Aerodynamic Forces Acting on Valve Discs

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INTRODUCTION

In 1826 CLEMENT and THENARD were investigating an air blower in a French forge. They tried to close off the air duct by gradually approaching a wooden disc to the flange of the exit hole. Suddenly they noticed that the force repelling the disc disappeared and the disc was violently pulled towards the flange. At a certain distance it remained hovering.

CLEMENT further pursued the observation and obtained the same results in an experiment carried out with steam. He reported about it at the Royal Academy of Science in Paris and explained the phenomenon by the compressibility of air and steam. This motivated HACHETTE to perform systematic tests with water. He thereby disproved CLEMENT's theory and found the right explanation. Today every textbook on fluid mechanics mentions the phenomenon as a good example for BERNOULLI's theory.

However, when it is necessary to more precisely define the behaviour of the forces, e.g. for compressor and pump valves, the idealized description of the textbooks turns out to be no longer adequate. STRAUBE [1] apparently was the first one to recognize the importance of friction and pressure losses and to clearly establish the correlation between force and valve clearance.

The variation of the force with the valve lift may lead to vibrations and valve flutter [2], particularly in connection with a valve spring. The phenomenon therefore is an undesirable one for the majority of valve designs. NIEDERMAYER [3], on the contrary, has tried to take advantage of it in the concept of a high speed, low mass, springless compressor valve [4]. For this objective it was necessary to more clearly understand the mechanism involved, which led to the investigation reported in this paper. Its results are, of course, generally applicable and may contribute to a better understanding of some peculiarities in valve behaviour.

PRESSURE DISTRIBUTION ON A VALVE DISC

Fig. 1 shows a valve disc with the diameter D at a distance h from the valve seat. The inlet hole has the diameter d. The valve separates two volumes "0" and "6" with different pressures p_o and p_6. The points "1" to "5" mark characteristic stations along the flowpath of the medium flowing through the valve. v_i and p_i are the local velocity and the static pressure at the point indicated by the subscript "i". The density of the medium is ρ.

The simplified pressure diagram shown below is based on an incompressible, steady state flow free of losses. Boundary layers are not considered. The static pressure p at each point then equals the total pressure p_o less the local dynamic head:

$$p_i = p_o - \rho \cdot v_i^2/2 \qquad (1)$$

Within the area of the bore the disc feels an additional pressure which is created by the momentum due to the change of flow direction between "2" and "3". It amounts to twice the dynamic head of the velocity v_2 and adds to the static pressure p_2.

The valve gap forms a radial ring diffusor, hence at the inner diameter the velocity v_3 is larger and the pressure p_3 smaller than at the outer edge "5".

The dynamic head of the discharge velocity v_5 is assumed to be dissipated completely and amounts to the total pressure drop through the valve, as all other losses are being neglected:

$$p_o - p_6 = \rho \cdot v_5^2/2 \qquad (2)$$

The upper surface of the disc is uniformly exposed to the pressure p_6.

The hatched areas of fig. 1 represent the pressure difference between both sides. Portions trying to push the disc away from the valve seat (repulsive) are being called "positive", portions trying to pull the disc towards the valve seat (attractive) are labeled "negative". The magnitude of these portions will vary with the valve travel. At a certain lift they will equal each other and leave the disc hovering.

Obviously, this condition will also be influenced by the valve dimensions. The disc diameter D in fig. 2, for example, is only slightly larger than the bore diameter d. For the same massflow as in fig. 1 this leads to a larger exit velocity v_5 and thus to a larger pressure difference $p_o - p_6$. The repulsive portion therefore always surpasses the attractive one and under no circumstances can equilibrium be achieved.

Similar considerations apply to return flow ("5" to "1"). The conditions prevailing here are described in the lower parts of the figures 1 and 2. The pressure p_6 is now larger than p_o, the difference amounting to

$$p_6 - p_o = \rho . v_1^2/2 \qquad (3)$$

In the majority of the cases in return flow the resulting force, as desired, acts in the direction of closing the valve. However, the disc with a large diameter D has again an advantage over the smaller one (compare the figures 1b and 2b).

It should be emphasized that the condition of equilibrium can never occur in ring valves. Their diameter difference is too small and the inner and outer gap of each ring counteract each other.

FORCES IN A FLOW FREE OF LOSSES

Applying the continuity equation and integrating the pressure difference across the disc over its whole area yields in an expression for the total force F. (The weight of the disc is not being considered.)

$$F = (d^2 \pi \rho v_1^2/8)\left\{1 + (d/4h)^2[1 - 2.\ln(D/d)]\right\} \qquad (4)$$

It can be converted into a dimensionless equation by dividing by the portion of the dynamic head within the bore area:

$$\phi = F/(d^2 \pi \rho v_1^2/8) = 1 + (d/4h)^2 [1 - 2.\ln(D/d)] \qquad (5)$$

As long as the ratio D/d of disc to bore diameter is smaller than 1,65, the force is always positive (repulsive) and no equilibrium is possible at all. In the special case of D/d = 1,65 the second term is zero, the force is repulsive and independent of the valve lift. For each ratio D/d larger than 1,65 there is one lift h where the force becomes zero. This equilibrium is not a stable one: small changes of the lift will create a force trying to increase this change.

Integration of the pressure distribution for return flow (subscript R) results in

$$\phi_R = F_R/(d^2 \pi \rho v_1^2/8) = 1 - 2(d/4h)^2 . \ln(D/d) \qquad (6)$$

A negative (attractive) force is here possible for all diameter ratios D/d.

The above results apply only to an ideal flow free of losses. The picture changes considerably when friction is involved.

FORCES IN FRICTIONAL FLOW

Wall friction is introduced in the usual manner by assuming it proportional to a constant coefficient of friction f, the local dynamic head, the length of the flowpath, and inversely proportional to a hydraulic diameter defined by 4 times the crossectional area divided by the circumference.

The pressure loss due to friction at the disc within the area of the diameter d can thus be approximated by the term [5]:

$$\Delta p_{2,3} = f.\rho.v_3^2.(d/4h)/8 \qquad (7)$$

Friction within the valve clearance must be introduced differentially. It would total to a pressure drop [5]:

$$\Delta p_{3,5} = f.\rho.v_1^2.(d/4h)^3.(1-d/D)/2 \qquad (8)$$

With friction, equation (5) extends to

$$\phi = 1 + (d/4h)^2 . [1 - 2.\ln(D/d)] + $$
$$+ f.(d/4h)^3.(D/d - 3/4) \qquad (9)$$

Fig. 3 shows a set of curves for various coefficients of friction f and a constant diameter ratio D/d = 2. They all start at zero clearance with a positive infinite value. This is due to the fact that the dimensionless force ϕ is being related to the velocity in the bore and thus to a constant mass flow which at zero clearance requires infinite v_5. The actual force at h=0 is not exactly defined as the pressure in the sealing area is not known.

With increasing clearance the force is rapidly diminishing and disappears at a certain clearance (example: point I for f = 0,05 in fig. 3). This equilibrium is a stable one: a small change of clearance will create a force that brings the disc back to its original position.

Considering the inertia of the disc, this situation can lead to vibrations which, in connection with a valve spring, may cause valve fluttering [2].

Further increase of the clearance brings about an attractive force that pulls the disc towards the valve seat. This attractive force reaches a maximum (point IV for f = 0,05) and than diminishes until it again becomes zero (point II for f = 0,05). This second equilibrium is labile and corresponds to the one encountered in lossfree flow (point V in the figures 3 and 4).

The range of clearance within which attractive force occurs (between points I and II for f = 0,05) becomes smaller with increasing friction. At a given friction coefficient (f = 0,074 in fig.3) the first and second equilibrium coincide with the clearance for maximum attractive force (point III in fig. 3). Higher friction no longer allows an attractive force.

A system of curves similar to fig. 3 could be drawn for all diameter ratios d/D. In order ot give an impression of the influence of the diameter ratio, the points I and II (first and second equilibrium) are selected and plotted in fig. 4.

For a diameter ratio d/D = 0.5 the points I, II, III and V correspond with the points having the same number in fig. 3. For a diameter ratio of d/D = 0,4 and a friction coefficient of f = 0,1, to give another example, the first equilibrium is reached at h/d = 0,056. An attractive force is acting on the disc between h/d = 0,056 and 0,194, at which lift the second equilibrium is reached.

The dotted line connects all points III where first and second equilibrium would coincide. For d/D = 0,4 this would occur at an interpolated friction coefficient f = 0,17. With higher friction no attractive force can occur at this diameter ratio.

The islands defined by the curves f = constant therefore indicate the area within which the force is negative (attractive) for a given friction. These islands become smaller with increasing friction and shrink to a point at f = 0,306 which is the maximum friction to allow equilibrium at all. This value, of course, is highly hypothetical and only serves the purpose of helping understand the mechanism of the phenomenon. Even in a flow free of losses no attractive force can occur above the curve f = 0 which indicates the absolute limits for the valve dimensions allowing equilibrium. Discs with dimensions beyond this curve will be repelled under all circumstances.

The points for maximum attractive force (IV) are being shown separately in fig. 5 in order not to over complicate fig. 4. The combination of both figures 4 and 5 allows to find the five significant points I, II, III, IV and V defined in fig. 3.

Return flow with friction is described by

$$\phi_R = 1 - 2(d/4h)^2 \cdot \ln(D/d) - f(d/4h)^3 \cdot (D/d - 3/4) \quad (10)$$

In this case friction increases the attractive force, extends its region and helps closing the valve more quickly. The curves in fig. 3 and 4 show this influence. The only equilibrium possible is labile and therefore can hardly be realized experimentally.

LOSSES DUE TO CHANGE OF MOMENTUM

Applying the same mathematical model the total pressure drop of the flow through the valve was calculated and compared with test results [6]. It thereby proved necessary to also consider the pressure drop caused by the change of flow direction and cross-sectional area between "2" and "3" (fig. 1). These losses are assumed to be proportional to a constant coefficient ζ. Furthermore it turned out to be beneficial to partly relate them to the dynamic head of v_2, and partly to v_3. Hence, the additional pressure drop caused by this influence is

$$\Delta p_{2,3} = \zeta_2 \varrho v_2^2/2 + \zeta_3 \varrho v_3^2/2 \quad (11)$$

The complete formula for the force can now be written for discharge flow

$$\phi = 1 + \zeta_2 + (d/4h)^2 \cdot \left[1 + \zeta_3 - 2 \ln(D/d)\right] + f \cdot (d/4h)^3 \cdot (D/d - 3/4) \quad (12)$$

and for return flow

$$\phi_R = 1 - \zeta_2 - (d/4h)^2 \cdot \left[\zeta_3 + 2 \cdot \ln(D/d)\right] - f \cdot (d/4h)^3 \cdot (D/d - 3/4) \quad (13)$$

In discharge flow ζ_2 has the same effect as a downward shift of the abscissa in fig. 3. This is indicated in fig. 6. The maximum amount of attractive force and the range of lift between the two equilibriums are thereby reduced.

The influence of ζ_3 increases with decreasing clearance. Its effect would be the same as if the abscissa of fig. 3 were shifted and bent to fit the corresponding curve ζ_3 = constant in fig. 6. In principle, the effects of ζ_2 and ζ_3 are the same and superimpose each other.

The influence on the set of curves in fig. 4 is shown in fig. 7. The curves f = 0 are shifted and bent downwards and all islands f = constant are squeezed accordingly.

The range of dimensions allowing an attractive force is consequently being reduced. For a given diameter ratio the frictional limit (point III) is also lowered as well as the absolute maximum friction still allowing equilibrium. The value f = 0,306 of fig. 4 is hence reduced and may well reach a level feasible in practical applications.

In the formula for return flow (13) ζ_2 and ζ_3 both have negative signs and therefore act favourably for rapidly closing the valve.

COMPARISON WITH TEST RESULTS

STRAUBE [1] has measured coefficients of friction and forces on a disc of a diameter ratio of D/d = 20. Losses due to friction in this case will by far surpass the losses due to change of momentum which therefore can be neglected. The following table compares STRAUBE's test results with results obtained from equations (9) and (5).

D = 200 mm d = 10 mm	Value measured by STRAUBE [1]			Calculated from (9)	
	f	h(mm)	F(N)	h(mm)	F(N)
First equilibrium (I)	0,0354	0,35	0	0,344	0
Maximum attractive force (IV)	0,06	0,67	15	0,868	14,3
Second equilibrium (II)	0	8,7 ?	0	5,59	0

Considering the very small clearances involved (0,35 mm = .014 inches!), the agreement is remarkably good except for the second equilibrium where both test and theory are questionable. As the equilibrium is labile, it is hard to be established: STRAUBE had trouble defining the point. On the other hand, the assumption of constant velocity across the lift as a basis for equation (9) is no longer adequate for clearances of this magnitude.

According to STRAUBE the coefficient of friction for a given valve seems to vary with the lift. Even though the author could achieve remarkable agreement between measured and calculated pressure drops assuming friction coefficients constant and independent of the lift [6], it is not advisable to use equations (12) and (13) for quantitative predictions of valve behavior. They are, however, perfectly suited to qualitatively describe the conditions, to clarify the influence of each parameter and to approximately predetermine the dimensions.

CONDITIONS IN ACTUAL VALVE OPERATION

In order to study the actual pressure distribution in the valve gap during compressor operation, a very small piezoelectric pressure transducer was brought close to the valve.

As an example for the results obtained fig. 8 shows the pressure distribution in the gap at various phases of the valve motion:

a) Immediately before valve opening
b) During valve opening (disc in motion)
c) Immediately after valve opening
d) Valve open, disc in resting position
e) Immediately before valve closing
f) During valve closing (disc in motion)

There is a time difference of half a millisecond between each of the first 4 phases and between the last two. Phases d) and e) are 3 milliseconds apart. Compressor speed was 2600 RPM, pressure ratio was 2.

The unsteady flow in its initial phase is also influenced by the valve motion. The pressure diagram obviously needs some time to assume a shape similar to fig. 1. The time of valve opening, therefore, is not always long enough to allow the effect to fully develop. Its main influence will be during rebouncing from the valve stop.

Valve closing occurs at top dead center of the piston. Piston speed and flow velocity through the valve are close to zero, hence the pressure difference across the valve is very small, as indicated in the phases e) and f) of fig. 8. It is therefore desirable to enhance valve closing by increasing the force by a favourable valve design.

SUMMARY

The pressure distribution around a valve disc creates an aerodynamic force. Its amount and direction depend on the valve dimensions, the lift and on the flow losses. For a sufficiently large ratio of disc to bore diameter the disc will reach a stable equilibrium at a certain lift. An increase of lift will first result in an attractive force,then lead to a second, labile, equilibrium and finally bring about a force repelling the disc from the valve seat. A simplified mathematical model is used to qualitatively describe the phenomenon, to point out the main influences and to establish the limits for the occurence of equilibrium.

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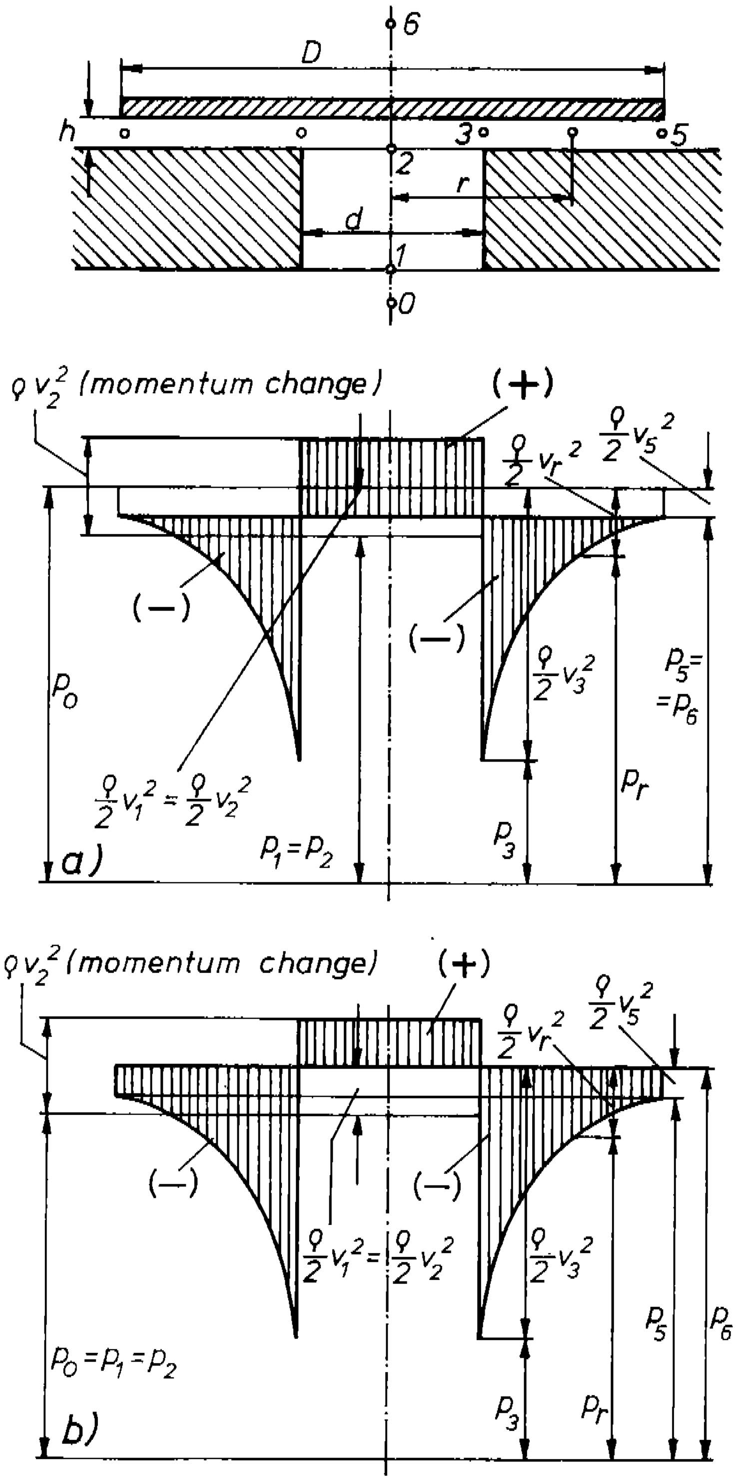


Fig.1: Valve with large diameter ratio D/d

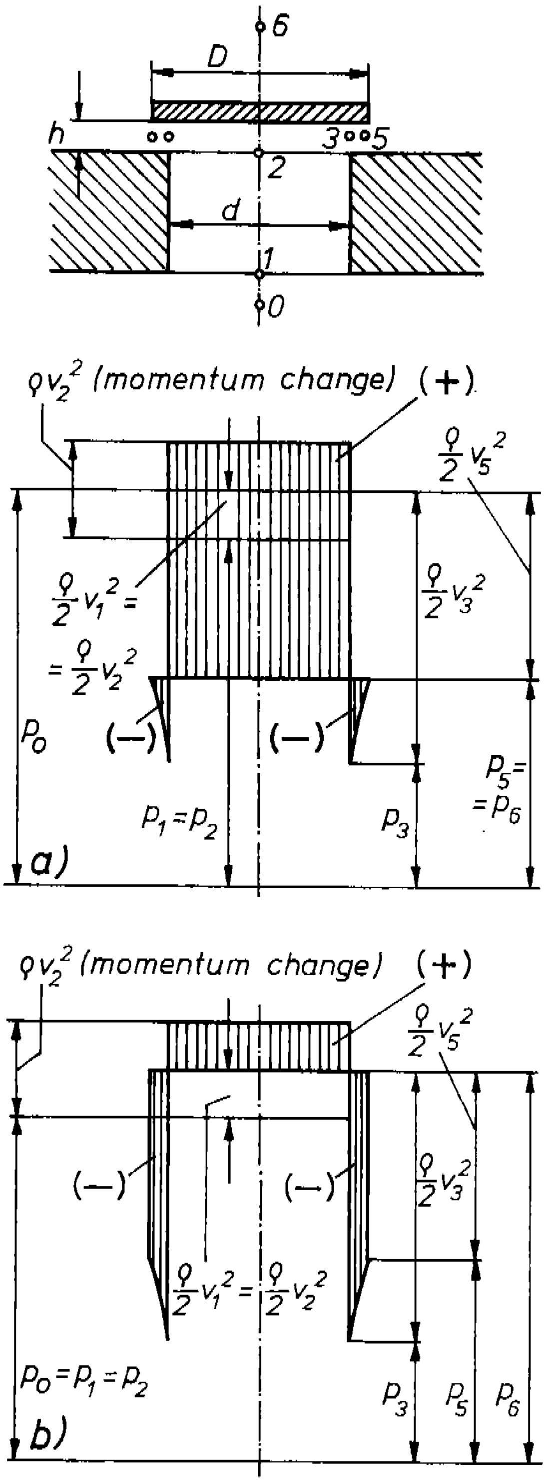


Fig. 2: Valve with small diameter ratio D/d

Idealized pressure distribution around the disc in loss-free flow.

a) Discharge flow (from "1" to "5")

b) Return flow (from "5" to "1")

(+) : Repulsive portion (pushing away from the valve seat)

(-) : Attractive portion (pulling towards the valve seat)

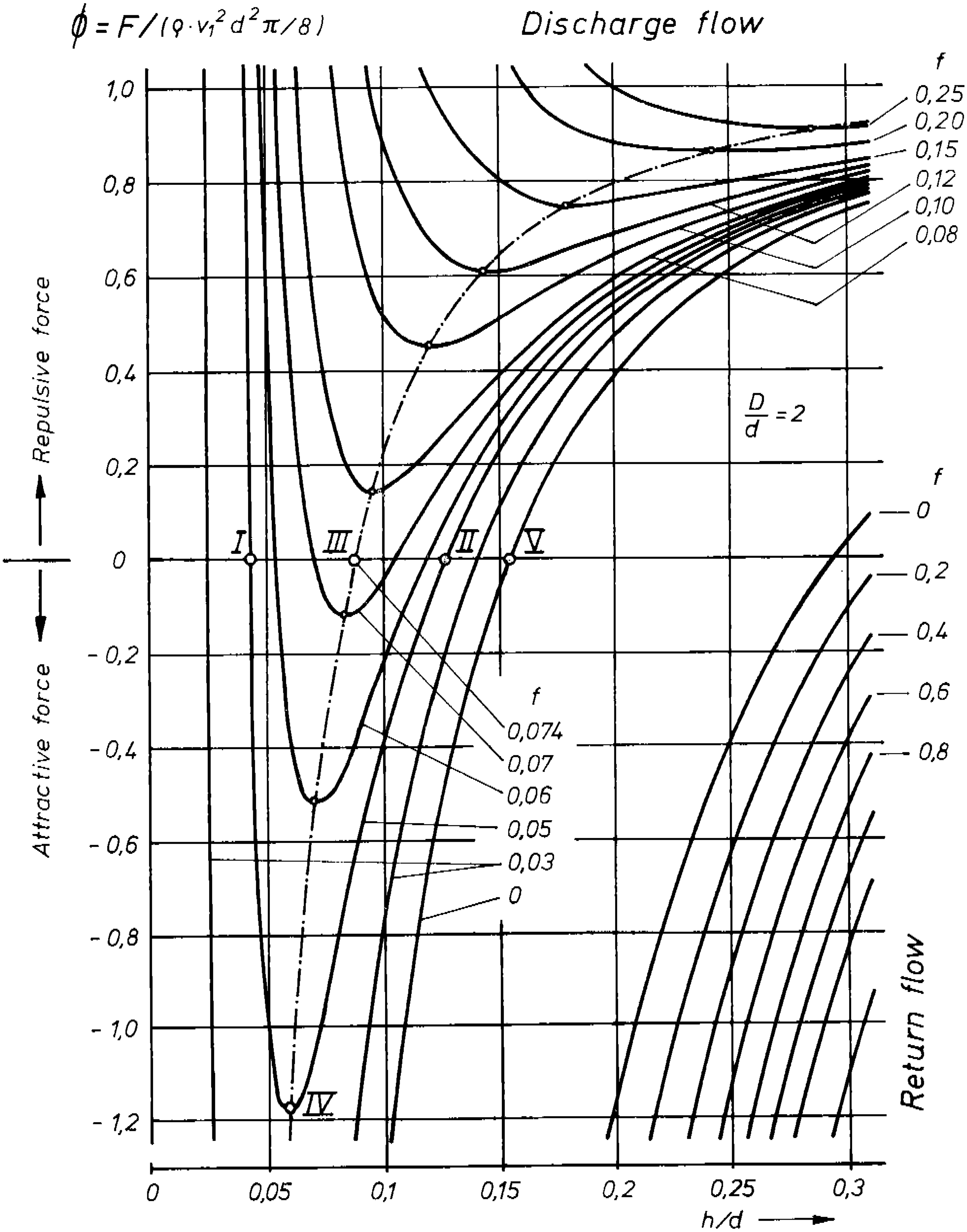
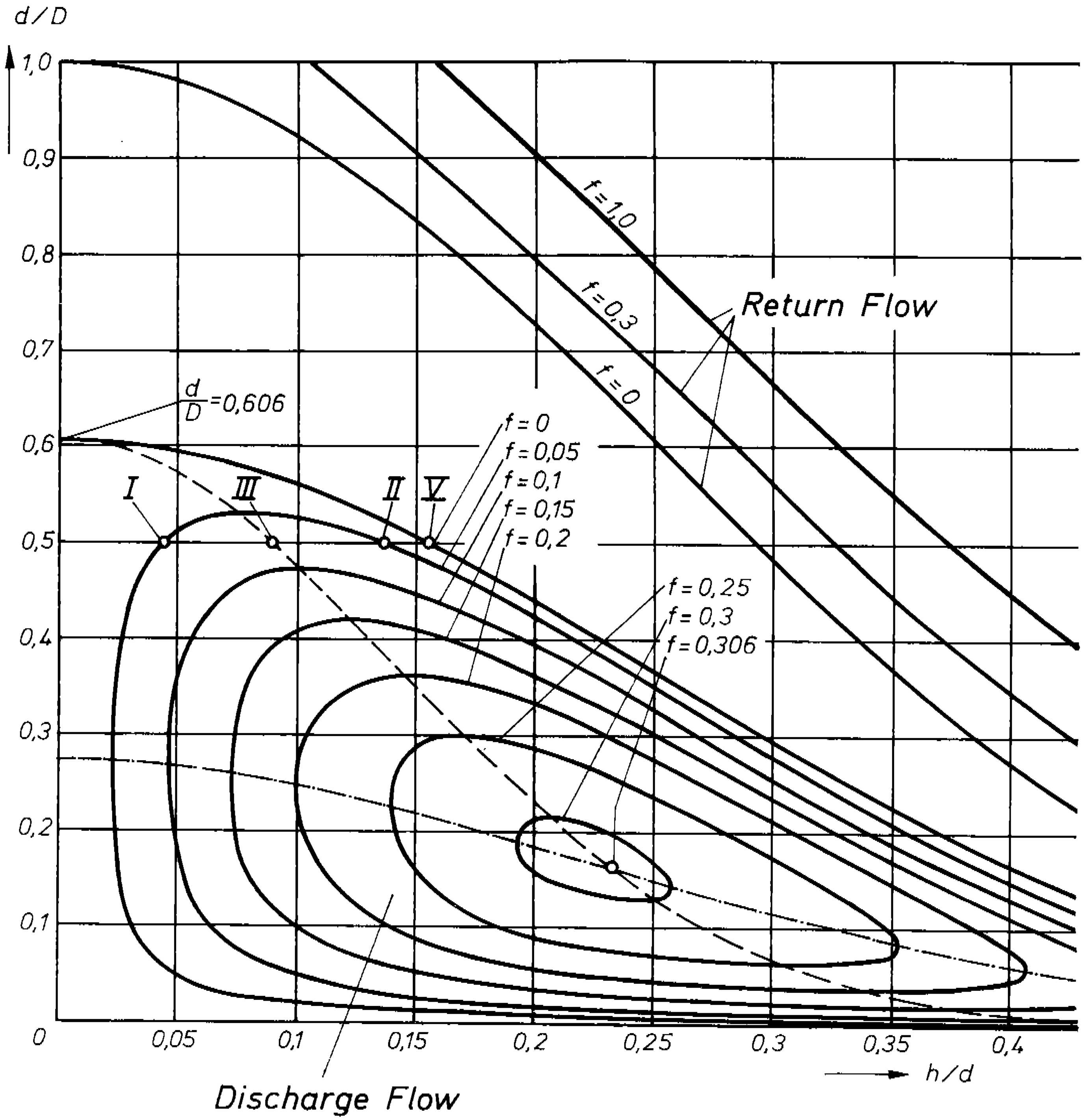


Fig. 3: Influence of valve lift h and skin friction f on the aerodynamic force acting on the disc.



——— **Dimensional conditions for equilibrium, constant friction coefficient f**

– – – **Limits for the diameter ratios d/D allowing equilibrium with a certain friction coefficient f**

—·— **Limits for the clearances h/d allowing equilibrium with a certain friction coefficient f**

Fig. 4: *Influence of the clearance h, the diameter ratio d/D and the coefficient of friction f on the force equilibrium.*

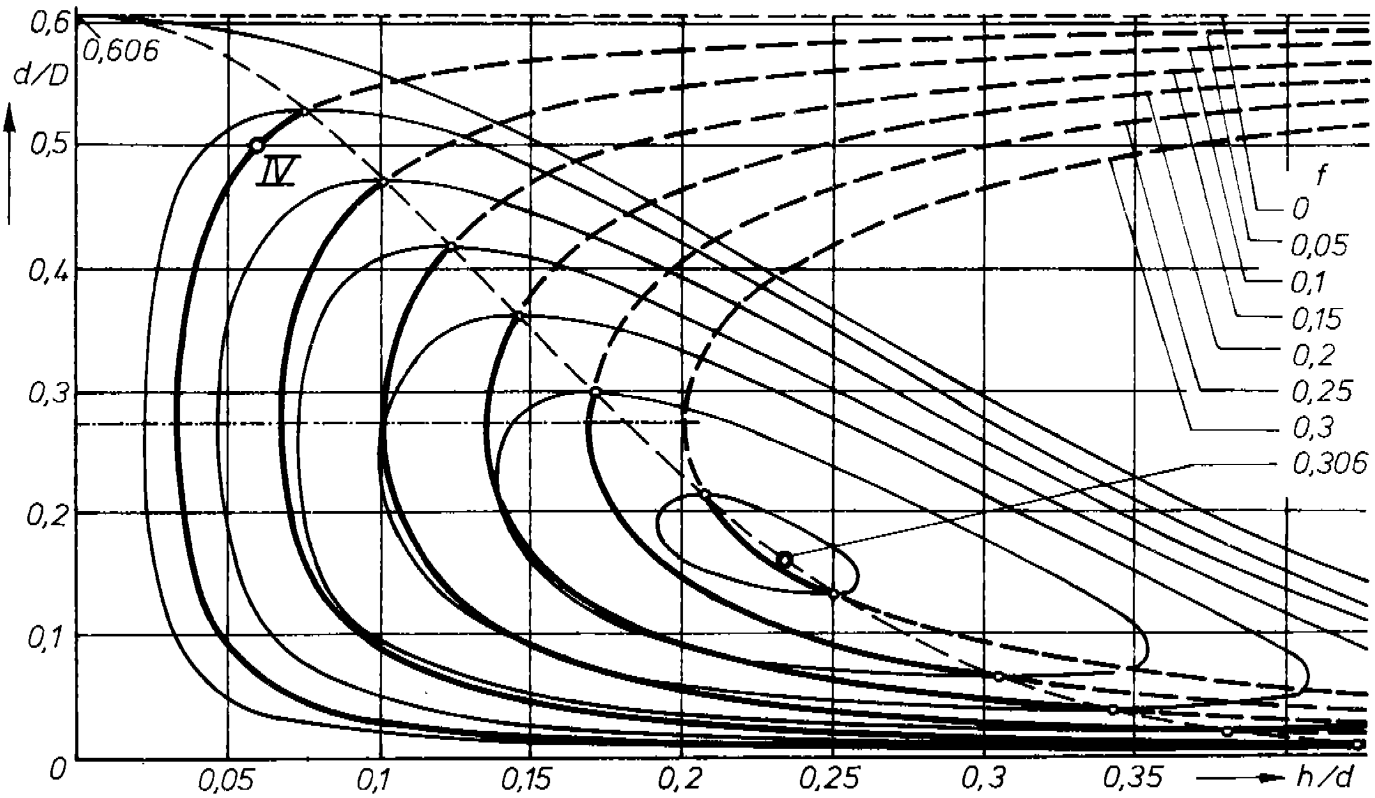


Fig. 5: Influence of clearance h, diameter ratio d/D and friction coefficient f on the maximum attractive force.

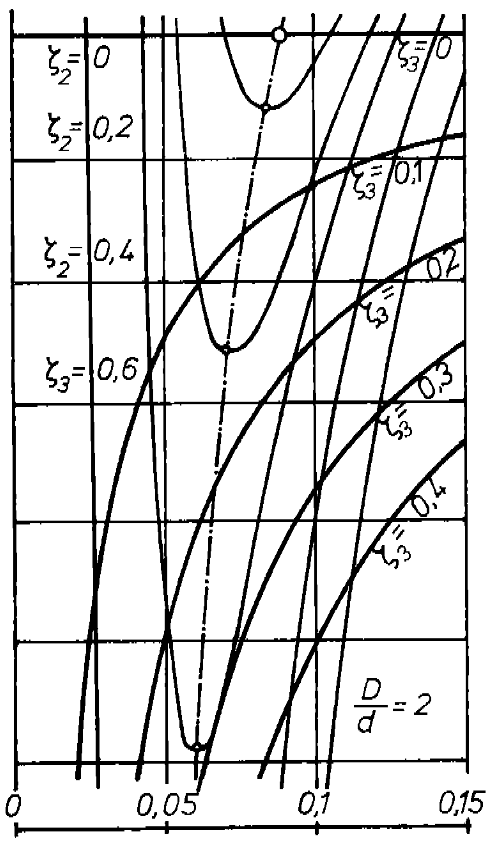


Fig. 6: Influence of ζ_2 and ζ_3 on the force in fig. 3

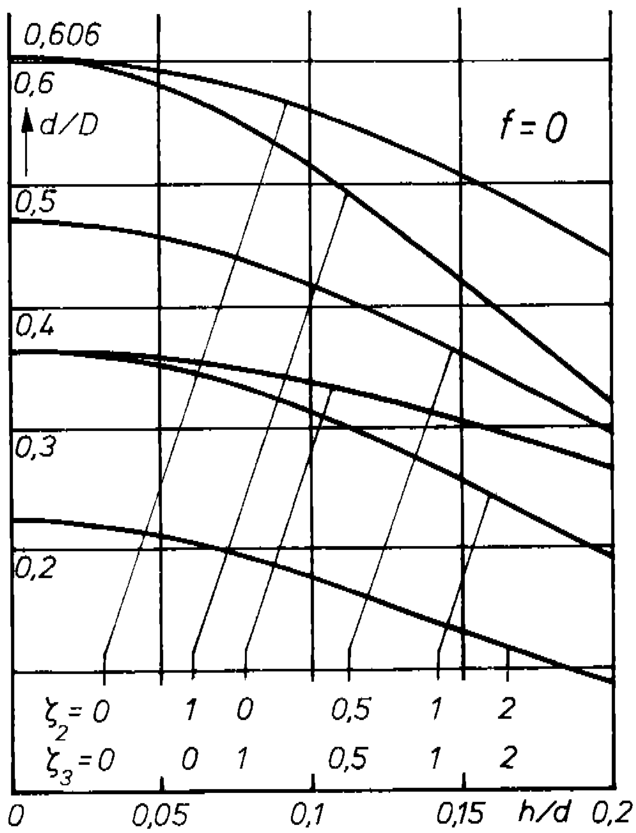


Fig. 7: Influence of ζ_2 and ζ_3 on the force equilibrium in fig. 4

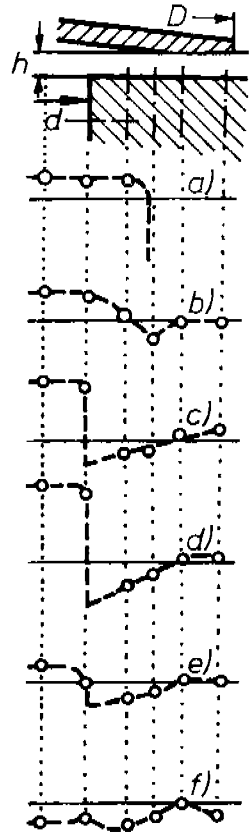


Fig. 8: Pressure distribution in valve operation