1982

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A SIMPLE BUT EFFICIENT EQUIPMENT
FOR EXPERIMENTAL DETERMINATION OF VALVE LOSS COEFFICIENTS
UNDER COMPRESSIBLE AND STEADY FLOW CONDITIONS

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1. ABSTRACT

A brief introductory survey of equations defining effective mass flow rate and loss coefficients, based on the theory of one-dimensional steady flow, is given. Then an experimental arrangement for compressor valve investigations is presented. This test set-up makes use of a supersonic wind tunnel equipment and enables one to study the flow properties of a valve over a wide range of pressure ratios. The performance of experiments is described and typical results gained from an investigation of a multi-ring plate valve are discussed in more detail.

2. NON-DIMENSIONAL COEFFICIENTS CHARACTERIZING PRESSURE LOSSES IN COMPRSSOR VALVES

A variety of non-dimensional coefficients characterizing friction effects can be found in the literature [1-4]. For sake of simplicity theoretical considerations dealing with pressure losses in compressor valves are usually based on the assumption of steady one-dimensional flow as in the theory of ducted flows [5]. While some authors use equations of compressible flow, others use equations of incompressible flow and take into consideration compressibility effects by means of suitable parameters.

Table 1 gives a survey of frequently used equations defining the effective mass flow rate as well as loss coefficients.

The so-called discharge coefficient $C_D$ is defined as the ratio of the effective mass flow rate $\dot{m}$ and the theoretical mass flow rate $\dot{m}_{th}$.

$$C_D = \frac{\dot{m}}{\dot{m}_{th}} \quad (2.1)$$

An additional subscript, I or C, (for $C_D$, $\dot{m}$ and $\dot{m}_{th}$ respectively) indicates whether the theory of incompressible or compressible flow underlies the considerations.

As reference area some authors choose the variable opening area formed by the valve plate (or reed) and the seat edge, others choose the constant geometrical flow area provided in the seat plate. The latter is used here throughout (denoted by $A$).

In the case of incompressible flow the density $\rho_1$ is equal to $\rho_{th}$ by definition. This has been taken into account in equation (2.3) already.

Further if, as in the described test set-up, the flow velocity upstream of the valve is small compared with the velocity in the valve the pressure difference $P_{th} - P_1$ can be neglected in practical calculations. By this means one obtains, according to equations (2.3) and (2.4), a simple relation between pressure loss coefficient $\bar{f}$ and discharge coefficient $C_{DI}$.

$$\bar{f} = \frac{1}{C_{DI}} \quad (2.2)$$

3. TEST SET-UP AND PERFORMANCE OF EXPERIMENTS

A small supersonic wind tunnel facility enabled compressor valve investigations both under steady flow conditions and up to high pressure ratios.

This intermittent indraught tunnel consists of a vacuum tank (ca. 80 m³), a vacuum pump, a fast acting globe valve, a supersonic test section and an air drier.

After a simple adaptation it was possible to use the wind tunnel, or rather some of its main parts, with additional measuring equipment for the experimental determination of pressure losses and flow rates of compressor valves.

A schematic diagram of the experimental set-up is shown in Fig. 1. The air flows through a straight intake pipe with a replaceable standard orifice plate, then through the valve to be investigated and finally through the globe valve leading into the vacuum tank. Due to the very low pressure in the vacuum tank at the beginning of the test (a vacuum of about 96 % is achievable) the flow properties of the valve can be studied over a wide range of pressure ratios.
Table 1: Survey of equations defining effective mass flow rate and loss coefficients

### Notation

- $A$ ........ geometrical flow area in the seat plate, $m^2$
- $C_{DC}$ ........ discharge coefficient (compressible flow)
- $C_{DI}$ ........ discharge coefficient (incompressible flow)
- $\dot{m}$ ........ effective mass flow rate, kg/s
- $\dot{m}_{th}$ ........ theoretical mass flow rate, kg/s
- $\dot{m}_{thmax}$ ........ maximum of theoretical mass flow rate (based on theory of isentropic one-dimensional flow), kg/s
- $P_0$ ........ atmospheric pressure, N/m$^2$
- $P_1$ ........ pressure upstream of the investigated valve, state 1, N/m$^2$
- $P_{1t}$ ....... total pressure, state 1, N/m$^2$
- $P_2$ ....... pressure downstream of the investigated valve, state 2, N/m$^2$
- $V_1$ ....... volume flow rate (related to $P_1$, $\varphi_{1t}$, state 1), $m^3$/s
- $\Delta P_F$ ....... pressure difference at the orifice plate, N/m$^2$
- $\varepsilon$ ....... expansibility factor
- $\phi$ ....... pressure loss coefficient
- $\kappa$ ....... ratio of specific heats ($=c_p/c_v$)
- $\rho_1$ ....... density (related to $P_1$, state 1), kg/m$^3$
- $\rho_{1t}$ ....... density (related to $P_{1t}$, state 1), kg/m$^3$
- $\phi_{1t}$ ....... discharge coefficient (incompressible flow)
- $\xi$ ....... discharge coefficient (compressible flow)
- $\gamma$ ....... ratio of specific heats ($=c_p/c_v$)
- $\rho_{1t}$ ....... density (related to $P_{1t}$, state 1), kg/m$^3$

### Incompressible Flow

\[
\dot{m} = P_1 \cdot V_1 = \dot{m}_{th} \cdot C_{DI} = C_{DI} \cdot A \cdot \sqrt{2 \cdot \rho_1 \cdot (P_{1t} - P_2)} 
\]

\[
P_1 - P_2 = \phi \cdot \frac{\rho_1}{2} \cdot \left( \frac{V_1}{A} \right)^2 
\]

### Compressible Flow

\[
\dot{m} = \dot{m}_{th} \cdot C_{DC} = C_{DC} \cdot A \cdot \sqrt{\frac{2 \cdot \kappa}{\kappa - 1} \cdot \frac{P_2}{P_{1t}} \cdot \left( \frac{P_2}{P_{1t}} \right)^{\frac{\kappa}{\kappa - 1}} - \left( \frac{P_2}{P_{1t}} \right)^{\frac{\kappa + 1}{\kappa - 1}}} 
\]

valid for: \( 1 \geq \left( \frac{P_2}{P_{1t}} \right) \geq \left( \frac{2}{\kappa + 1} \right)^{\frac{\kappa}{\kappa - 1}} \)

\[
\dot{m} = \dot{m}_{thmax} \cdot C_{DC} = C_{DC} \cdot A \cdot \left( \frac{2}{\kappa + 1} \right)^{\frac{\kappa + 1}{2(\kappa - 1)}} \cdot \sqrt{K \cdot P_{1t} \cdot \phi_{1t}} 
\]

valid for: \( \frac{2}{\kappa + 1} \cdot \left( \frac{\kappa}{\kappa - 1} \right)^{\frac{K}{\kappa - 1}} \geq \left( \frac{P_2}{P_{1t}} \right) \geq 0 \)

\[
\dot{m} = \dot{m}_{th} \cdot C_{DI} = \phi \cdot C_{DI} \cdot A \cdot \sqrt{2 \cdot \rho_1 \cdot (P_{1t} - P_2)} 
\]
During the first stage of a test run critical flow conditions are usually established in the smallest flow area of the investigated valve. That means that the flow velocity in this "critical cross-section" reaches the local speed of sound and the flow through the valve becomes "choked".

Before each test the vacuum tank is evacuated. After the globe valve has been opened the air flows through the test arrangement into the tank refilling it gradually. That means that normally all tests start at very small values of the pressure ratio \( p_2/p_1 \).

Compressor valves with a geometrical flow area in the seat plate of up to 30 cm² have so far been investigated in this set-up.

Throughout a test run with a large valve the pressure ratio alters continuously but so slowly that the flow characteristics remain quasi-steady.

The pressures upstream and downstream of the orifice plate as well as upstream and downstream of the tested valve are measured by means of electromagnetic pressure transducers and registered by a multichannel recorder.

BETZ - manometers and a precision mercury manometer are used for test measurements and for calibration.

If small valves (e.g. valves with a geometrical flow area in the seat plate of about 5 cm²) are tested in the described set-up the globe valve can be used successfully as an adjustable throttle. By this means the pressure downstream of the compressor valve can easily be controlled and thus, for a certain period, exact steady flow conditions are maintained.

In this way the flow conditions are changed step-wise and it is possible to record the signals of the pressure transducers by a programmable data-logger as well.

In order to analyse the experimental data simple computer programmes were developed.

For the valve investigations normally ambient air is used as test fluid. Application of additional and special equipment allows the measurement of the valve lift as well as of the force acting on the valve plate (or reed).

In many cases, especially for plate valves, the tests are carried out with several discrete valve lifts. The adjustment of the valve plate corresponding to a desired valve lift can be achieved by means of suitable spacers between valve plate and seat plate.

If measurement of the valve lift is required (in order to determine how the lift depends on the pressure difference) the installation of a special equipment (e.g. by use of a displacement transducer) is necessary as mentioned above.
Great care must be taken that this additional equipment does not influence the flow conditions and forces at the valve plate (or reed) to much.

The experiments showed that the pressure tappings upstream and downstream of the investigated valve must be positioned carefully. Especially when a single downstream tapping is placed very close to the valve, wall jets can influence the pressure readings significantly.

For the tests with the multi-ring plate valve (described in section 4.1) multiple pressure tappings with annular chambers were used upstream and downstream of the investigated valve. If very high accuracy is required, for example in comparative measurements in the course of valve improvement, the same "standard" test arrangement should be used for each run.

4. DISCUSSION OF TESTS AND RESULTS

Apart from commissioned investigations several experiments with compressor valves (a multi-ring plate valve and some reed valves) were carried out in order to test the equipment and to collect experience with it as well. Some of these measurements and results are summarized in the following.

4.1 Multi-ring plate valve

A sketch of the investigated valve, a delivery valve of conventional design, is shown in Fig. 2.

As the spring plate was rather stiff no useful valve lifts could be reached by means of the available pressure differences. Therefore pieces of precision wire as spacers were used to achieve a certain lift and to fix it throughout a run.

As seen from the shape of the recorded pressure readings choked flow is established during the initial stage of the test.

Fig. 3 shows an example (reduced in size) of such a typical record of \( p_1 - p_2 \), \( \Delta p_f \) and \( p_0 - p_1 \) belonging to a certain valve lift.

After opening the globe valve the value of \( p_1 - p_2 \) reaches its maximum and decreases throughout the test (according to the pressure rise in the vacuum tank). By way of contrast the value of \( \Delta p_f \), representative for the mass flow rate \( \dot{m} \), remains constant (indicating choked flow) during the initial stage of the test and then decreases at first very gradually. The curve of \( p_0 - p_1 \) shows a similar shape, as in this set-up the pressure upstream of the investigated valve is influenced by \( \dot{m} \) only.

The effective mass flow rate \( \dot{m} \), dependent on the valve lift \( s \), describes the discharge of a certain valve quantitatively and directly. In Fig. 4 measured values of \( \dot{m} \) are plotted as function of the pressure ratio \( p_2/p_1 \) for 6 different values of the valve lift.

In this diagram an additional curve (the top one) shows the relationship \( \dot{m}_{thc}(p_2/p_1) \), calculated according to the theory of one-dimensional isentropic flow and with the ratio of specific heats of \( \kappa = c_p/c_v = 1.4 \). The theoretical mass flow rate reaches its maximal value at the critical pressure ratio of 0.528.
VALVE TESTING

MULTI-RING PLATE VALVE, TEST-RUNS NO. 11, 12, 13, 14, 15, 16
VALVE-LIFT 0.6, 1.0, 1.5, 2.0, 2.5, 3.0 MM

Fig. 4 Variation of the theoretical and the effective mass flow rate, $\dot{m}_{thc}$ and $\dot{m}_{c}$, with the pressure ratio $p_2/p_1$ (from tests with a multi-ring plate valve)
The measured $\hat{m}$-curves indicate that choked flow occurs in the valve at pressure ratios being significantly smaller than 0.528. A very similar behaviour was found in discharge experiments, [6] and [7], with orifices, although the flow patterns of valves and orifices are quite different.

The shape of the curves in Fig. 4 suggests that one could attempt to calculate the $\hat{m}$-values from the $\hat{m}_{thc}$-curve by means of a two-parameter similarity transformation with $p_2/p_1 = 1$ and $\hat{m} = 0$ as fixed point.

One of the introduced parameters, here denoted by $\lambda$, takes into account that the critical pressure ratio $(p_2/p_1)_{crit}$, at which choked flow can first be established in the valve (coming from $p_2/p_1 = 1$), is smaller than the theoretical value for isentropic flow $(p_2/p_1)_{thcrit} = 0.528$. Obviously the simplest way to define $\lambda$ is:

$$\lambda = \frac{1 - (p_2/P_{it})_{thcrit}}{1 - (p_2/P_{it})_{crit}}$$  \hspace{1cm} (4.1)

The other parameter, called $\mu$, arises from the observation that the measured values $\hat{m}$ are smaller than the corresponding theoretical values $\hat{m}_{thc}$.

This leads to the transformation:

$$\hat{m} = \mu A \left(\frac{2K}{K-1} P_{it} \frac{2}{P_{it} - p^{K+1}}\right)$$  \hspace{1cm} (4.2)

with

$$P = 1 - \lambda (1 - \frac{p_2}{P_{it}})$$  \hspace{1cm} (4.3)

valid in the range:

$$1 \geq P \geq (\frac{p_2}{P_{it}})_{thcrit} = 0.528$$  \hspace{1cm} (4.4)

and

$$\hat{m} = \mu A \left(\frac{2}{K+1} \frac{P_{it}}{P_{it} - p^{K+1}}\right)^{-1}$$  \hspace{1cm} (4.5)

valid in the range:

$$\frac{p_2}{P_{it}}_{thcrit} = 0.528 \geq P \geq 1 - \lambda$$  \hspace{1cm} (4.6)

The parameters $\lambda$ and $\mu$ have to be determined from experimental data.

When valves of different construction, for example in the course of valve improvement, are to be compared with regard to their efficiency of discharge, non-dimensional coefficients must be used. Most suitable for this purpose is the discharge coefficient $C_{DC}$ according to equations (2.5) and (2.7).

Evaluation of the same experimental data as for Fig. 4 leads to the diagram Fig. 5, showing the variation of $C_{DC}$ with the pressure ratio $p_2/P_1$.

The corresponding plot of $C_{DI}$, the discharge coefficient based on theory of incompressible flow, is presented in Fig. 6.

The dashed lines in these diagrams extend the curves of experimental data up to the limiting pressure ratio $p_2/P_1 = 0$.

They were found according to theoretical considerations. The values of $C_{DC}$ do not alter for $p_2/P_1$ smaller than the critical pressure ratio, as in this range choked flow conditions are maintained and the flow rate remains constant. The corresponding values of $C_{DI}$ for $p_2/P_1 = 0$ and the initial slope of the curves can be calculated easily by means of the equations in Table 1 and a limiting process for $p_2/P_1$ approaching zero.

It is clear that these calculated values are of theoretical interest only and not relevant to practical application.

The coefficient $C_{DC}$ describes the discharge efficiency of the Valve for all pressure ratios correctly. A comparison of Fig. 5 and Fig. 6 shows immediately that $C_{DC}$ and $C_{DI}$ are only equivalent at pressure ratios very close to 1.
Variation of the discharge coefficient $C_{D1}$ with the pressure ratio $p_2/p_1$ for different lifts (0.6, 1.0, 1.5, 2.0, 2.5 and 3.0 mm) of a multi-ring plate valve

As soon as compressibility must be taken into account $C_{D1}$ gives a wrong picture of the real discharge efficiency.

The $C_{D1}$-curves (Fig. 5), especially those belonging to greater values of valve lift, indicate that their behaviour in the range of $p_2/p_1$ between about 0.8 and 1.0 makes further considerations necessary.

In this range, when $p_2/p_1$ is approaching 1, the flow velocities in the valve become smaller and the influence of compressibility decreases. But it is obvious that other effects are superimposed and become more predominant.

From test results and from theoretical considerations one can deduce that for smaller velocities viscous effects, especially the growing of the boundary layer (e.g. expressed by the displacement thickness), play an important role and influence the discharge behaviour essentially.

This fact becomes evident also by a treatise of the experimental data in order to evaluate the transformation parameters $\lambda$ and $\mu$, defined by the equations (4.1) and (4.2) to (4.6). As mentioned already, the aim of this transformation was to match the measured flow rates to one-dimensional isentropic nozzle flow.

The accurate location of $(p_2/p_1)_{\text{crit}}$ in the pressure recordings (see Fig. 3) and therefore the determination of $\lambda$ is difficult, since the $\Delta p_2$-curve approaches its constant value (indicating choked flow) very gradually.

Careful evaluation of the experimental data showed that $\lambda$ does not depend significantly on the valve lift $s$ and no clear relationship between $\lambda$ and $s$ was obvious. Thus, in the further calculations, a mean value of $\lambda$ is used. By that the values of $\mu$ can be computed. They are plotted in Fig. 7.

From this diagram one can see that in a wide range of pressure ratios (up to about 0.8) the $\mu$-curves can be approximated, with reasonable accuracy, by horizontal lines. This means that $\mu$ depends there only on the valve lift as it is desired for the transformation.

Proceeding from the "horizontal lines" in Fig. 7 the variation of $\mu$ with $s$ is found. A plot of $\mu(s)$ for this valve is sketched in (8).

Using $\lambda$ and $\mu(s)$ in the equations (4.2) to (4.6) the effective mass flow rate $\dot{m}$ can be approximated by this transformation quite well within that range of pressure ratios in which compressibility effects are predominant.

For greater pressure ratios, approximately for $0.8 \leq p_2/p_1 \leq 1.0$, the transformation fails.

The influence of viscosity effects on $\mu$ is discussed by L. BOSWORTH (8) elsewhere in these proceedings. In (8), based on experimental data, an attempt is made to estimate how $\mu$ depends on a typical Reynolds-number characterizing the boundary layer in the entrance channel of the seat plate.
In spite of the considerations mentioned above, extensive and special investigations are still necessary in order to get more knowledge about the influence of viscosity on the discharge efficiency of a valve.

4.2 Reed valve

Several investigations of reed valves, suction and delivery valves, were performed with the described test equipment as well. All these experiments were carried out with a variable valve lift during a test run.

As the valve lift was not an adjustable parameter, the results obtained did not give much information about the influence of lift on the discharge efficiency of the tested reed valve.

Similar to the multi-ring plate valve the tests with the reed valves also showed that choked flow occurs at small pressure ratios.

A series of comparative measurements with a reed valve was performed with the aim of determining, in how far different main parts of the valve influence the overall discharge coefficient. In order to separate the different effects such a series consisted for example of distinct tests with the seat plate only, with seat plate and reed, with seat plate, reed and cylinder head. The results give a good insight in the flow behaviour of the different components. In addition they enable one to estimate where fluid mechanical improvements can be made.

5. CONCLUDING REMARKS

The presented test equipment proved to be very useful for investigations of compressor valves with the aim of determining discharge coefficients and pressure losses as well as to examine the flow behaviour over a wide range of pressure ratios.

The obtained results show that in compressor valves choked flow can be established at pressure ratios being smaller than a critical value. This critical pressure ratio depends on the design and on the valve lift.

But in order to understand the discharge behaviour of a valve in more detail, for example to get more knowledge about the influence of viscosity on discharge efficiency, extensive investigations and research are necessary.

As the described experimental set-up permits low-cost testing it is suitable for commercial investigations and for special research work.

6. REFERENCES


