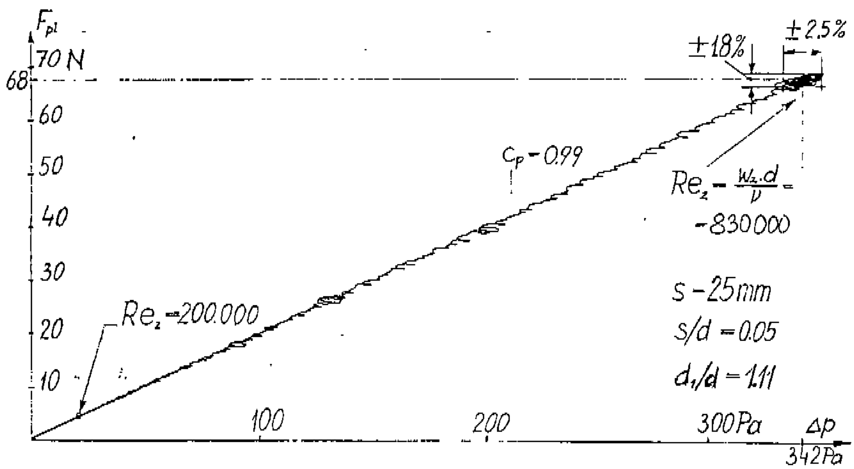


Fig.8 Record Of The Linear Relation Between F_{pl} And Δp

Fig.9 gives further c_p results as a function of relative lift s/d. Therefore, for a great range of s/d,$c_p \approx 1$ which corresponds to the theoretical jet flow result(90°-deflection of flow). This agreement is obviously due to the fact that boundary layers inside the valve are extremely thin.

Following theoretical considerations(not presented here) an approximate c_p-curve could be constructed as indicated in Fig.10(valid for rounded entrance and sharp seat edges)

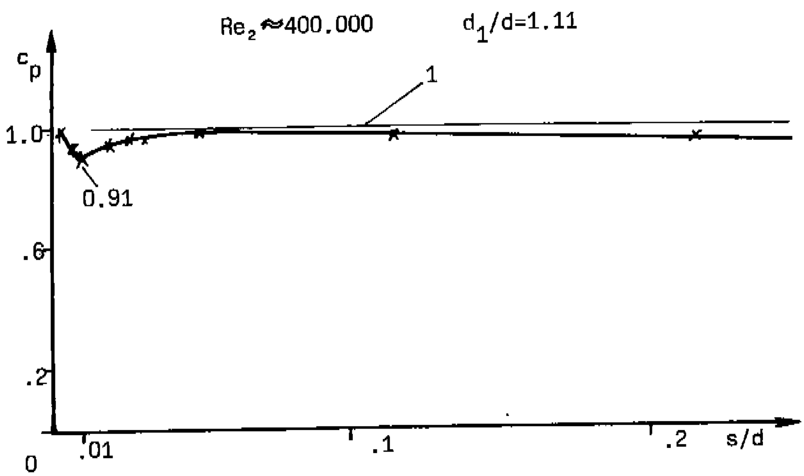
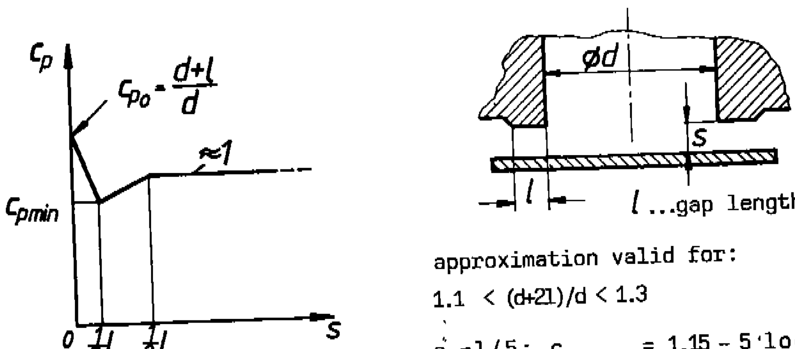


Fig.9 Force Coefficient $c_p(s/d)$ Measured With Model



approximation valid for:

$1.1 < (d+2l)/d < 1.3$

$s = l/5$: $c_{p,min} = 1.15 - 5 \cdot \log\frac{(d+2l)}{d}$

Fig.10 c_p-Curve Constructed From Geometric Data

3.1.4 Force On Inclined Plate

The experimental set up allowed without additional effort to investigate flow against inclined valve plates Fig.11.The principal findings from these experiments can be summarized as follows:

- The plate force remains normal to the plate
- The value c_p did not change significantly when

compared with parallel plate(α=0) at a lift $s=s_m$

- The point of application of F_{pl} moves from center towards the plate edge with the narrowest lift (excentricity e). Experimental data can be represented by the formula

$$e = 0.012 d \cdot \alpha^{\circ} \quad \pm 5\% \qquad (11)$$

The angle α varied between 0 and 10°; $Re_2 = w_2 d/\nu$ was about 400.000. Deviations from the formula (e increases) were found only in the case where considerable parts of the plate edge had lifts smaller than 1/2 (l=gap length, see Fig.10). Clearly this is due to partial flow reattachment.

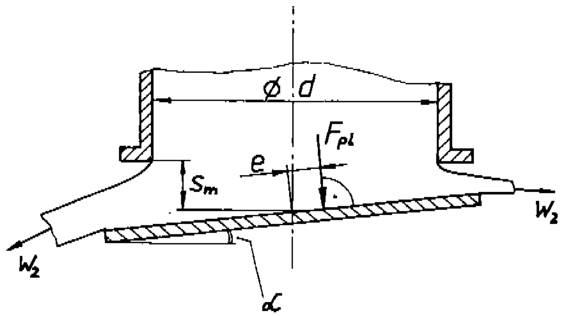


Fig.11 Flow Against Inclined Valve Plate

Equ(11) indicates that the excentricity e of the force is not dramatic but in case of a clamped reed the displacement of the force may contribute to the excitation of higher modes of flexural vibrations.

3.1.5 Turbulent Mixing Process

As has been discussed in [1] energy dissipation does not occur within the valve passage but behind the valve in a turbulent mixing process of the emerging jet.

Fig.12 shows 2 velocity traverses gained with hot wire anemometer. Traverse 1 was taken in a distance of a_1=30mm from plate(a_1/s=0.3;s=100mm) while a_2 was 150mm (a_2/s=1.5). Irregular oscillations of velocity indicate turbulent mixing while constant and smooth velocity distribution is characteristic for jet core. At a distance of a_2 the turbulent mixing process has just reached the center of the jet. For $a > a_2$ the smooth jet core disappeared.

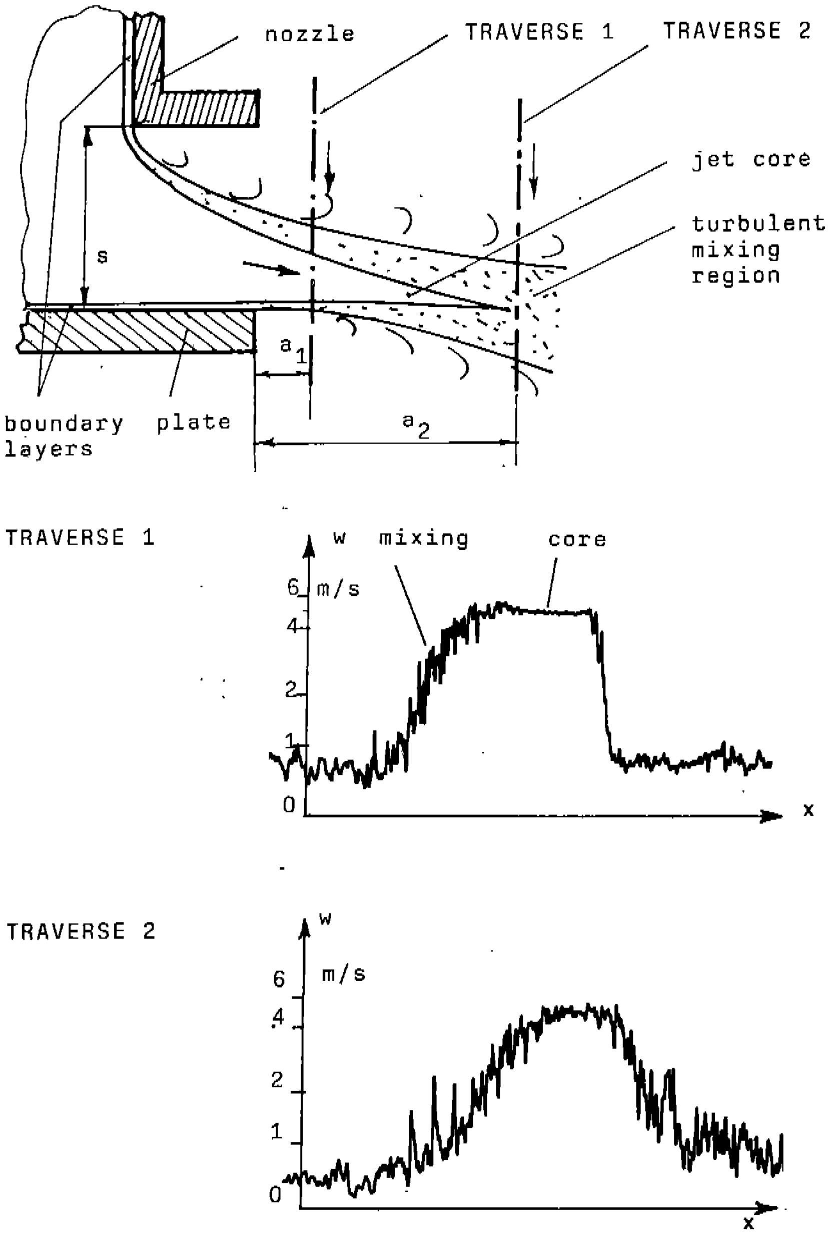


Fig. 12 Jet Mixing Process

Reynolds number Re_2 was about 160.000. Nothing changed significantly up to Re_2=500.000. For various lifts s the unafected jet core always reached to about 1.5s:

$$l_j/s = 1.5$$

l_j...length of unaffected jet core

3.1.6 Remarks On Reynolds Number Effects

Reynolds number effects on c_D could not be detected in the case of simple nozzle-plate configurations. It seems that such effects are confined to extremly low Reynolds numbers which are not of interest in the field of compressor valve flow. Experimental results reported in [2] indicate that multiring plate valves with 2 x 90 deflection of flow show a Re-effect for low Re values, Fig.13.This may be explained by the fact that these valves have internal wake regions and mixing processes

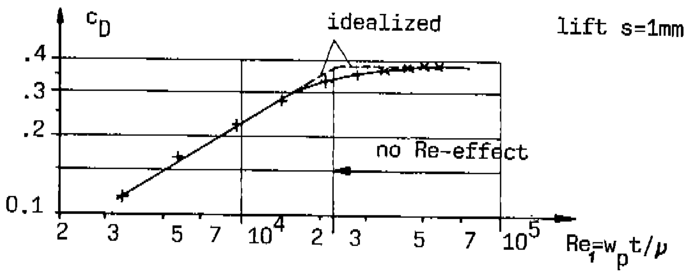


Fig.13 Reynolds Number Effect On Discharge Coefficient c_D For Multiring Plate Valve According To [2]

within the valve.

As the experimental set up described in this paper had no possibility to measure the flow rate $\dot{V}$ no discharge coefficients could be measured. As found for c_p no Re-effect is to be expected for c_D for simple nozzle-plate configurations.

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VALVE FLOW EXPERIMENTS WITH ENLARGED MODELS - PART II

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1.ABSTRACT

Non steady flow experiments are described which had been carried out to back theoretical insights in the field of valve flow presented to the 1984 Compressor Engineering Conference.

2.NON STEADY FLOW EXPERIMENTS

The principal purpose of the adaptation of the educational wind tunnel for valve flow experiments was to look for an experimental confirmation of the non steady flow model for valves presented to the 1984 Purdue Conference [3] . As the problem is very complex the simpliest case is considered where the valve plate is at constant position and non steady effects are due to rapid changes in Δp only. Flow in this case is described by equ(4.6), (4.7) in [3] . All experiments in Part II belong to case 4,Table 1.

Preliminary calculations(using Strouhal number) and experiments indicated that the installation used(see Fig. 5) could model non steady flow in compressor valves with pressure pulses of about 1 second duration and 10 Pascal peak pressure(depression pulse in the wind tunnel box). It was somewhat surprising that such pulses could be realized simply by a rapid opening push of the door of the wind tunnel box.

One of the difficulties to overcome was the presence of the frictional losses in the honeycomb type flow straightener(because the flow model to be checked assumes frictionless flow). This was achieved by using the pressure difference p_1-p_2 between points N(instead of M) and Q,Fig.5.($p_2=p_Q$ pressure inside the wind tunnel box; exit opening to compressor shut). Calculated inertia parameters of nozzle I, I_2 and equations (4.6),(4.7) were slightly modified to match this special situation. Fig.14 shows inertia parameters I, I_2 as calculated from geometrical data of nozzle form and plate dimensions.

Fig.15 shows a typical record(direct plot on manuscript sheet) of a depression pulse $\Delta p(t)$ and the induced valve plate force pulse $F_{pl}(t)$ as measured with the electronic balance. The transducers and the whole experimental set up had a sufficient fast time response to catch all non steady effects.

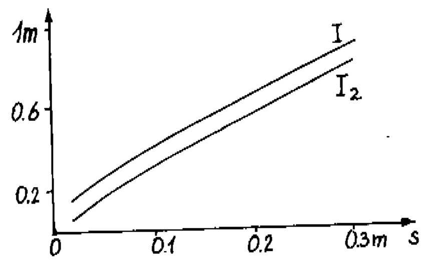


Fig.14 Inertia Parameters Of Wind Tunnel Nozzle - Plate Configuration

Two facts are remarkable:

- The peak of the force pulse is delayed by a certain time Δt with respect to the peak of the depression pulse
- The duration of the induced force pulse is considerably greater than the duration of the pressure pulse

The following procedure was adopted to check the author's non steady valve flow model: The measured depression pulse $\Delta p(t)$ was used to calculate non steady exit velocity $w_2(t)$ and non steady force pulse $F_{pl}(t)$, equ(2.4),(4.7), (5.3) in [3] . This calculated force pulse was than compared with the measured one. Fig.16 shows the result. In Fig.16 included is the force pulse calculated with steady state flow model, where F_{pl} is simply proportional to $\Delta p (F_{pl} = A_p \cdot c_p \cdot \Delta p)$.

It comes out that the non steady flow model gives a good approach to the experimental curve while steady state flow model gives considerably greater peaks. As the non steady flow equations are well established since long time the agreement confirmes mainly the simplifying assumptions introduced by the author in his model. A time increment of 0.1 s was used in calculations which means that these calculations were of relative rough nature.

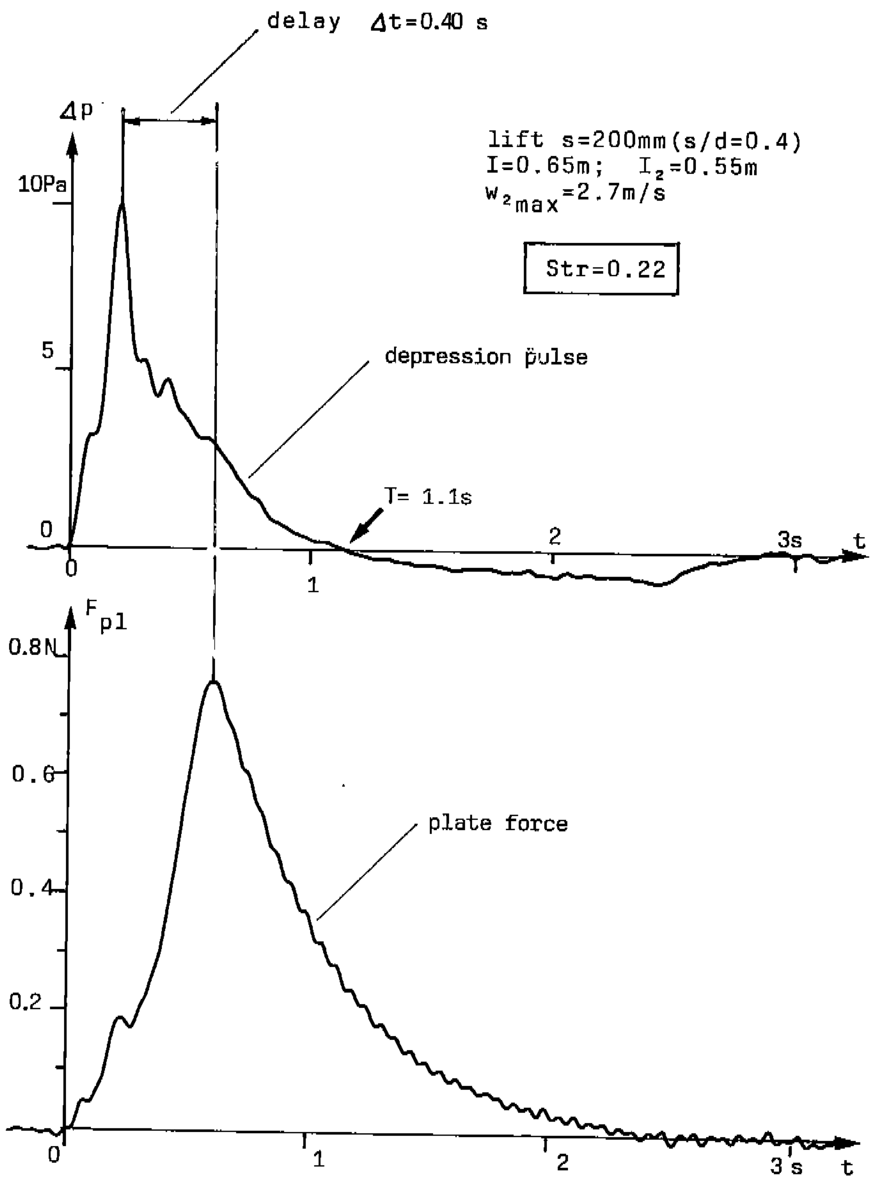


Fig.15 Response Of Plate Force On A Sharpe Depression Pulse

Maximum calculated exit velocity was 2.7m/s. This value is used to form Str and Re:

$$Str = \frac{I}{w_{2max} \cdot T} = \frac{0.65}{2.7 \cdot 1.1} = 0.22$$

$$Re_2 = w_{2max} d/\nu = 2.7 \cdot 0.505/15 \cdot 10^{-6} = 91.000$$

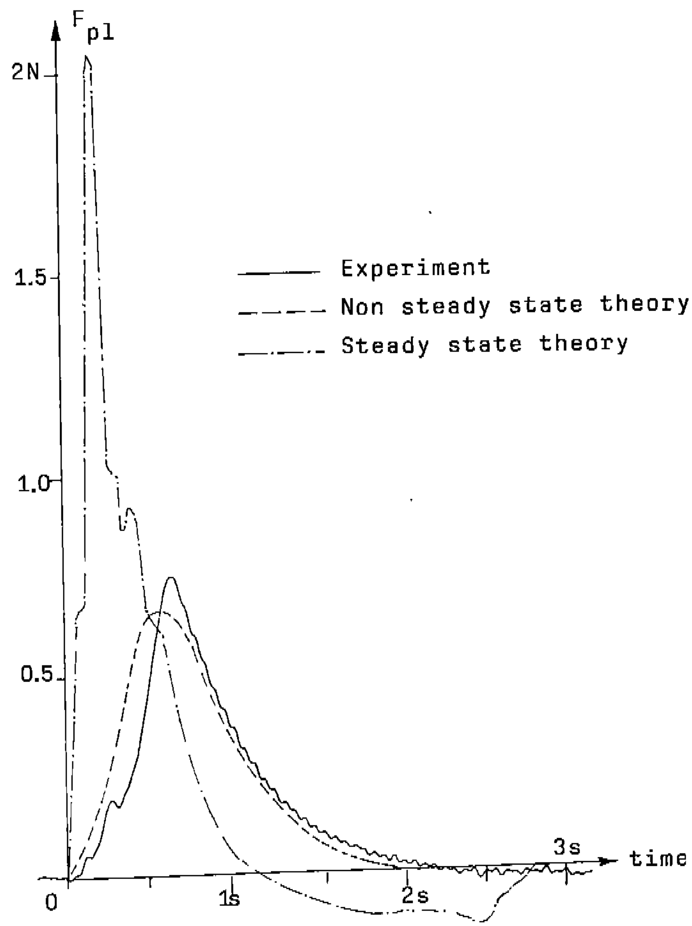


Fig.16 Force Pulse According To Fig.15.Comparison Of Experiment,Steady State Theory And Non Steady State Theory

Str is the non dimensional quantity expressing the extent of non steady flow. The experiment indicates that,if we have $Str \approx 0.22$ in a compressor valve, the extent of non steady flow effects(e.g. time delay) will be the same as in model experiment.This will be discussed later.

Concerning the Reynolds number Re_2 it should be noted that Re_2 was in a range which could be found very often in compressor valve operation. But this is not crucial for our experiment because non steady effects were investigated.

A relative great lift s=200mm(s/d=0.4) with corresponding great inertia parameters I, I_2(Fig.14) was used for the reported experiment. Fig.17 shows results from an experiment with a lift s=50mm(s/d=0.1). The depression

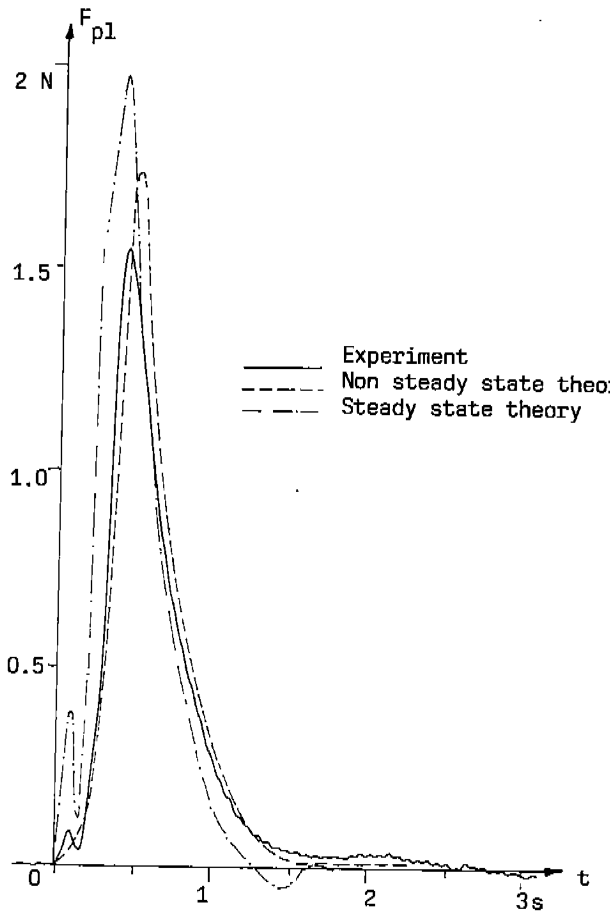


Fig.17 Force Pulse For A Lift Of s=50mm(s/d=0.1)

pulse is not shown in Fig.17. This information is easily derived from calculated force pulse according to steady state flow model, because $\Delta p(t)$ is proportional to $F_{pl,st.st}$

There is nearly no time delay between depression- and force pulse in this experiment. Here also the steady state flow model gives a good approach.

A series of experiments has been carried out to find a general relation between Str and time delay Δt (non dimensional variable: $\Delta t/T$) and a relation between Str and a socalled reduction factor for the force peak:

$$r = \frac{\text{peak of non steady force pulse(experiment)}}{\text{peak of calculated force pulse acc. to steady state model}}$$

Fig.18 gives the results. The scatter is partially caused by the fact that each depression pulse had its individual shape. The peak value is only a rough overall characterization of the pulse. From Fig.18 we may derive:

Str < 0.05	gas inertia effects negligible

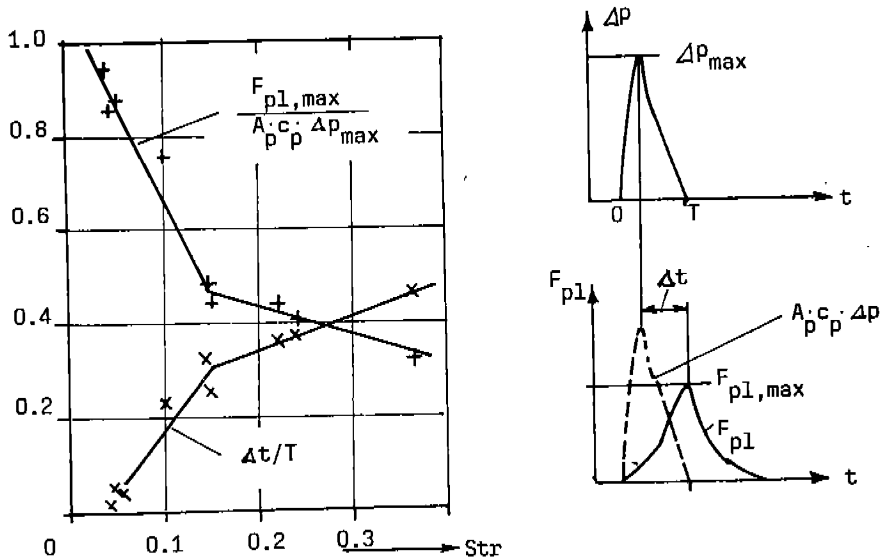


Fig.18 Non Steady Force Pulses. Time delay And Reduction Of Peak

It must be noted that this statement is valid only if Str is formed with inertia parameter I as a characteristic length. If one looks for correct modelling only, principally Str can be formed with any length of valve or model e.g. seat plate thickness t. But if we form Str with I we may use the plot of experimental results in Fig.18 in a very general manner, irrespective of valve lifts and special channel shapes.

Annother experiment was carried out in which the response of exit velocity $w_2(t)$ on a depression pulse was plotted, Fig.19. A comparison between experimental and theoretical curves confirms again the author's non steady flow model. Note that velocity scale is not linear (hot wire anemometer).

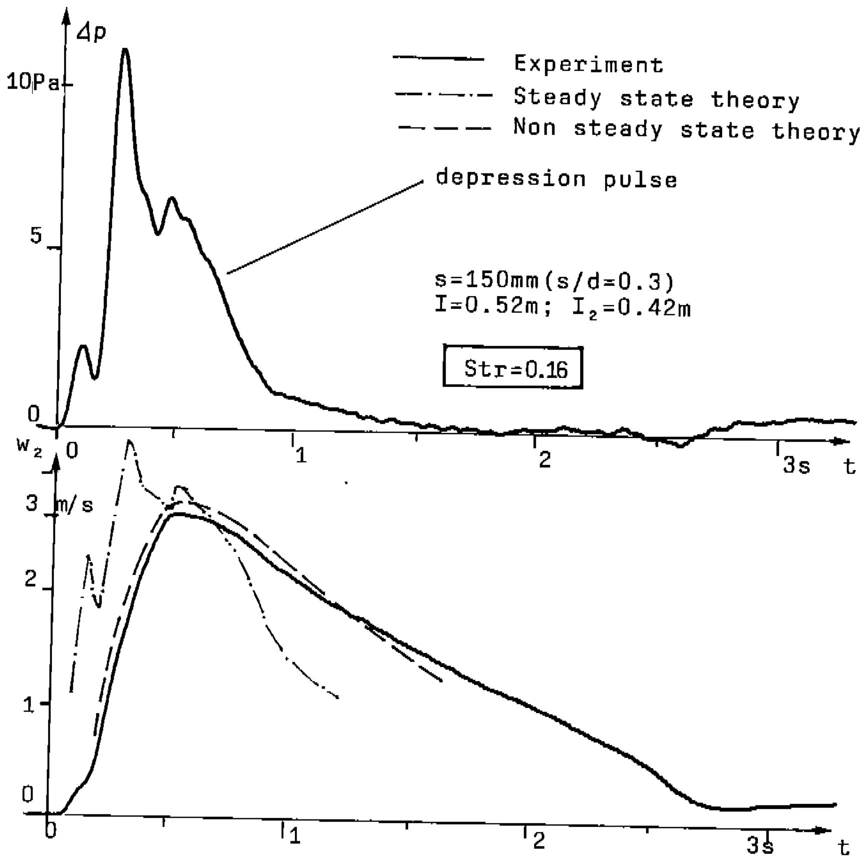


Fig.19 Response Of Exit Velocity On A Depression Pulse

3. DISCUSSION OF NON STEADY FLOW EFFECTS

Speaking in a somewhat simpified manner non steady flow effects could be explained as follows:
A sharp pressure pulse doesnot charge instantaneously the valve plate but is used at first to accelerate the gas in the valve passage. Therefore a sharp pressure pulse acts with a certain time delay and with a reduced amplitude on the valve plate.

The phenomenon that time delay and reduction in peak increase with increasing lift, Fig.18, may be explained as follows: With small lifts high velocity regions are restricted to a small volume near seat plate edge ;Fig.20 a. For the acceleration of a small gas mass only a small portion of momentum is necessary leaving the rest of the pressure pulse acting on the valve plate. With increasing lift, acceleration of gas mass in the whole duct becomes more and more important.

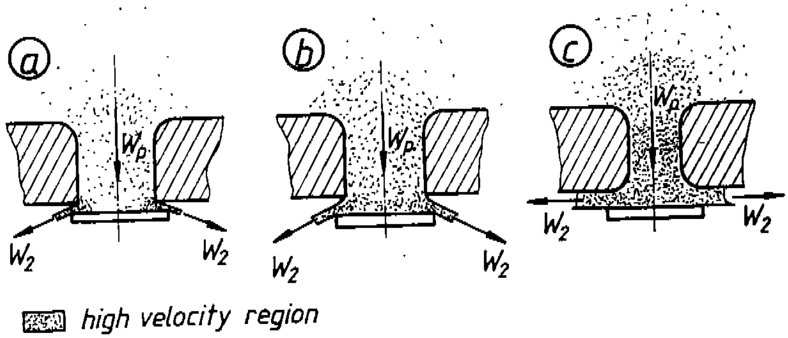


Fig.20 Valve With (a)Low, (b) High Gas Inertia Effect

Following these arguments one could imagine a valve for high speed compressors with a valve passage designed specially for high inertia effects. This could probably milder the consequences of rapid pressure changes on valve plate dynamics(impact velocities,flutter). The time delay effect doesnot necessary mean that backflow occurs because the inertia of the gas is capable of pushing through gas also against negative pressure differences.

The design of a passage with high inertia effects could be based on the following guide line: The velocity along a representative mean stream line in the valve passage should be nearly as great(or even greater) as exit velocity w_2 thus producing a great "high velocity volume",Fig.20c.This leads to a design with rounded seat edges and small port diameters,Fig.21. The losses may increase with such designs but not considerably.

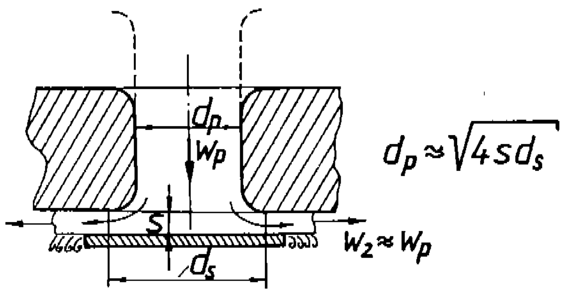


Fig.21 Valve Passage Design With High Gas Inertia Effect

Now let us apply the results from model experiments, Fig.18, to estimate the amount of non steady flow effect in a compressor valve. Fig.22 compares a typical passage design "a" which leads to low values I and a design "b" according to Fig.21 with a high value I. In case of "b" $w_p \approx w_2$ and I is about the length of a representative mean stream line in the passage. Pulse duration corresponds to 45°c.a. at a speed of 3600 min^{-1}. Comparing Str with Fig.18 tells us that in case of design "a" non steady effects are negligible while with design "b" considerable time delay and force peak reductions are to be expected. Additional non steady effects will occur due to rapid valve plate motion but this is not discussed in this paper.

4.CONCLUSIONS

- Enlarged models are an excellent tool for investigating details of valve flow. Similarity theory and relevant non dimensional quantities are the rational basis for such experiments.
- The non steady flow model for valve flow proposed by the author has been confirmed by such experiments.
- The valve designer should be aware of the possibility to design a valve passage with high or with low gas inertia effect.

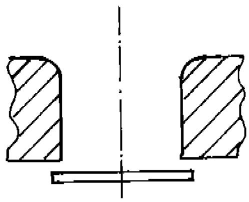
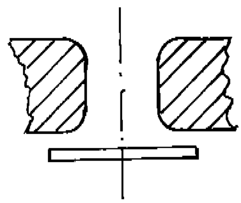
	Design "a" (Fig.1)	Design "b" (Fig.21)
		
s mm	1	1
d_s mm	10	10
d_p mm	10	6.5
d_1 mm	12	12
I mm	4.1	12.5
w_{2max} m/s	60	60
T ms	2.1	2.1
Str -	0.033	0.10
$\Delta t/T$ -	≈ 0	0.18
r -	≈ 1	0.65

Fig.22 Comparison Of Two Different Valve Designs