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EXPERIMENTAL ASSESSMENT OF LEAKAGE IN COMPRESSOR VALVES

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Abstract. *Refrigeration compressors commonly adopt automatic valves, which open and close due to the pressure difference between the compression and suction/discharge chambers. The specification of such valves is critical in the design of a high efficiency compressor. In addition to reliability issues, incomplete valve sealing due to surface irregularity or misalignment gives rise to gas leakage, which significantly reduces the compressor volumetric and isentropic efficiencies. This paper reports an experimental setup developed to characterize leakage in valve designs typically adopted in hermetic reciprocating compressors.*

Keywords: *Refrigeration compressor, Valve, Leakage*

1. INTRODUCTION

The efficiency of reciprocating compressors adopted in household refrigerators has dramatically increased in the last two 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 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 loss 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) clearance between the piston and cylinder; ii) clearance between the valve and seat. Most studies in the literature focus on leakage through the piston-cylinder clearance. In lubricated compressors, gas may leak directly by breaking the oil film between the piston and cylinder, or indirectly by dissolving itself in the oil flowing through the clearance (Pandeya and Soedel, 1978; Ferreira and Lilie, 1984; McGovern, 1990).

This paper investigates leakage between the valve and seat, through microchannels formed by geometric irregularities on both surfaces, as depicted in Fig. 1. These irregularities result from macrogeometric characteristics of these components, such as alignment and flatness; or from microgeometric aspects, such as surface roughness. The resulting clearance will be referred herein as valve-seat gap. The driving force for leakage in valves is due to the pressure difference between the compression and suction/discharge chambers. In addition to the valve-seat gap geometry, other features contribute to leakage, namely working fluid and valve/seat materials.

Some studies have demonstrated the relevance of valve leakage in compressor performance. For instance, Machu (1990) developed a mathematical 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, Machu (1990) concluded that: i) volumetric efficiency drops with valve leakage; ii) superheating increases with valve leakage, decreasing the isentropic efficiency. Silva and Deschamps (2014) developed a simulation model to evaluate 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 1μm may reduce the volumetric and isentropic efficiencies by 2.7% and 4.4%, respectively.

Leakage in the valve-seat gap is a micro flow, with mass flow rates of the order of 10^{-5}kg/s. Under such small mass flow rates, the direct measurement of leakage is hampered, but accurate indirect measurement techniques, commonly used to analyze gas flows through microchannels, can be employed. 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*., 1986) and the constant volume method (Yamaguchi *et al*., 2011; Pitakarnopp *et al*., 2010). The constant volume method monitors pressure drop in a constant volume reservoir to measure mass flow rate. This paper reports an experimental setup, based on the constant volume method, developed to measure leakage in the valve-seat gap of refrigeration compressors.

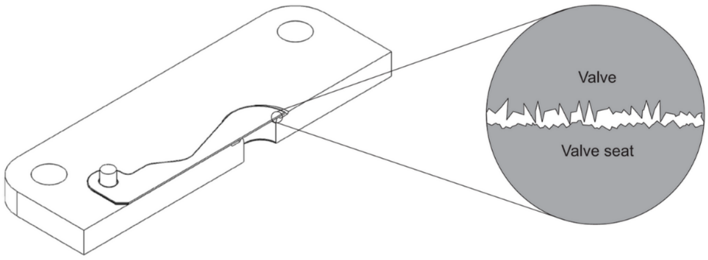


Figure 1. Microchannel formed between the valve and the valve seat due to surface roughness.

2. EXPERIMENTAL RIG AND PROCEDURE

2.1 Constant volume method

The constant volume method was applied in the present experimental investigation, since it is easier to implement than both the constant pressure method and the drop method. In the constant volume method, the variations of pressure p_r, and temperature T_r are measured inside a reservoir of known volume V_r to evaluate its leakage. Nitrogen was used as the working fluid, and considered an ideal gas at the pressure and temperature ranges of the experiment. Hence, gas mass m_r inside the reservoir can be calculated from the equation of state for an ideal gas as a function of pressure and temperature:

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

where R is the gas constant. By deriving Eq. (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 simplified to:

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

with

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

The parameter ε represents how much the leakage measured in the experiment deviates from an isothermal process, where $\varepsilon = 0$. Small temperature variations were observed in the experiments ($\varepsilon < 0.01$), and Eq. (3) can be simplified as follows:

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

Hereafter, dm_r/dt will be denoted by $\dot{m}$.

2.2 Test rig

Figure 2 represents the main components of the test rig: tank (Swagelok®, 304L-HDF4-500); pressure gauge PG (WIKA®, type P-30); type T thermocouple TC; valves V1-V5 (Norgren®, Herion 95000). The compressor mechanical kit 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 is referred as the reservoir (R1), which is subjected to a pressure decrease associated with the leakage through the valve-seat gap.

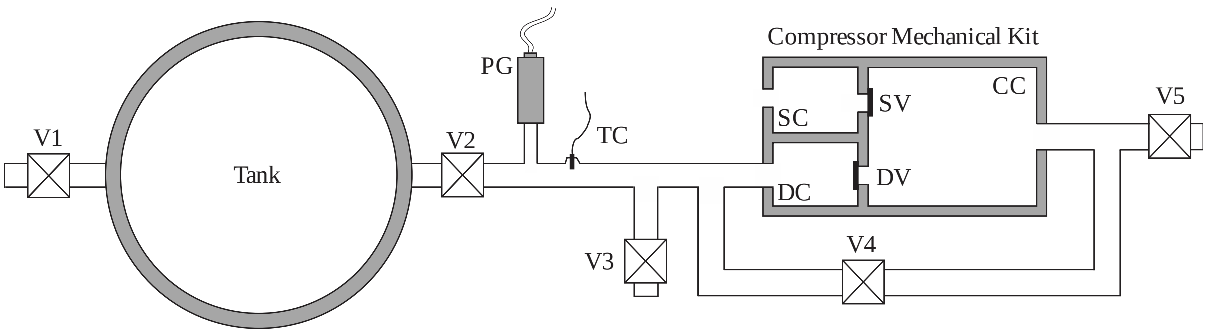


Figure 2. Experimental rig.

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 discharge chamber (DC) pressure does not depend on the test setup and always has the same pressure of the reservoir, which ranges from p_{max} to the atmospheric pressure. On the other hand, the pressure in the compression chamber (CC) may be adjusted to the pressure in the reservoir or atmospheric pressure, depending on the test to be carried out. The suction chamber (SC) pressure is independent from the setup, since it is always open to the atmosphere.

The effective volume of the reservoir, V_r, 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 can be described as follows: (i) Connect a second reservoir (R2) with a known volume to valve V3; (ii) Set the pressure in the reservoirs R1 and R2 to pressure p_1; (iii) close 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 valve V3 to connect the reservoirs R1 and R2 again and let the system establish the equilibrium pressure p_3 ($p_2 > p_3 > p_1$). By using Eq. (1) and the aforementioned procedure, the volume of the reservoir R1 can be determined for the suction and discharge setups. Naturally, this procedure had to be applied to all suction/discharge compressor systems analyzed, since their dimensions are different.

2.3 Experimental procedure

The valves (V1-V5) allow the adjustment of the required pressure difference ($\Delta p = p_r - p_{atm}$) between the compressor chambers. The reservoir R1 was pressurized to the maximum value of pressure (p_{max}) and left leaking until its pressure reach $0.2p_{max}$. Table 1 shows the open/closed settings of the valves for the suction and discharge setups. As can be seen, adjustments were required only in valves V4 and V5. The corresponding pressures in the chambers SC, DC and CC are indicated in Fig. 3 for the suction and discharge setups.

Special care was taken when manipulating the compressor components, especially valves and valve plates, since small scratches or oxidation on the surfaces may affect microscopic characteristics and, hence, leakage. Therefore, these components were stored in recipients with silica gel and handled with rubber gloves to avoid oxidation.

Table 1. Valves settings for the analysis of leakage in suction and discharge valves.

Valve	Suction valve leakage	Discharge valve leakage
V1	Closed	Closed
V2	Open	Open
V3	Closed	Closed
V4	Open	Closed
V5	Closed	Open

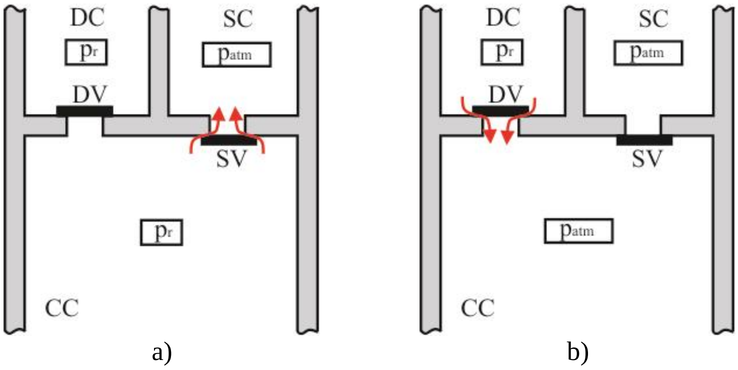


Figure 3. Valve leakage and pressure inside the chambers. (a) Suction valve setup. (b) Discharge valve setup.

2.4 Secondary leakages

Leakage was also detected through the joints of the mechanical kit that were designed to secure sealing. Attention was taken to tighten screws always with adequate torque and to use joint material of actual compressors to protect metal surfaces from damaging one another when tightened together. Since the joint material has high porosity, silicone was used to reduce leakage to a minimum.

Leakage, in a test run, represents the total leakage ($\dot{m}$) in the experimental setup, including the leakage in the valve-seat gap ($\dot{m}_{vl}$) and secondary leakage in the joints ($\dot{m}_{sl}$). Therefore, the secondary leakage had to be obtained to allow measurement of the actual leakage in the valve-seat gap. Accordingly, the compressor valves were removed and the suction and discharge orifices were completely sealed. Then a new test was run, and the secondary leakage quantified for different pressure values in the reservoir. Valve leakage was obtained by subtracting the secondary leakage from the total leakage, i.e.,

$$\dot{m}_{vl} = \dot{m} - \dot{m}_{sl}. \tag{6}$$

2.5 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:

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

The reservoir volume V_r was evaluated from a series of indirect measurements, and its uncertainty is associated with test repeatability. Each compressor model resulted different volume standard uncertainties: $u(V_{r1}) = 2.86 \times 10^{-5}$m³; $u(V_{r2}) = 8.68 \times 10^{-6}$m³; $u(V_{r3}) = 4.03 \times 10^{-6}$m³. The reservoir 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 comes from oscillations in the ambient temperature, and its maximum standard deviation during the tests was $u(T_r) = 0.31°C$.

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 that of the linear fitting coefficient, following a coefficient of determination r^2>0.99. The highest value found for $u(dp_r/dt)$ was 256Pa/s.

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

$$u^2(\dot{m}_{vl}) = u^2(\dot{m}) + u^2(\dot{m}_{sl}). \tag{8}$$

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

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

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. RESULTS

Figure 4 shows experimental data of pressure decrease in the reservoir associated with leakage in three different suction valve designs adopted in refrigeration compressors. The pressure drop with time in the reservoir is the most important data to calculate valve leakage. Figure 4a shows pressure drop in the reservoir associated with total leakage ($\dot{m}$), whereas Fig. 4b depicts the pressure drop caused by secondary leakage ($\dot{m}_{sl}$) alone. It is important to note that the total leakage gives rise to a much steeper pressure decay than that resulting from secondary leakage. For instance, the test of secondary leakage for the valve system 1 took almost 25 hours, whereas the test of total leakage ran for approximately 1.3 hours.

Figure 4a shows that different valve-seat designs (1, 2 and 3) present quite different leakages, with $\dot{m}_1$<$\dot{m}_3$<$\dot{m}_2$. The valve system 2 brings about the highest pressure drop rate and therefore the worst sealing capability. It is also interesting to note that the valve system 3 shows the smallest pressure drop rate in the initial stage of the leakage test, but ends up emptying the reservoir earlier than the valve system 1. This indicates that the valve system 1 leaks less than the valve system 3 at lower pressure differences, which can be a consequence of reed valve deflection due to pressure load. In fact, valve geometry and manufacturing processes may considerably affect reed valve deflection.

Figures 5-7 show data of valve and secondary leakage for the same three suction valve designs. In each figure, leakage is made dimensionless with reference to the maximum leakage in the valve system 3 ($\dot{m}_3^*$). In general, large compressors present greater leakage, since valve-seat gaps are larger. However, the present results indicate that leakage rates in the valve system 2 is almost ten times larger than those in the valve systems 1 and 3, which are employed in compressors of greater capacities. Therefore, leakage through the valve system 2 affects more the efficiency, and efforts should be directed to improve its design.

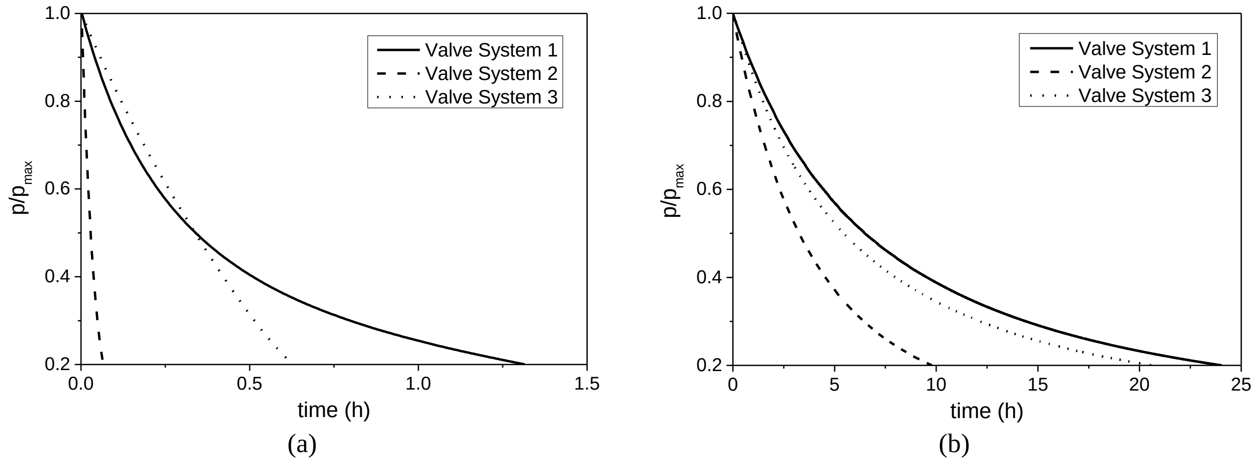


Figure 4. Pressure drop in the reservoir. (a) Total leakage ($\dot{m}$). (b) Secondary leakage ($\dot{m}_{sl}$).

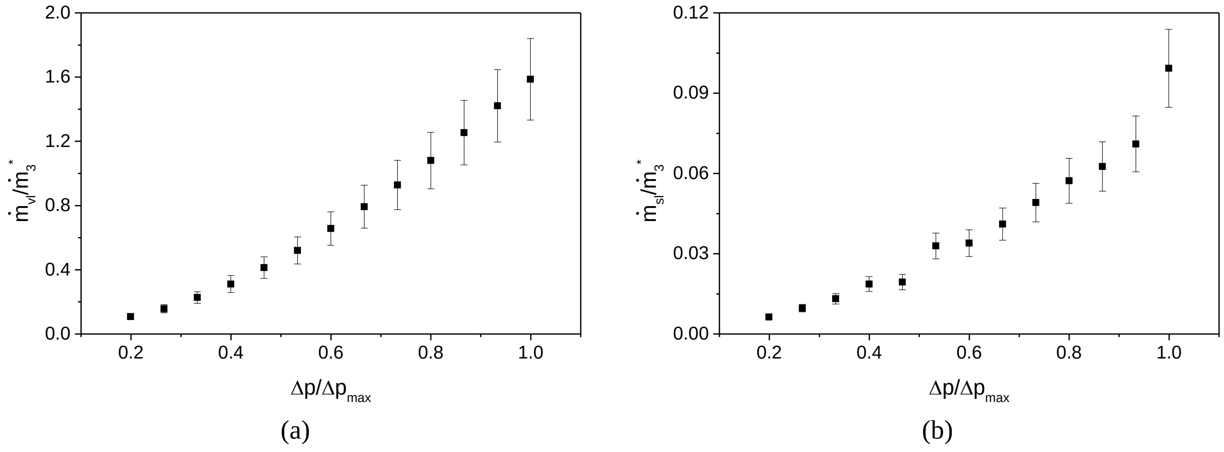
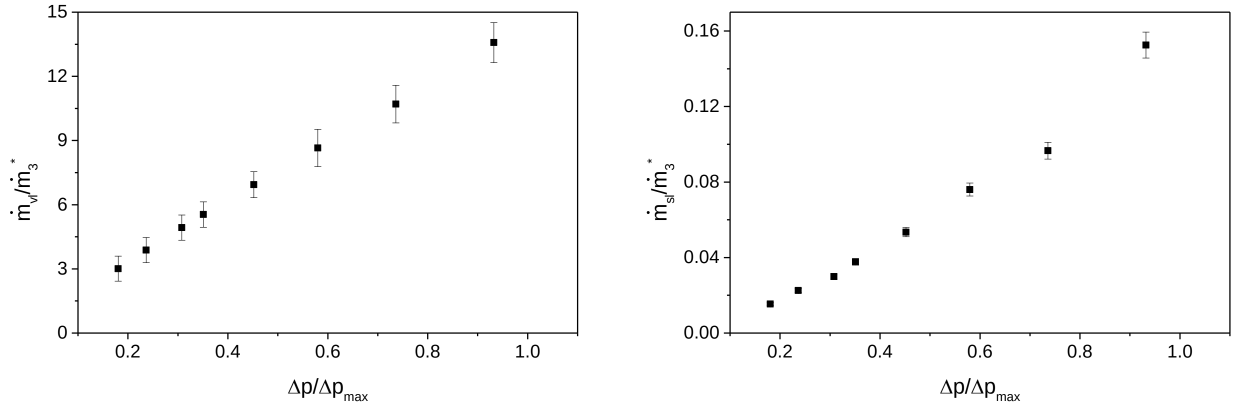
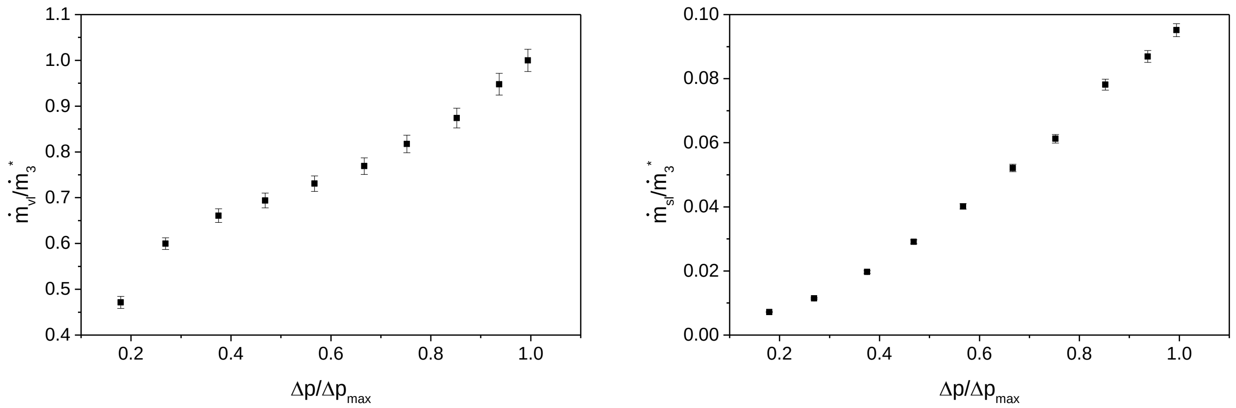


Figure 5. Valve system 1. (a) Valve leakage ($\dot{m}_{vl}$). (b) Secondary leakage ($\dot{m}_{sl}$).



(a) (b)

Figure 6. Valve system 2. (a) Valve leakage ($\dot{m}_{vl}$). (b) Secondary leakage ($\dot{m}_{sl}$).



(a) (b)

Figure 7. Valve system 3. (a) Valve leakage ($\dot{m}_{vl}$). (b) Secondary leakage ($\dot{m}_{sl}$).

4. CONCLUSION

An experimental rig was developed to allow measurements of leakage through clearances in valves of refrigeration compressors. The constant volume method was used to measure leakage and proved to be reliable and easy to implement with commercially available pressure transducers. The experimental setup was found suitable to analyze the effect of valve design on leakage for different compressor models. The results showed that leakage is strongly dependent on the valve-seat design. Future studies will consider the effect of different refrigerant fluids on valve leakage and a more detailed analysis about the effect of leakage on the compressor performance.

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6. RESPONSIBILITY NOTICE

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