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A Combined Experimental-Numerical Procedure to Estimate Leakage Gap of Compressor Valves

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ABSTRACT

Leakage through valves can significantly reduce the volumetric and isentropic efficiencies of compressors. Despite its importance in compressor design, the dimensional characterization of leakage gaps is not a trivial task. In this paper, we present a combined experimental-numerical method developed to estimate the leakage gap of compressor valves. Measurements of leakage were carried out via the constant volume method, which is widely employed in the analysis of gas flow through microchannels. Additionally, predictions were obtained with a one-dimensional flow model, taking into account viscous friction, slip at the walls, and gas compressibility. The leakage gap was adjusted in the simulation model so that predictions matched the measurements of leakage for different pressure differences. The procedure was applied for the analysis of three valve designs of refrigeration compressors.

1. INTRODUCTION

The efficiency of reciprocating compressors adopted in household refrigerators has dramatically increased in the last decades (Possamai and Todescat, 2004). However, further development is still necessary to allow the production of refrigeration systems in compliance with the ever-increasing energy-saving requirements. The main energy losses in compressors are usually classified in three groups: electrical losses, associated with inefficiencies of the electrical motor; mechanical losses, due to friction in bearings; thermodynamic losses, brought about by irreversibilities in the compression cycle. Leakage is a significant source of thermodynamic inefficiency and occurs in components of the compressor where high-pressure gas is not completely sealed.

Leakage in reciprocating compressors occurs through two main paths: (i) the clearance between the piston and the cylinder; (ii) the clearance between the valve and the seat. Most studies in the literature focus on leakage through the piston-cylinder clearance (Pandeya and Soedel, 1978; Ferreira and Lilie, 1984; McGovern, 1990). However, valve leakage is becoming increasingly important as compressors are manufactured with smaller dimensions.

This paper investigates leakage in valves that takes place through microchannels formed by geometric irregularities, which may result from macrogeometric characteristics, such as alignment and flatness; or from microgeometric aspects, such as surface roughness. The driving force for leakage in valves is the pressure difference between the compression and suction/discharge chambers. In addition to the valve-seat gap geometry, other aspects may contribute to leakage, namely the working fluid and valve/seat materials.

Some studies have demonstrated the relevance of valve leakage in compressor performance. For instance, Machu (1990) developed a model to account for the influence of leakage in a double acting cylinder compressor. By varying the leakage area in the suction and discharge valves, he obtained two main conclusions: (i) the volumetric efficiency can significantly drop with valve leakage; and (ii) superheating increases with valve leakage, decreasing the isentropic efficiency. Fujiwara (1998) evaluated the performance of an oil-free microcompressor and showed that a 1 μm gap in the valve would reduce the discharge pressure to half of the value that could be obtained with a completely sealed valve. Silva and Deschamps (2014) developed a simulation model to predict valve leakage in reciprocating compressors and its impact on the compressor volumetric and isentropic efficiencies. For a particular household compressor considered in their study, the authors showed that a valve-seat gap of 1 μm would reduce the volumetric and isentropic efficiencies by 2.7% and 4.4%, respectively.

Leakage in the valve-seat gap is associated with mass flow rates on the order of 10^{-5} kg/s. Under such small mass flow rates, the direct measurement of leakage is challenging and a more convenient approach is to employ accurate indirect measurement techniques used to analyze gas flow through microchannels. The three most used indirect measurement techniques are the drop method (Pitakarnnop *et al*., 2010; Ewart *et al*., 2006), the constant pressure method (Jousten *et al*., 2002; McCulloh *et al*., 1987) and the constant volume method (Yamaguchi *et al*., 2011; Pitakarnopp *et al*., 2010).

In this paper, we report an experimental setup based on the constant volume method to measure leakage in the valve-seat gap, $\delta(r)$, of reciprocating compressors, as depicted in Figure 1. As shown by Silva and Deschamps (2014), the varying gap between the valve and the seat, $\delta(r)$, can be expressed as a function of the valve bending due to pressure load and the reference gap at the edge of the valve port, δ_e, which will be denominated hereafter as 'leakage gap'. An experimentally validated model is used to estimate δ_e and hence characterize valve sealing. This combined procedure is used to analyze leakage in three valve designs under different pressure differences.

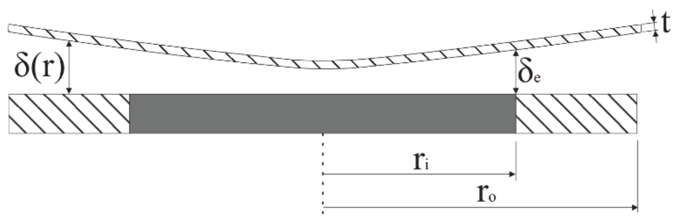

Figure 1: Cross-section of the valve system.

2. EXPERIMENTAL SETUP AND PROCEDURE

2.1 Constant Volume Method

We chose the constant volume method to measure valve leakage, since it is easier to implement than both the constant pressure method and the drop method. This method consists in monitoring the pressure p_r and temperature T_r inside a reservoir of known fixed volume V_r from which an ideal gas leaks. Hence, the mass of gas inside the reservoir, m_r, can be calculated from the equation of state for an ideal gas as a function of pressure and temperature:

$$m_r = \frac{p_r V_r}{RT_r}, \tag{1}$$

where R is the gas constant. By differentiating Equation (1) with respect to time, the mass flow rate of gas leaving the reservoir can be determined by:

$$\frac{dm_r}{dt} = -\frac{V_r}{RT_r}\frac{dp_r}{dt} + \frac{p_r V_r}{RT_r^2}\frac{dT_r}{dt}. \tag{2}$$

The equation above can be rewritten as

$$\frac{dm_r}{dt} = -\frac{V_r}{RT_r}\frac{dp_r}{dt}(1-\varepsilon), \tag{3}$$

where

$$\varepsilon = (dT_r / T_r)/(dp_r / p_r). \tag{4}$$

The parameter ε is a measure of how much the actual leakage process deviates from an isothermal process, i.e., ε = 0. Since very small temperature variations were observed in our experiments (ε < 0.01), Equation (3) was simplified to

$$\frac{dm_r}{dt} = -\frac{V_r}{RT_r}\frac{dp_r}{dt}. \tag{5}$$

It should be noted that dm_r/dt in the previous equations represent the gas leakage, which will be denoted hereafter by $\dot{m}$.

2.2 Experimental Setup

As shown in Figure 2, the test rig is composed by the tank TK, pressure gauge P, thermocouple T and five control valves V1-V5. The compressor mechanical kit with the valve plate contains the compression chamber CC, suction chamber SC and discharge chamber DC, as well as the compressor suction valve SV and discharge valve DV. The tubes connecting the components of the test rig were made of copper alloy. The volume formed by the tank, tubes, discharge and compression chambers of the mechanical kit will be referred herein as the reservoir R1, being subjected to pressure decrease due to the leakage through the valve-seat gap.

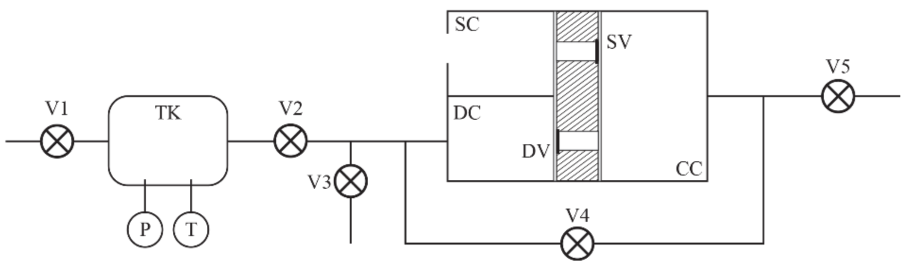

Figure 2: Experimental setup.

Leakage can be measured in one of the compressor valves (SV or DV) at a time, by modifying the setup of the test rig. The pressure in the discharge chamber (DC) does not depend on the test setup and is always equal to the pressure in the reservoir R1, varying from p_{max} to the atmospheric pressure. On the other hand, the pressure in the compression chamber (CC) can be adjusted to the pressure in the reservoir or to the atmospheric pressure, depending on the required setup. The suction chamber (SC) pressure is independent from the setup, since it is always open to the atmosphere.

The effective volume V_r of the reservoir R1 varies according to the valve being analyzed (SV or DV) and must be determined in each case. Accordingly, an auxiliary indirect measurement method was used to calculate V_r, which follows five steps: (i) Connect a second reservoir (R2) of known volume to the control valve V3; (ii) Set the pressure in the reservoirs R1 and R2 to pressure p_1; (iii) Close the control valve V3 to maintain the reservoir R2 at p_1 isolated from the reservoir R1; (iv) Increase the pressure in the reservoir R1 to p_2; (v) Open the control valve V3 to connect the reservoirs R1 and R2 again and let the system to establish the equilibrium pressure p_3 ($p_2 > p_3 > p_1$). By using Equation (1) and the aforementioned procedure, the volume of the reservoir R1 can be determined for the suction and discharge setups. Naturally, this procedure has to be applied to all suction/discharge compressor systems analyzed, since their dimensions are different.

2.3 Experimental Procedure

The control valves (V1-V5) are used to adjust the required pressure difference ($\Delta p = p_r - p_{atm}$) between the compressor chambers (SC, DC and CC). Each test consists in pressurizing the reservoir to p_{max} and then letting the gas to leak until the pressure in the reservoir $p_r = 2\ p_{atm}$. Table 1 shows the open/closed settings of the control valves for the suction and discharge setups. As can be seen, adjustments are required only in valves V4 and V5.

We took special care to manipulate the compressor components, especially valves and valve plates, since small scratches or oxidation on their surfaces may affect microscopic characteristics and, hence, the gas leakage. Therefore, these components were stored in recipients with silica gel and handled with rubber gloves.

Table 1: Control valve settings for the analysis of leakage in the suction and discharge valves.

Control valve	Suction setup	Discharge setup
V1	Closed	Closed
V2	Open	Open
V3	Closed	Closed

Control valve	Suction setup	Discharge setup
V4	Open	Closed
V5	Closed	Open

Two types of tests were carried out: (a) total leakage; and (b) secondary leakage. The first procedure measures the total leakage, i.e., leakage in the compressor valve ($\dot{m}_l$) and unwanted leakages in the experimental workbench. In fact, the compressor mechanical kit is composed of many assembled parts and leakages through different paths are prone to happen. Such leakages will be referred as secondary leakage ($\dot{m}_s$). To mitigate secondary leakage, we tightened the screws always with an adequate torque and used actual compressor gaskets with silicon treatment. To measure the secondary leakages, we removed the compressor valves from the valve plate and sealed completely the suction and discharge orifices.

New samples of valve assemblies were used to measure the total leakage ($\dot{m}$). We noticed that leakage decreased as the tests were repeated and attributed this effect to deformation of the valve assembly, which resulted in better sealing. To mitigate this effect, we repeated the tests until the measurements yielded similar results. When this condition was met, five measurements of total leakage tests were carried out and valve leakage was subsequently obtained by subtracting the secondary leakage from the average value of total leakage, i.e.,

$$\dot{m}_l = \dot{m} - \dot{m}_s. \tag{6}$$

2.4 Measurement Uncertainties

The standard uncertainty of the valve leakage is a function of uncertainties associated with the volume V_r, temperature T_r and pressure drop rate dp_r/dt in the reservoir:

$$u^2(\dot{m}) = \left(\frac{\partial \dot{m}}{\partial V_r}u(V_r)\right)^2 + \left(\frac{\partial \dot{m}}{\partial T_r}u(T_r)\right)^2 + \left(\frac{\partial \dot{m}}{\partial (dp_r/dt)}u(dp_r/dt)\right)^2 \tag{7}$$

The reservoir volume V_r was evaluated from a series of indirect measurements, and its uncertainty is associated with test repeatability. The three valve designs used in the present investigation resulted different volume standard uncertainties: $u(V_{r1})$ = 8.69×10^{-6} m³; $u(V_{r2})$ = 2.86×10^{-5}m³; $u(V_{r3})$ = 1.34×10^{-6}m³. The temperature T_r was taken as the average value along the test, in line with the hypothesis of an isothermal process. The main uncertainty source of T_r arises from oscillations in the ambient temperature and the calibration of the thermocouples, resulting a standard deviation $u(T_r)$ = 6.2×10^{-2} °C.

From experimental data of pressure in the reservoir as a function of time, local linear curve fitting was used to determine dp_r/dt for different pressure ranges. Therefore, the uncertainty of dp_r/dt is associated with this linear curve fitting with a coefficient of determination r^2> 0.99. The highest value found for $u(dp_r/dt)$ was 107 Pa/s.

As suggested by Equation (6), the valve leakage uncertainty is also affected by the secondary leakage uncertainty. Therefore,

$$u^2(\dot{m}_l) = u^2(\dot{m}) + u^2(\dot{m}_s). \tag{8}$$

Once the standard uncertainty is obtained, the expanded uncertainty $U(\dot{m}_l)$ is evaluated through a t-Student distribution:

$$U(\dot{m}_l) = t\, u(\dot{m}_l). \tag{9}$$

where t is the Student factor for a 95% confidence level with a number of degrees of freedom (ν) obtained from the Welch-Satterthwaite equation.

3. NUMERICAL MODEL

We adopted the simulation model developed by Silva and Deschamps (2014) to estimate the leakage gap at the edge of the valve port, δ_e. This section briefly explains this model and its validation based on comparisons between predictions and experimental data for flow of different gases through a metallic microtube (Silva *et al.*, 2016).

The model calculates leakage in compressor valves given the pressure difference across the valve and gas properties at the inlet. It solves the conservation equations for an ideal gas, assuming one-dimensional, steady, adiabatic, laminar flow through a channel of variable cross-sectional area. Reed valves are thin structures and the model calculates valve bending due to pressure load, which is required to define the flow geometry, as represented in Figure 1. The valve-seat gap can be regarded as a radial nozzle in which $\delta(r)$ is a function of both the local radius and the pressure load. On the other hand, the gap at the edge of the valve port, δ_e, can be regarded as the valve leakage gap. The valve-seat gap $\delta(r)$ is usually on the order of micrometer and rarefaction effects may affect the flow when δ_e< 0.25 µm, as shown by Silva and Deschamps (2014).

Since the model was originally developed to predict leakage in compressor valves, we had to modify some equations in the model to allow its validation by comparing predictions with measurements of mass flow rate in a microtube. Moreover, the fanning friction factor required in the model was corrected for slip flow condition via the following expression:

$$C_f = \left(\frac{16}{\mathrm{Re}}\right)\left(\frac{1}{1+8\left[\left(2-\sigma_v\right)/\sigma_v\right]Kn}\right), \tag{10}$$

where the tangential accommodation coefficient σ_v was obtained from Silva *et al.* (2016), Re is the Reynolds number and Kn is the Knudsen number used to characterize flow rarefaction.

The experimental setup developed by Silva *et al.* (2016) allows measurements of mass flow rate in microchannels, including tests in the continuum and slip flow regime. The experimental rig consists of two reservoirs connected by a stainless steel microtube (diameter 438.6 ± 4.5 µm and length 9.22 ± 0.01 mm).

Pressure and temperature inside the reservoirs are monitored with pressure transducers and thermocouples. In order to reproduce rarefaction flow conditions that occurs in valve leakage, the pressure inside the test section has to be decreased. This is achieved by means of a vacuum pump connected to the reservoir located downstream of the microtube. In the reservoir located upstream a tank with gas at high pressure was connected.

Numerical and experimental results of mass flow rate were compared for three different gases (Nitrogen, R134a and R600a) and pressure ratio Π = 3. The relative difference between both results is given by

$$\Delta = \left|\frac{m_{\exp} - m_{num}}{m_{\exp}}\right|, \tag{11}$$

where $\dot{m}_{exp}$ and $\dot{m}_{num}$ are the mass flow rates from measurements and predictions, respectively.

Figure 3 shows the relative difference between the results as a function of the Knudsen number. It should be mentioned that Equation (10) is valid for the continuum flow regime ($Kn < 10^{-2}$) and slip flow regime ($10^{-2} < Kn < 10^{-1}$). The agreement between numerical and experimental is acceptable for these range of Knudsen numbers, with a maximum difference of approximately 8%, and the model was assumed to be validated.

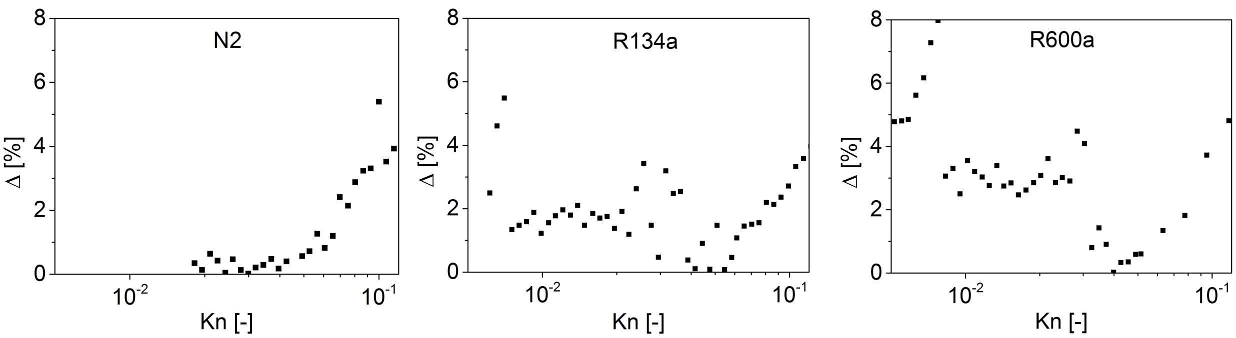

Figure 3: Difference between experimental and numerical results for mass flow rate in microtube.

4. RESULTS

We tested three valve system designs (D1, D2 and D3). The valve system D1 is adopted in a high-capacity compressor, and the valve systems D2 and D3 are employed in low-capacity compressors. We used nitrogen as the working fluid at the ambient temperature (T = 23 °C). The pressure differences set across the valves correspond to conditions typically found in refrigeration compressors.

4.1 Mass Flow Rate Measurements

Figures 4 to 7 present experimental (black squares) and numerical (gray circles) results for valve systems D1, D2 and D3. Measurements of leakages were made dimensionless with respect to the maximum leakage, $\dot{m}_1^*$, measured in the valve system D1. Figure 5 shows that the secondary leakages of valve system D1 is small compared to the valve leakage itself, not exceeding 7% of the valve leakage.

Figure 6 indicates that the leakage in the valve system D2 is much higher than the leakages in the other two valve systems (D1 and D3). Naturally, leakage has a greater impact on the efficiency of low-capacity compressors. Since the valve system D2 has the worst sealing and is adopted in a low-capacity compressor, a significant reduction of efficiency is expected.

An interesting behavior is observed in the measurements of leakage for the valve system D3. As can be seen in Figure 7, after reaching a local maximum at $\Delta p/\Delta p_{max} \cong 0.3$, the leakage is decreased to a local minimum at $\Delta p/\Delta p_{max} \cong 0.5$, before increasing again with the pressure difference. This behavior may be the result of combined effects in the valve system D3, such as valve bending and microgeometry interactions between the valve and seat surfaces.

4.2 Estimate of Leakage Gap

The gray circles in Figures 4, 6 and 7 show estimates of the leakage gap (δ_e) in the valve systems for different pressure differences. Such estimates were obtained by adjusting δ_e in the simulation model so as to make the predicted leakage equal to the measurement. The leakage gap can be regarded as a convenient parameter to assess the sealing of valve systems. However, caution should be taken when comparing valve systems of different dimensions, since the leakage area is as function of the valve orifice diameter.

The smallest leakage gaps were found for the valve system D1 while the valve system D2 had the greatest leakage gaps. Hence, it is natural the higher levels of leakage in the valve system D2, $\dot{m}_{L_2}$, in comparison to that of the valve system D1, $\dot{m}_{L_1}$, as shown in Figure 6. For instance, considering the same pressure difference ($\Delta p/\Delta p_{max} \cong 0.4$), the ratio between the leakage gaps of these two valve systems $(\delta_e)_2/(\delta_e)_1 \cong 8$ and $\dot{m}_{L_2} \cong 20\,\dot{m}_{L_1}$. It is worthwhile to note that the leakage gap decreases in all three valve systems as the pressure difference increases, suggesting that valve sealing is improved with the pressure load on the valve.

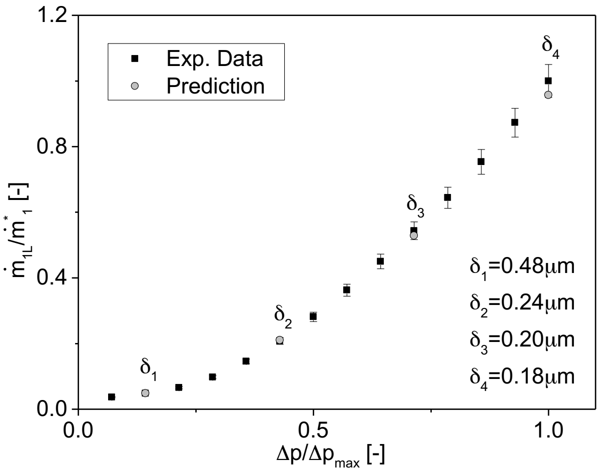

Figure 4: Leakage in the valve system D1.

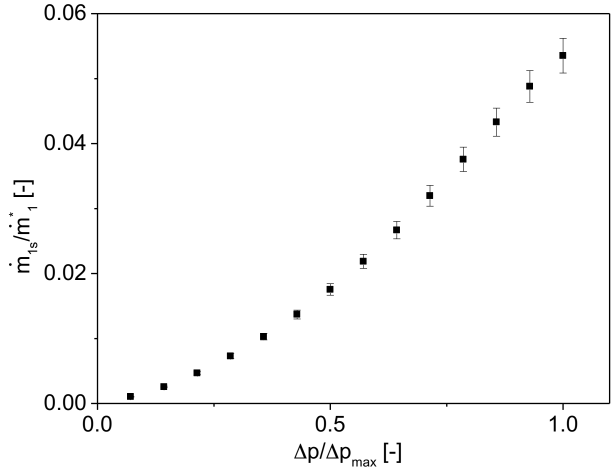

Figure 5: Secondary leakage in the valve system D1.

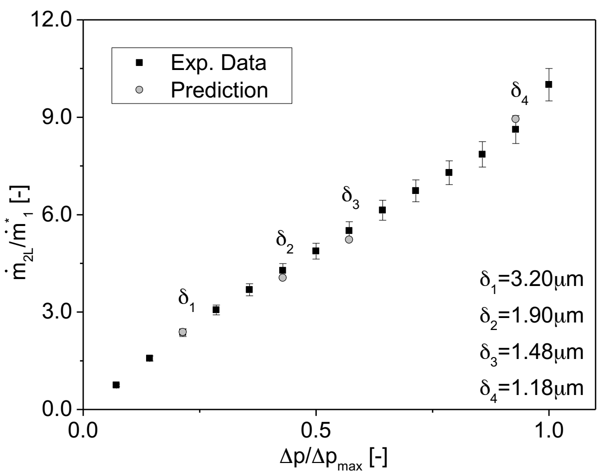

Figure 6: Leakage in the valve system D2.

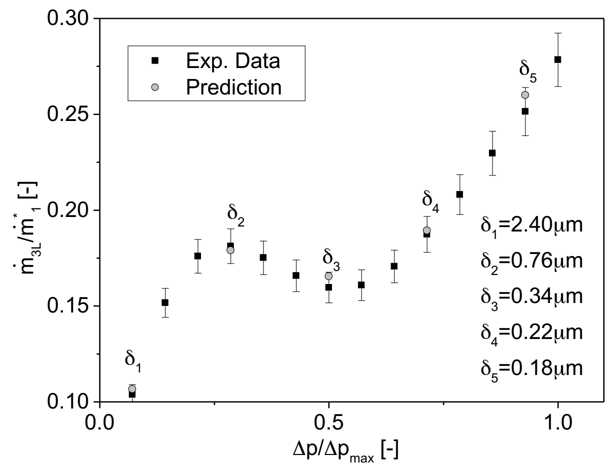

Figure 7: Leakage in the valve system D3.

5. CONCLUSIONS

Valve leakage can significantly reduce the efficiency of small reciprocating compressors, but the dimensional characterization of leakage gaps is not a trivial task. We developed and applied a combined experimental-numerical procedure to estimate the leakage gap of compressor valves. Measurements of leakage were carried out via the constant volume method and predictions of leakage gap were obtained with an experimentally validated simulation model, considering viscous friction, slip at the walls, and gas compressibility. The procedure was applied for the analysis of three valve designs, with leakage gaps being found in the range from 0.18 to 3.2 μm. We found that the leakage gap can be strongly affected by valve design and is reduced with the pressure load on the valve, probably due to valve bending and interactions between the valve and seat surfaces.

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