MODELLING VALVE DYNAMICS AND FLOW IN RECIPROCATING COMPRESSORS

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1. INTRODUCTION

Reciprocating compressors are widely employed in a number of industry and transportation branches, and it can be freely stated that some of the applications would hardly be possible without this type of machinery. The latter refers to such extreme cases as compressing ethylene to pressures upwards of 300 MPa for the purpose of producing LDPE (low-density polyethylene), very low suction temperatures (of the order of -150 ºC) in the field of liquefied gas transport and storage, or for compressing gases contaminated with particles. In commercial vehicles for road transportation, reciprocating compressors are customarily used for obtaining pressurized air for auxiliary purposes, such as braking, gear shifting, etc. Common to almost all reciprocating compressor applications is the fact that the compressor is a rather small component in comparison with the process and/or system that it supplies with gas, but its reliability determines the availability and safety of the entire plant. Therefore, the plant designers and owners require trouble-free operation from their compressors over long periods of time. Indeed, expected service time for a small hermetic compressor in a common household refrigerator is more than 20 years.

Conceptually, a reciprocating compressor stage consists of a cylinder, the volume of which varies periodically due to the motion of a piston that closes one end of the cylinder. The other end of the latter is closed by two valve sets, one each for admitting the gas to be compressed into the machine (suction valve), and for allowing the high pressure gas to be delivered to the process and/or machines utilizing it (discharge valve). One speaks here in terms of valve sets because there are machines (usually large process compressors) which may be equipped with more than one valve pro suction and discharge side, respectively.

From the standpoint of thermodynamic performance, the cylinder must be completely sealed at both ends during the compression process, and the suction and discharge processes are to be realized exclusively through the respective valves, which translates into the zero-leakage requirement for the machine. While the piston can be reliably sealed by means of one or more rings (lubricated or dry-running) that press against the cylinder wall, securing zero-leakage function of the valves is by no means a simple task.

The key feature of compressor valves that simultaneously affects both their sealing performance and reliability is that they, unlike their counterparts in a conceptually similar IC engine, are not actuated. They are held closed by elastic forces internal or external to the sealing element; and they open and close automatically, in accordance with the balance of gas pressure forces and the previously mentioned elastic ones. Under the gas pressure forces one understands both the force due to static pressure difference across a closed valve and the

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aerodynamic drag brought about by the gas flow impinging onto the sealing element in a partially or fully open valve. Clearly, a high degree of coupling between the gas flow and the sealing element motion, nowadays referred to as fluid-structure interaction (FSI), is to be expected in such a device. The latter and the absence of a reliable guide device that would provide for the plan-parallel motion of the sealing element give rise to non-parallel impacts between the latter and other parts of the valve assembly, leading to the so-called Dynamic Stress Concentration Effect [6], which in turn causes premature sealing element fracture and the machine shutdown.

According to a survey carried out in the process compressor field in 1996, compressor valves represent the primary cause of unscheduled reciprocating compressor shutdowns, with a relative frequency of 36% [23]. Since the second-ranking cause of machine failure is piston rod packing (17.8%), which are present only in crosshead and compound machines, one may surmise that valves are responsible for even a larger percentage of failures in small compressors, such as e.g. used to compress air in commercial motor vehicles.

Given the passive nature of the automatic compressor valves, the mass-spring oscillatory system of the sealing element cannot be fully matched to the compressor over a wide range of operating conditions. While large process compressors normally run either at a constant speed or within not very wide speed ranges, giving thus the valve designer a chance to optimise the valve parameters or even excludes dangerous compressor speeds, this is definitely not the case with small air compressors of commercial vehicles. The latter is expected to perform reliably within the entire engine speed range, i.e. from the idle to the maximum rpm. The matching and optimisation of the valves have been the subject of active research in the past several decades, resulting in a body of knowledge whose salient features are to be surveyed in the present contribution.

2. VALVE MODELLING

2.1. GENERAL CONSIDERATIONS

Conceptually, the influence of the valve upon the compressor performance and reliability can be analysed in terms of the following three phenomena, i.e. models:

1. Mechanical, accounting for the opening and closing processes motion of the spring-mass assembly of the sealing element, and whatever impact processes within the valve. This area is commonly referred to as valve dynamics.
2. Flow, describing the relationships between the mass flow rate through the valve and the gas states in the cylinder and the valve attachments.
3. Coupling i.e. interaction between the valve action and the fluid dynamics at the upstream and downstream sides of the valve.

While the first two phenomena can be studied both analytically and experimentally in isolation from the compressor i.e. by specifying constant or variable fluid states upstream and downstream of the valve, the third one is a system phenomenon and can thus only be analysed together with the cylinder and the attached piping and fittings. Experimentally, the latter calls for measurements at a suitable test rig or in the compressor installation; and analytically, a comprehensive system model is needed that includes all relevant components and processes. Regarding the modelling depth, it was an established practice in the past to lump the respective piping at the suction and discharge sides of the machine into volumes, neglecting thus the wave motion that inherently takes place there. This approach simplifies the plant model considerably, saving also the computation time. MacLaren [30] argued that the accuracy of performance prediction is affected by this simplification, and it was shown in [13][32][33][21] that large differences do exist between the performance figures obtained with the two model assumptions. However, further discussion of this subject is beyond the scope of the present paper.

2.2. MODELLING OF THE VALVE DYNAMICS

In principle, an automatic compressor valve consists of a movable sealing element, a seat against which the latter rests when the valve is closed, means for generating a force that presses the sealing element against the seat, and means for limiting the extent of the sealing element motion when the valve is fully open. The sealing element must always be physically present as a distinctive part; other items from the above list may be realized by employing parts of the compressor or of the sealing element itself. For example, in a reed valve, which represents the simplest valve design, one finds only the sealing element as an individual part: the elastic force that keeps the valve at the seat is generated by the elasticity of the reed, the seat is machined into the cylinder head, and the stroke limiting is achieved by the bending resistance of the reed (Fig.1). This type of valve is commonly encountered in small refrigerating compressors for domestic use.

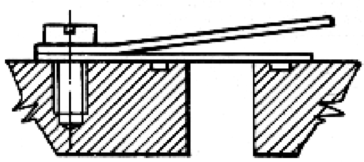

Figure 1: *Simple reed valve*

Large process compressors are equipped with valves of a more elaborate design, Referring to Fig. 2 below, in all three valve designs shown one finds the above mentioned four basic components realized as individual parts: the seat at the bottom, the sealing element with coil springs in the middle, and the stroke limiter at the top. They only differ in the type of the sealing element used; from left to right one sees a plate, concentric rings, and poppet. The plate valve shown has a sealing element made of reinforced plastic; a valve with a metal plate would normally also incorporate another metal plate whose function is to decelerate the sealing element on its way toward the stroke limiter, reducing thus the impact velocity on reaching the latter and increasing the chances for plan-parallel motion. The damper plate is normally not needed in metal plate valves used in small air compressors; the discussion shall henceforth be limited to such valves.

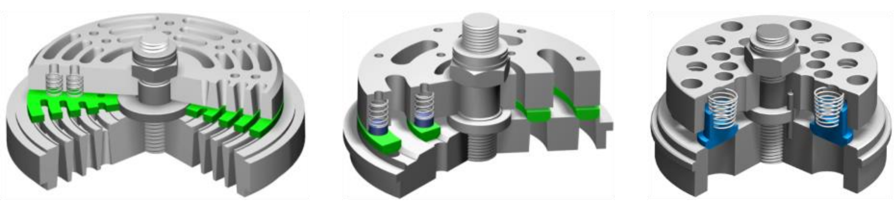

Figure 2: *Common valve designs (from [45])*

For the purpose of modelling, all valve designs without a damper plate can be abstracted to a configuration with a single sealing element depicted in Fig. 3 below.
In order to fully describe the operation of the valve, one must account for several states, events and processes that take place within the valve assembly. These are:

1. Valve in a closed state.
2. The opening event. When the force due to pressures acting upon the two sides of the plate overcomes the forces holding the plate at the seat, the valve begins to open and the gas starts flowing through the gap. Generally, there are three forces that oppose separation of the plate from the seat, these being the spring force, the stiction force due to a possible oil film at the contact surfaces, and the pressure force due to the unequal areas of the plate exposed to the respective pressures at the two sides of the plate.
3. Plate in motion between the seat and the stroke limiter, subjected to the spring force, the force due to the drag of the gas flowing past the plate, and the fluid friction opposing the motion (process).
4. Impact upon the stroke limiter. Note that there may be repeated impacts before the plate settles down.
5. Valve in a fully open state.

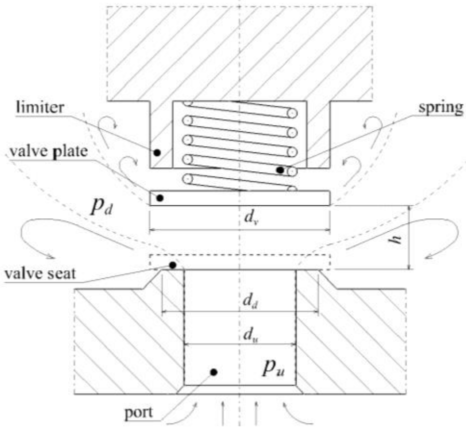

Figure 3: *Generic model of a compressor valve*

6. The detachment event. When the spring force prevails over the forces holding the valve open, the plate disengages from the stroke limiter. Similarly to the opening event, oil and/or gas stiction should be modelled (if present).
7. Plate in motion between the stroke limiter and the seat, same as Point 3 above.
8. Impact upon the stroke limiter. Note that there may be repeated impacts before the plate settles down.
9. Valve in a closed state.

2.3. VALVE FLOW MODELLING

Practically in all compressor performance prediction models, energy losses incurred at valves are modelled by measuring the performance of the valve in question on a test rig, and defining the so-called *discharge coefficient* as the quotient of the mass flow rate measured and a reference (ideal) value under the same flow conditions:

$$\alpha = \frac{\dot{m}_{\text{meas}}}{\dot{m}_{\text{ideal}}} = \frac{\dot{m}_{\text{meas}}}{\dot{m}_{\text{isen}}} \tag{1}$$

The reference component used is the ideal nozzle, having the same cross-sectional area at the throat as the valve being modelled. The true mass flow rate through the real valve is then calculated by assuming isentropic expansion between the upstream and downstream conditions, and multiplying the ideal mass flow rate by the discharge coefficient. In reality, however, the (presumably) lower mass flow rate through the real device is caused by irreversibility, generating thus entropy in the flow process.

The discharge coefficient concept has its origin in the practice of flow rate measurement by means of standardized flow restrictions, such as e.g. orifices, nozzles, Venturi meters etc. [36]. In order to apply this concept to practical mass flow rate calculations, one must merely specify a suitable geometric cross-sectional area for the device in question, denoted by e.g. $A_{v.geo}$. Knowing the thermodynamic states at the device inlet and outlet, the actual mass flow rate in subsonic flow is calculated by invoking the well-known Saint-Venant-Wantzel equation of 1839:

$$\dot{m} = \alpha \cdot \dot{m} = \alpha \cdot A_{v.geo} \cdot \frac{p_{1t}}{\sqrt{R \cdot T_t}} \cdot \sqrt{\frac{2\kappa}{\kappa - 1}\left(\Pi^{-\frac{2}{\kappa}} - \Pi^{-\frac{\kappa+1}{\kappa}}\right)} \qquad (2)$$

wherein $\Pi = p_{1t} / p_b$ stands for the inlet total (stagnation) to back (static) pressure ratio. The isentropic part of the above formula refers to a reversible outflow from a pipe (hence the stagnation pressure and temperature terms) or a vessel; in the latter case the total pressure and temperature are replaced by their respective static quantities. The term to the right of the cross-sectional area represents the mass flux of the reversible outflow process under the thermodynamic conditions specified. It may be thought of as to result from a flow process taking place in an ideal nozzle, leading thus to the term *equivalent nozzle* for the reference element used.

The product of the discharge coefficient and the geometric cross-sectional area is referred as the *effective cross-sectional area* or *effective flow area* of the valve:

$$A_{v.eff} = \alpha \cdot A_{v.geo} \qquad (3)$$

It should be borne in mind that the above formula set does not constitute a valve flow model in the sense of gas dynamics; it is merely a calculation device for arriving at the mass flow rate under given thermo dynamical conditions. However, discussing this subject is beyond the scope of the present paper; the interested reader is referred to e.g. [44] for further details.

Generally, the discharge coefficient for a given valve depends upon the geometric flow area and the pressure ratio across the valve. However, since the term under the square root of Eq. (2) has a maximum at:

$$\Pi_{\text{crit}} = \left[0.5(\kappa + 1)\right]^{\kappa/\kappa+1}, \text{ with } \Pi_{\text{crit}} = 1.893 \text{ for } \kappa = 1.4 \qquad (4)$$

which is customarily referred to as the critical pressure ratio, and since it is an established fact (e.g. [6]) that compressor valves choke at much higher pressure ratios (up to 10), the above equation cannot be used as a mass flow rate model beyond the critical pressure ratio. The information as to the mass flow rate at overcritical pressure ratios must thus be supplied by the discharge coefficient, and this involves measurements and/or CFD studies.

There are flow model formulations, such as e.g. the Fanno flow theory [3], that take into account the effects of irreversibility in throttling flow upon the pressure ratio value at the onset of choking, but they have so far not been used in the compressor simulation practice.

3. VALVE DYNAMICS SUBMODELS

The dynamics of self-acting compressor valves was first considered in a systematic manner by Costagliola [14]. Although his model assumed stable behaviour of the sealing element, i.e. it did not allow for flutter, it provided a foundation upon which the majority of models developed in its aftermath were built. The developments up to 1972 were reviewed by MacLaren [30]and, somewhat later and in much more detail, by Touber [39]. Bukac [12] attempted to present the entire field of valve dynamics and flow simulation in a compact manner, but his paper should be read as an overview, for it lacks arguments for choosing particular formulae and treats the flow calculation inconsistently in that a polytrophic change is used simultaneously with the isentropic choking criterion. Habing [23] presents a modern view of the field and includes measurements to verify the theories used.

The only book devoted entirely to compressor valves was published by Böswirth [6], who has also been one of the most prolific authors in the field.

3.1. FORCE BALANCE AT THE OPENING EVENT

Using A_{u} and A_{d} to denote the respective upstream and downstream plate areas in contact with the seat, $F_{\mathrm{s,c}}$ for the spring force in a closed valve and F_{adh} for the adhesion force, the valve is in a closed state when the following inequality is satisfied:

$$p_{\mathrm{u}} \cdot A_{\mathrm{u}} - p_{\mathrm{d}} \cdot A_{\mathrm{d}} \leq F_{\mathrm{s,c}} + F_{\mathrm{adh}} \qquad (5)$$

whereby p_{u} corresponds to the cylinder pressure in the case of a discharge valve, and to the plenum (valve chamber) pressure in the case of a suction valve.

The valve starts opening when the inequality condition in Eq. (5) reverses, i.e.

$$p_{\mathrm{u}} \cdot A_{\mathrm{u}} - p_{\mathrm{d}} \cdot A_{\mathrm{d}} > F_{\mathrm{s,c}} + F_{\mathrm{adh}} \qquad (6)$$

The opening process is strongly dependent upon the adhesion force in the contact area between the valve plate and the seat. If no liquid is present in the contact area the adhesion force may be due to molecular forces and/or under pressure; Touber [39] uses the term "stiction" to refer to this phenomenon. He models the adhesion force by expressing it as an integral of the pressure distribution in the contact surface, i.e.

$$F_{\mathrm{adh}} = \int_{A_{\mathrm{c}}} \left(p_{\mathrm{u}} - p\right) \mathrm{d}\, A_{\mathrm{c}} \qquad (7)$$

Wherein A_{c} represents the contact surface area and p the pressure distribution function.

The situation is much more complicated if liquid is present in the contact area, which is always the case with lubricated machines and/or when liquid droplets are carried by the gas being compressed. Giacomelli and Giorgetti [20] considered this effect to be much stronger at the stroke limiter, which was also the opinion of Bauer [2].The former authors [20] also measured the stiction force on a custom test rig, and found out that the shape of the stroke limiter surface has a strong influence upon the stiction force.

Khalifa and Xin Liu [26] concentrated on the suction valve stiction at the seat, and concluded that the primary reason for the stiction is the force arising from the oil film in the contact area being dilated in the opening process. Departing from the Reynolds equation of hydrodynamic lubrication

$$\frac{\partial p}{\partial r} = \mu \cdot \frac{\partial^2 u_r(z)}{\partial z^2} \tag{8}$$

where:
p - pressure variation in the film between the valve plate and seat,
r - radial coordinate of the oil film (meniscus), $R_{mi} \le r \le R_{me}$
R_{mi} - internal radius of the meniscus,
R_{me} - external radius of the meniscus,
h - distance of the plate from the seat,
$0 \le z \le h$ - axial coordinate,
u_r(z) - velocity profile, and
μ - dynamic viscosity of oil,

they arrived at the following general equation:

$$F_{adh} = C \cdot \frac{\dot{h}}{h^3} \tag{9}$$

wherein the term C is referred to as stiction coefficient. Generally, the latter depends on the geometric features of the valve, and the viscosity of the liquid, e.g. oil, in the valve contact area. Several formulae for calculating the stiction coefficient are available in the literature; for illustration, we quote here two such formulae for the same physical configuration of the valve seat:

Table 1: *Two formulae for the calculation of the stiction coefficient*

Source	Stiction force of separation for raised flat valve seat and oil-starved gap
[1]	$C = \mu \cdot \left(\frac{d_d - d_u}{2}\right)^3 \cdot \frac{d_d + d_u}{2}$
[12]	$C = \frac{3 \cdot \pi \cdot \mu \cdot (r_B^4 - r_A^4)}{32} \cdot \left(\frac{r_B^2 - r_A^2}{(r_B^2 + r_A^2) \cdot (\ln r_B - \ln r_A)} - 1\right)$ $r_A = \frac{d_u}{4} \cdot \left(1 + \frac{h_0}{h}\right) + \frac{d_d}{4} \cdot \left(1 - \frac{h_0}{h}\right)$ $r_B = \frac{d_u}{4} \cdot \left(1 - \frac{h_0}{h}\right) + \frac{d_d}{4} \cdot \left(1 + \frac{h_0}{h}\right)$

where h_0 stands for initial thickness of the oil film, and d_u and d_d are the inner and outer seat diameters, respectively (see Fig. 3).

In a most recent contribution to this area [35], the authors augment the model of Eq. 9 by introducing terms that take into account the curvature of the meniscus (capillary force), and the interfacial tension force. Upon comparing their formula with the one of Ref. [26] the authors state that their approach could not be validated due to the lack of experimental data.

3.2. FORCE BALANCE AT A MOVING VALVE PLATE

As soon as the valve begins to open the gas starts flowing through the gap between the plate and the seat, causing the upstream pressure at the former to diminish, which may in its turn lead to a temporary closure of the valve. This may give rise to an alternation of the opening and closing events i.e. to the "bouncing" of the plate at the seat.

Denoting by h the distance of the plate from the seat, by F_s the spring force, by F_d the drag exerted upon the plate by the flowing gas, and by F_r the fluid friction force opposing the motion, one can write the instantaneous force balance at the plate as:

$$m_p \cdot \ddot{h} + F_r + F_s - F_d = 0 \tag{10}$$

which represents a general, single degree of freedom differential equation of the motion of mass m_p in the direction perpendicular to the valve seat.

Away from the seat and stroke limiter, damping of valve plate motion may be caused by friction forces acting upon the moving plate; in the vicinity of the seat and at the limiter, there are also the respective partially elastic impacts. The friction force is customarily modelled by assuming proportionality to the velocity $\dot{h}$ of the valve plate [11][30][39]:

$$F_r = C_f \cdot \dot{h} \tag{11}$$

where C_f denotes the valve plate damping coefficient. Touber [39] found that the same value of the damping coefficient applies to both the suction and discharge valves if they are of identical design and size, implying thus the former's independence on the gas density. Touber also observed a weak dependence of C_f upon the oil content in the gas, and generally low values of the coefficient.

The same author [37] obtained a very reasonable agreement of experimental data and theoretical results by modelling the friction force as proportional to the spring load and acting in a sense opposite to the velocity $\dot{h}$ of the valve plate. This empirical result could be explained to some extent by assuming mechanical friction at the spring surfaces which make a slight sliding action with respect to each other and to the valve plate when the latter is moving.

In most cases, valve spring force can be considered to be linearly dependent upon the deformation for the small distances over which a valve plate usually travels. The spring force is then given by:

$$F_s = k \cdot (h + h_0) \tag{12}$$

where k denotes the spring constant, h is the distance of the plate from the seat, and h_0 is the spring preload length. For a detailed analysis of the dynamic stresses in valve springs the reader is referred to [24].

Although Eq. (10) tacitly assumes that m_p represent the valve plate mass, one should also take into account the motion of the springs. Touber [37] was concludes that the inertial effect of the springs can be accounted for by adding an equivalent mass equal to one-third of the mass of the spring to the mass of the valve plate.

The force that gives rise to the plate motion, denoted by F_d in Eq. (19), and referred to as the drag or gas force, is the result of pressure distribution in the flow field around the valve plate. It is calculated as the area integral of the gas pressure load present at both sides of the valve plate, and customarily expressed as [19][39]0:

$$F_d = C_d \cdot A_v \cdot (p_u - p_d) \tag{13}$$

where C_d denotes the drag coefficient, and A_v the valve plate area. In analogy with the effective flow area of Eq. (2), the product $C_d \cdot A_v$ is referred as the *effective force area*.

The drag coefficient is assumed to be constant for a given valve configuration, but as MacLaren [30] points out, this is rather not the case. Its value is determined empirically. However, Touber [37] demonstrated a theoretical way to determine this coefficient by applying the momentum equation to a control volume enclosing the valve plate, obtaining:

$$C_d = \left[1 + (\alpha \cdot \varepsilon \cdot \frac{\pi \cdot d_v \cdot h}{A_u})^2 \right] \cdot \frac{A_u}{A_v} - \frac{\left[\alpha \cdot \varepsilon \cdot (A_v - A_u)\right]^2}{A_v - A_u} \tag{14}$$

where α and ε denote the flow and expansion coefficients, respectively (see [39] for details).

Schwerzler and Hamilton [38] developed an analytical method to obtain the effective force areas by assuming incompressible flow, and arrived at equations that depended only on the geometry of the valves; the agreement with the measurements was good. Yuejin and Yongzhang 0 combined the theoretical and experimental studies, and concluded that their mathematical model of the drag coefficient was too complex and strongly dependent on the experimental data; they replaced it subsequently by a curve fit of the experimental data.

Valve plate and the associated springs constitute a potentially oscillating system; the excitation necessary to give rise to oscillations is provided by the interaction of this system with the flow. This phenomenon is referred to as flutter, and since it can lead to premature valve plate failures it was investigated by several researches in the field. In an early work by Upfold [43], experimental records of valve motion were used to arrive at design criterions a valve/spring system should fulfil in order to avoid flutter. In the notation of the present paper, an approximate relationship between design parameters and operation conditions of valve when oscillations or flutter of the inlet valve would not occur is defined as:

$$\omega \cdot p_u > \frac{k^{1.5} \cdot h_{max}}{3 \cdot C_d \cdot A_v \cdot m_p^{0.5}} \tag{15}$$

where ω denotes rotational speed of the compressor, and h_{max} is the maximal stroke of valve plate. According to the author [43], good design of inlet valves for reciprocating compressors would be to ensure that the value $\omega \cdot p_u$ is preferably never less than twice the calculated value of the parameter on the right hand side of Eq. 15.

Böswirth [11] formulated a theoretical model of valve flutter, and constructed an enlarged valve model that was installed into a suitable experimental device in order to verify the flutter model. Various effects, such as e.g. those of gas springs and inertia were studied, and a similarity theory was developed as an aid for understanding unsteady valve behaviour. Although only simple reed valves were dealt with in the study, the author considers that the know-how gained could also be brought to bear on other valve designs as well.

Often value flutter is seen in conjunction with pressure pulsations in the piping. Although these are two quite different phenomena with possibility for mutual interference, there is a strong case for not neglecting the valve chambers and associated piping in the simulation of compressor performance, as already remarked in the General Considerations section of the present paper.

3.3. VALVE PLATE IMPACTS

Generally, impacts between the valve plate and the seat or stroke limiter give rise to stress concentration, leading to impact fatigue [34] which in its turn affects the service life of the plate. No generic modelling of the impacts seems to be possible, for the phenomena involved are rather complex. For example, it is questionable whether the classical impact between two solid bodies takes place in this case because, as the valve plate approaches either of the two limiting elements, it displaces the gas and oil present in the gap, adding thus a further mechanism to the impact process.

The valve plate is limited in its travel by the valve seat and in most valve designs by a limiter. It is assumed in the following that this limiter is fixed at an arbitrary distance from the seat and will not change its position when hit by the moving valve plate. When a moving body impacts at a fixed wall it will rebound with a velocity that is generally lower than the velocity before the impact. Only when its kinetic energy is absorbed or dissipated completely, it will remain in contact with the wall.

For the case that the valve plate rebounds, Habing [23] defines the so-called restitution coefficient as the ratio of the plate velocities immediately after (t^+) and before (t^-) the impact, i.e.

$$\left.\frac{dh}{dt}\right|_{(t^+)} = -e_{res} \cdot \left.\frac{dh}{dt}\right|_{(t^-)} \tag{16}$$

An impact is referred to as "fully elastic" when e_{res} equals 1, "inelastic" when $e_{res}=0$ and "semi-elastic" when $0 < e_{res} < 1$. Based on an analysis of his experimental results, Habing obtains $e_{res,s} = 0.3 \pm 0.1$ and $e_{res,l} = 0.2 \pm 0.1$ for the impacts of the valve plate at the seat and stroke limiter, respectively.

Performing a numerical investigation of the discharge valve dynamics with impact energy recuperation of 30%, which corresponds to a restitution coefficient of 0.55, Bukac [10] found almost negligible difference between hard stop and soft stop without oil on the valve lift and cylinder pressure. While the rebound from the stop has negligible impact on capacity, coefficient of performance (probably isentropic efficiency) and cylinder pressure, the rebound from the seat decreases compressor's capacity by 2.5% due to the flow-back of the gas.

3.4. VALVE IN FULLY OPEN STATE

A valve is fully open when the sum of forces holding the valve plate at the stroke limiter prevails over the spring force, which is expressed by the following inequality:

$$F_{s,o} \le F_d + F_{adh} \tag{17}$$

The adhesion force may be due to oil stiction or the vacuum between the latter valve parts; the term $F_{s,o}$ represents the spring force when the valve plate is at the stroke limiter.

3.5. FORCE BALANCE AT THE DETACHMENT EVENT

Note that this event is not the counterpart to the valve opening; it can be understood as the onset of the valve plate motion toward the seat. The condition for this to take place is:

$$F_{s,o} \geq F_d + F_{adh} \tag{18}$$

After this event is completed, motion of the valve plate proceeds in accordance with the relationships derived in the previous section on valve plate motion.

4. VALVE FLOW CALCULATION

With reference to the formula set for the calculation of mass flow rate through discrete fluid ports presented in the Valve Flow Modelling section above (Eq. (1) to (4)), one should bear in mind that they are derived for steady flow. Since the flow through cylinder valves is unsteady at least with respect to time, it is customary to consider the flow as being steady at a given time instant, and changing without delay to a new value at a subsequent one. This constitutes the so-called *quasi-steady* approximation, which is found in the vast majority of performance prediction programs for IC engines and reciprocating compressors. In effect, this approximation is equivalent to neglecting valve dimensions in direction of flow.

The key term in Eq. (2) above is the discharge coefficient, for if its value is known, the mass flow rate through the valve is calculated in a straightforward manner. It is reasonable to expect that the discharge coefficient depends upon the valve geometry and the flow conditions; in the latter case, pressure ratio across the valve would be the main factor due to the choking phenomenon. Although the particular relationships between the discharge coefficient and the above mentioned quantities are obtained experimentally, Böswirth [8][6]convincingly demonstrated that fairly accurate approximations can be obtained through the application of the jet and boundary layer theories. His derivations also offer valuable physical insight into the flow phenomena in a compressor valve.

Böswith's reports also contain experimental data for plate ring valve, which indicate that the strongest influence of the pressure ratio upon the discharge coefficient occurs at $1 < \prod < 1.3$ and that the flow is choked for $\prod \geq 3$. The author also introduces a different discharge coefficient which after the initial variation with pressure ratio, i.e. for $\prod > 1.3$ remains essentially constant, but the pressure ratio term in Eq. (2) must be modified in order to use his discharge coefficient values. This is in line with the analysis of Blair [5], who insists that in dealing with the discharge coefficient the formula employed for the mass flow calculation must be inverted for the purpose of determining the discharge coefficient from the experimental data.

Measurements of valve flow are usually performed by fixing the valve plate at a given lift and varying the flow conditions, i.e. the pressure ratio. Data obtained in this manner are thus valid for a fully open valve; flow regime in a partially open valve is not adequately represented by these data. However, performing measurements with a normally configured valve, i.e. with a spring-loaded plate, is extremely difficult because of latent flow instabilities and is thus rarely performed.

Calculation of valve flow in the field of IC engines is still based on Eq. (2), i.e. on the cross-sectional area and the discharge coefficient being separated from each other. The trend in the compressor industry has been to use the effective flow area, probably because most of the studies deal with small refrigerating compressors which are rather similar to each other. In addition, the effective flow area is treated with the effective force area in the majority of studies. This makes sense, for there is a significant interaction between the two phenomena.

One of the first investigations of this kind is due to Schwerzler and Hamilton [38] who obtained good agreement between their theoretical predictions and the experimental data, although the former were derived by assuming incompressible flow in the valve.

Several other research teams used the concept of flow and force effective areas to determine the mass flow rate through the discharge and suction valves, as well as forces acting on the valve plate. Ferreira and Driessen [17] performed extensive measurements on different valves and presented the results obtained in non-dimensional form; in most cases they found much smaller variations of effective flow areas with the respective valve plate lifts than was the case with the corresponding effective force areas. Dechamps et al. [16] continued the above study and combined the measurements with numerical predictions under the conditions of laminar flow. Price and Botros [37] compared measured data with their numerical predictions of the effective flow area at low Reynolds numbers, and having obtained a good agreement proceeded to predict the former at high Reynolds numbers. They also found that the effective force area significantly varies with the lift. Kerpicci and Oguz [25] performed a CFD study of transient, i.e. unsteady, flow in leaf valves and found out that the discharge coefficient may be a function of the valve lift and pressure ratio, which Blair [5] also established in the case of IC engine valves. Recently, Murakami et al. [31] expressed the effective flow area as a surface plot in terms of the relative valve lift and the distance between the valve and the piston, and concluded that the simulation accuracy was better with the latter approach than with the model expressed in terms of valve lift only.

As already mentioned, an alternative to the calculation of mass flow rate by means of the discharge coefficient would be a model that explicitly takes into account the total pressure loss across the valve. For a general flow device with loss, the latter is expressed in terms of the *loss coefficient, denoted by* ξ and defined as:

$$\xi = \frac{p_{1,t} - p_{2,t}}{\left(\frac{1}{2} \cdot \rho \cdot u^2\right)_{ref}} \tag{19}$$

where $p_{1,t}$ and $p_{2,t}$ stand for the total (stagnation) pressures at the inlet and outlet of the loss device, respectively, and the term in the denominator is the denormalizing dynamic pressure, taken at the inlet cross section. If the loss coefficient is expressed for an infinitesimal length and the resulting energy equation for compressible flow integrated between the inlet and the outlet, one obtains the so-called Fanno flow model [3]. Full discussion of such models is beyond the scope of this paper, but it is important to note that they, unlike the discharge coefficient concept, define the loss in a thermodynamically correct manner in that they relate the loss naturally with the entropy production as the ultimate loss metric [22].

Models based on the loss coefficient are used in the hydraulics, with an additional advantage of being augmented by vast collections of experimentally determined loss coefficient data for almost any device of importance for the practice. As to the reasons why this model class has not found application in the compressor simulation practice and, for that matter, in the area of IC engine simulation, the authors are of the opinion that it is the simplicity of Eq. (2), and the fact that the discharge coefficient concept has been the only one in use in the field of IC engines from the very beginning of their history. The latter has been especially the case in the simulation models that incorporate one-dimensional gas dynamics methods to describe the wave action in the engine manifolds [4].

5. COUPLED SIMULATION OF VALVE DYNAMICS AND FLOW (FSI)

The quasi-steady valve flow calculation approach is justified for the cylinder and/or valve configurations with short channels; if the latter is not the case, the delays in the valve dynamics due to the inertia of the gas resident in the valve channels should be taken into account. Trella and Soedel [41] were among the first to treat this problem in a discharge valve; the model [41] was subsequently used to investigate the behaviour of the valve under different operating conditions [42]. Although the simulation results were not compared with measurements, the authors recommend the inclusion of the gas inertia effects into the valve calculation models, especially for fast-running compressors. Böswirth [9] performed a comprehensive theoretical analysis of the coupling between the gas and valve dynamics, and performed a series of measurements on a custom-built test rig [10]. With regard to the phenomenon of coupling, two main mechanisms were identified: gas inertia and unsteady work exchange between the flow and the valve plate; and the flutter was found to be affected by the so-called "gas spring effect". According to the author, replacing quasi-steady models by simplified unsteady models would constitute a small effort in comparison with gains in terms of simulation accuracy and understanding of the flutter phenomena. Habing [23] considered the unsteadiness in valve flow by augmenting the steady flow pressure loss formula by a transient flow term, and the already mentioned study of Kerpicci and Oguz [25] documents large differences between the quasi-steady and unsteady modelling approaches. Recently, after performing a CFD study of the fluid/structure interaction in a discharge valve, Link and Dechamps [29] state that "standard definitions of effective flow and force areas are not capable of describing mass flow rate and flow induced force in the opening and closing stages of the valve displacement". They rate their study is an initial step towards the understanding of flow inertial effects in compressor valves, and indicate the need for general correlations of effective flow and force areas for transient flow conditions.

6. SIMULATION RESULTS

In order to illustrate some of the concepts and models discussed above, simulation of a small air compressor with a 74 mm bore and 35 mm stroke was performed with a compressor simulation program that can also consider unsteady flow in the valve chambers and attached piping. The compressor is currently being experimentally investigated on a custom test rig for small air compressors in the Engine Laboratory of the Faculty of Engineering of Kragujevac, Serbia. The rig currently allows for recording several quantities of importance for evaluating thermo dynamical performance of the machine under test, such as cylinder pressure, exit mass flow rate, gas temperatures etc. Data recorded at an operating point characterized by the delivery pressure of 5 bar and the machine speed of 1000 rpm were selected for comparison with the simulation.

The machine obtains air through a small filter and approx. 50 mm of pipe from the laboratory, and delivers the compressed gas through 1.7 m of piping to an air-cooled heat exchanger. The model included all pipes up to the heat exchanger; due to the lack of the heat transfer data, the latter was modelled as a volume discharging the gas through a resistance to the environment. On account of the intricate construction of the cooler it was felt that a major part of the pulsations on the discharge side would be dissipated in the latter, obviating thus the need for including the rest of the piping in the simulation model. All cylinder walls were assumed to be adiabatic.

The valve simulation model consisted of the following sub models:

- Valve opening force balance without oil stiction, but with finite seat land area, i.e. $d_d \neq d_u$ (see Fig. 3)
- Calculation of the friction, spring, and drag forces per Eq. (11) – (13), respectively; the drag and friction force coefficients are assumed to be constant
- Valve plate impacts at the seat and stroke limiter per Eq. (16), and the restitution coefficient values within the bounds established by Habing [23]
- Valve mass flow rate calculation by means of the discharge coefficient model, as defined by Eq. (1) – (4)

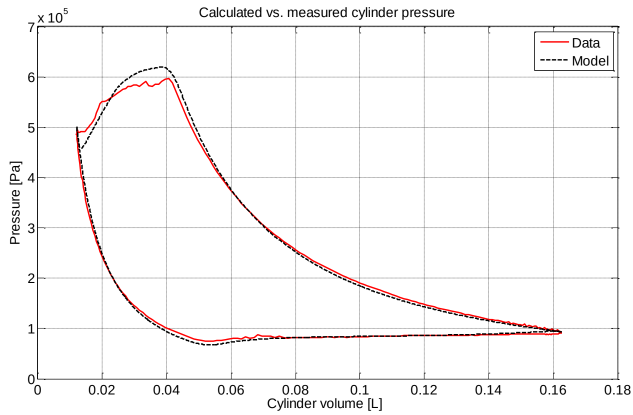

Figure 4: *Calculated vs. measured indicator diagram for a small air compressor*

Referring to Fig. 4, the calculated indicator diagram in terms of pressure is compared to the one obtained by averaging 40 measured cycles. In spite of the rather simple valve and cylinder models used, the agreement is satisfactory, with the exception of the discharge process, where one can claim a fair average agreement. Apparently, the discharge valve opens slower and remains open longer in the simulation as in the reality, indicating thus the need for model improvements. The valve lift traces of Fig. 5 suggest normal valve function at this operating point.

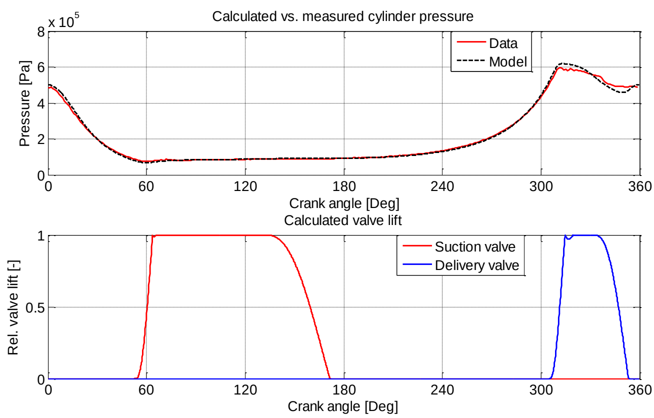

Figure 5: *Pressure traces and calculated valve action for a small air compressor*

7. CONCLUSIONS

The survey of the open literature presented in the paper shows that the modelling of valve dynamics is still a research subject. In spite of the ever increasing availability of the hardware and software for CFD and FSI simulations, these approaches are still neither practical nor developed enough in order to replace the one-dimensional models of the sealing element dynamics used heretofore. The latter, however, are in need of further improvement in some areas, such as establishing the variation of the drag force with the flow, considering the unsteady effects at small valve opening, refining the models of the resistance to valve plate motion, etc. This can be achieved by combining the measurements with the numerical methods, i.e. the CFD and FSI simulations.

Considering the valve mass flow rate calculation, it is felt that the research of the alternatives to the discharge coefficient concept should be actively pursued. The stagnation pressure loss model, which is thermodynamically sound and widely used in the hydraulics could represent a good starting point.

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