2022

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## ABSTRACT

Due to the changing energy regulations of domestic refrigerators, the performance of hermetic reciprocating compressors becomes very crucial. Therefore, the number of refrigerators on which the variable capacity compressors are used, increase considerably in the last decade. Previous investigations show that there is still higher potential on thermodynamic efficiency to increase the compressor's coefficient of performance (COP). Gas leakages are also an important parameter affecting the capacity so the COP of the compressor. Variation of the gas leakages with respect to the operating conditions needs to be investigated to optimize the piston-cylinder bearing design and improve the volumetric efficiency of the compressor.

In this study, a gas leakage measurement setup was established. The compressor to be tested is modified with a fully direct suction connection. All possible leakages between the suction line and the compressor shell are avoided. Due to the modified connection of the pipes, the risk of oil suction into the cylinder was too high. To avoid this risk an oil separator is placed in the suction line before the compressor. In this way, however, after running for a certain period the oil level in the compressor will decrease. The oil accumulator, therefore, was constructed from glass to be able to observe the oil quantity accumulated.

The service connection of the compressor shell is connected to a large vessel and that vessel is again connected to the suction line of the compressor via a valve. The total system is placed on a compressor calorimeter to simulate the ASHRAE test conditions. After starting the compressor and waiting until all parameters have stabilized (suction pressure, condensation pressure, compressor shell temp., and vessel temperature), the valve between the vessel and the suction line is closed. The suction pressure is kept constant by the calorimeter, but due to piston leakage, the pressure in the compressor shell and the vessel starts to rise. By measuring this pressure increase in time the piston leakage can be calculated. Measurements were conducted at different rotational speeds and various operating conditions.

## 1. INTRODUCTION

Compressor leakages that occur due to the pressure difference between the compression chamber can importantly affect the efficiency of small reciprocating compressors. Coulomb et. al. (2005), stated in the 29th IIR informatory note that, air conditioning and refrigerating systems are responsible for 17% of the overall electric consumption used worldwide and nearly 85% of this energy is assigned to commercial and household applications. An important part of the refrigeration systems employed in such applications is driven by hermetic reciprocating compressors and, due to that, energetic efficiency is one of the main drivers in the design of these refrigerating machines. Pérez-Segarra et al. (2005), remarked the energy losses in the compressor as (i) electrical losses caused mainly by ohmic resistance in the electric motor; (ii) mechanical losses generated by friction in the bearings, and (iii) thermodynamic losses related to irreversibility during the compression of the gas. In hermetic compressors used in refrigerators, the piston design, and piston geometry directly affect the COP due to the compressor capacity. Gas leakages that occur in the gasket and piston during compression work are one of the important reasons that reduce the compressor capacity. The crank-connecting rod mechanism converts the rotational motion of the electric motor into linear motion. During the transition from this rotational motion to linear motion, the piston moves from the center axis of the cylinder to the cylinder wall in addition to the linear motion and hits the wall. This movement of the piston is called the "secondary motion". In compressor pistons, gas leaks due to rotational movement (tilt motion) on the pin axis and piston secondary movement that will occur as a result of leakage from the cylinder axis can reach higher orders. For this reason, the piston geometry is one of the most important parameters that determine the gas leakages that occur during compression work. Secondary motion affects the noise level of the compressor and gas leaks. A good understanding of the secondary motion of compressors is needed to develop quieter and more efficient cooling devices. Reciprocating compressors without rings are used in refrigerators. In these devices, the piston affects important parameters that determine the performance, such as the compression ratio of the machine, noise, and energy savings. The reason why the piston is so effective is that the piston dynamic characteristics, i.e., gas leaks, piston strokes, and friction losses, affect performance. The piston, moving linearly along the lubricated cylinder bore, makes a rotational movement in the radial protrusion and oil layer cavity due to unbalanced loads. Secondary motion, which enters into piston dynamics, has been a critical issue in improving performance and stability. Gas leakage through the clearance between piston and cylinder is a major source of thermodynamic inefficiency in household reciprocating compressors. Besides the nominal clearance between piston and cylinder, geometric characteristics resulting from the manufacturing process, as roundness and cylindricity errors, and the dynamics of the piston as well, are factors that can significantly impact the gas leak through the piston-cylinder gap. Many of these factors are not considered in the numerical models commonly used to estimate the leakage through the piston-cylinder clearance. Analytical models are usually too simplified, while the computational cost of more complex differential models including these effects is still too high. From the experimental point of view, many parts, the high testing time, and the experimental uncertainties make this kind of analysis very expensive.

Many studies have been published regarding simulation models and experimental techniques to detach leakages in compressors. Cho and Moon (2004) are investigated the parameters that affect the secondary motion of the piston that is seen more effectively on performance. The secondary motion of the piston is influenced by many design parameters such as radial clearance, oil viscosity, pin port, crank leakage, and piston linear profile. In this type of multi-parameter engineering problem, the effective parameters should be carefully determined by investigating them using appropriate numerical methods. Prata (2000), investigated the position of the piston pin on the leakages. Undesirable piston vibration occurs when the pin position is placed lower than the axis of the piston. It has been stated that by shifting the pin connection point in a direction perpendicular to the linear motion of the piston, the moments that disrupt the piston balance can be reduced and the piston stability can be increased. Silva and Dutra (2021), applies a simulation model based upon an unsteady lumped parameter formulation of the mass and energy conservation equations to predict the piston trajectory that accommodates the highest thermodynamic efficiency for a reciprocating compressor, disregarding the constraints related to the driving mechanism. The optimum piston trajectory improved the thermodynamic efficiency from 88.3% to 92.1%, and the volumetric efficiency from 70.9% to 72.0%, primarily by reductions of the heat transfer and leakage losses. Cervelin et. al. (2019), focused to solve piston-cylinder leakages by proposing a concept of connecting rod instrumented with a strain gauge tee-rosette bridge and the proper signal conditioning method to measure the friction forces between the piston and the cylinder. A small-sized sintered connecting rod was used to confirm if the proposed method applies to a wide range of equipment. The obtained results are at least equivalent to the ones presented in the literature, but the suggested method uses a simpler setup and allows the test to be done using the motor in its application condition especially for high-frequency phenomena. Braga and Deschamps (2018), studied a numerical analysis of leakage in the piston-cylinder clearance of low-capacity reciprocating compressors based on the Reynolds equation for compressible fluid flow. A simulation model is developed and applied throughout the compression cycle to evaluate the effect of the clearance geometry, piston velocity, and piston secondary motion on the leakage and compressor performance. During numerical solutions compressibility effect of refrigerant gas is considered and it was found that the piston secondary motion can increase gas leakage by 90%. Lohn and Pereira (2014), analyzed the impact of simplifications using a three-dimensional computational fluid dynamics (CFD) model, including transient effects resulting from the piston movement as well as actual geometric fault characteristic of the usual manufacturing processes adopted for the household reciprocating compressor industry. Simpler models and their benefits are emphasized. Schreiner et. al. (2010), considered a systematic analysis of volumetric inefficiency sources in reciprocating compressors. The procedure is based on the efficiency withdrawal approach discussed by Pérez-Segarra et al. (2005) but includes a more detailed evaluation of inefficiencies associated with the compression cycle. An appraisal of several volumetric inefficiency sources is carried out regarding results for the thermodynamic process inside the cylinder, fluid flow through valves, piston-cylinder clearance leakage, gas pulsation in mufflers, and refrigerant thermophysical properties. Xin et. al. (2010), investigated a mathematical model to find the key factors influencing the lifetime of piston rings by using by simulating unsteady flow within the gaps of piston rings was established to present the pressure distribution and disclose the mechanism of the uneven abrasion of different rings. Several non-uniformities of the pressure distribution between piston rings were suggested to be the root cause of premature failure of the sealing rings. The pressure distributions between the rings were always dramatically nonuniform with the same ring cut, and the first ring afforded more than 75% of the total pressure difference. Frictional heat generation and conduction between piston rings and cylinders by the Finite Element Method is also investigated. Braga and Deschamps (2017), studied a model based on the Reynolds equation for compressible fluid flow to predict leakage in oil-free reciprocating compressors. The model is solved throughout the compression cycle to evaluate the effect of the clearance geometry and piston velocity on leakage and compressor efficiency. The compressibility effect of refrigerant fluid flow formulation must be considered for predictions of gas leakage in the cylinder-piston clearance. Silva and Deschamps (2015), recite a model evolved to predict gas leakage in cases of incomplete sealing of the reed-type valves of small reciprocating compressors adapted for household refrigeration. The one-dimensional formulation for the flow, considering the effects of viscous friction, slip-flow regime, and compressibility effects is considered. Leakage significantly reduces the compressor efficiency even for very small valve clearances and that leakage in the discharge valve is of greater importance than that in the suction valve. Dos Santos et. al. (2015), evaluated the influence of break-ins on the leakage through reed-type valves of low-capacity refrigeration compressors. The leakage was measured for the suction and discharge valves of an R600a compressor before and after the valves were subjected to a break-in period. The leakage was reduced by up to 76% for the suction valve and by up to 95% for the discharge valve after break-in, while the edge gap was reduced by up to 50% for the suction valve and by up to 73% for the discharge valve. Reed and seat surfaces of the suction valve had their maximum roughness reduced by 40% and 29%, respectively, after break-in, suggesting that the edge gap is related to the surface finishing. Dos Santos et. al. (2018), presents a procedure developed to analyze experimentally gas leakage in reed-type valves considering the wear effect. Leakage is measured in the valves of a small-capacity reciprocating compressor before and after the valves are subjected to wear, with the roughness of the reed and seat being measured in both conditions in a constant volume. The edge gap was found in the range between 0.13 and 0.94 μm before wear and between 0.11 and 0.47 μm after wear. Reed and seat roughness was reduced by approximately half after wear. Rezende et. al. (2018), developed a constant volume method to measure leakage in suction and discharge compressor valves independently. The valve leakage is measured for three different valve designs and the concept of edge gap is proposed as a useful parameter to characterize the sealing performance of such valves and to predict the resulting leakage of different refrigerants. Leakage can vary considerably with valve design, with typical edge gap values being between 0.11 and 0.61 μm.

The studies available in the literature on the effects of piston secondary motion are mainly focused on the fluid film lubricating in the piston-cylinder clearance, and few of them predict gas leakage either by using an incompressible fluid formulation (Rigola et al., 2009). In the scope of this study, compressor leakages were investigated experimentally by making the necessary changes to the calorimeter assembly located in the Arçelik R&D Center to examine the gas leakages that occur during compression work in the compressor. During the studies, leakage rates were examined by examining two different compressors comparatively. Suniso 1GSA (10cst) oil is used in compressors during manufacturing. During these studies, the compressor oil was replaced with Shell WE 7125 (7CST) oil to determine the change in gas leaks. In this way, the low viscosity oil shows the change in gas leaks that occur in the compressor when using it. Another parameter that affects the secondary movement of the piston is oil viscosity. The effect of oil viscosity, which was not taken into account during numerical studies, was considered in the experimental study. Piston leakage tests were carried out using three different viscosity oils on a compressor. In the experiments performed with 10 cSt, 7cSt, 5cSt oils, the effect of oil viscosity on piston leakage was tried to be determined. The results show the change in piston gas leaks depending on the viscosity.

## 2. EXPERIMENTAL SETUP AND PROCEDURE

In compressors, the refrigerant coming from the suction pipe is sucked into the cylinder and then compressed by piston movement and goes to the condenser. But before some of the trapped gas reaches the exhaust pipe the piston exits from places such as seals and returns to the compressor. Gas leakages that occur during compression work directly affect the COP due to the thermodynamic efficiency of the compressor affects. Leakages that occur during compression work are difficult to measure. Figure 1 shows schematically the device used to determine gas leakages proposed by the calorimeter test system manufacturer company. The experimental setup is being created by making some changes to measure gas leaks in a compressor calorimeter. In principle, the refrigerant from the evaporator is sent directly to the cylinder (direct suction) without being enclosed in the housing. In this case, a direct connection is provided between the intake pipe and the cylinder. The gas compressed by the movement of the piston in the cylinder passes through the exhaust port and muffler and is sent to the condenser via a vibration pipe. At the same time, piston leaks that do not go to the exhaust pipe and return from the gasket and piston surroundings remain in the housing and increase the pressure of the chamber. By measuring this pressure change, the amount of piston leakage can be calculated. The oil flowing with the fluid in the system will return to the compressor and enter the cylinder chamber by direct suction. In order to prevent errors that may occur as a result of this situation, an oil bracket has been placed before the compressor in the system. The gas leaks that occur during compression work cause the pressure inside the compressor to increase over time. An additional chamber is connected to the compressor chamber so that the increased pressure during the experiments does not cause the change of piston leaks during the measurements. In order for the calorimeter to work normally until the compressor enters the regime at the desired condensation and evaporation temperatures, the suction line and the additional chamber are connected to each other by a valve.

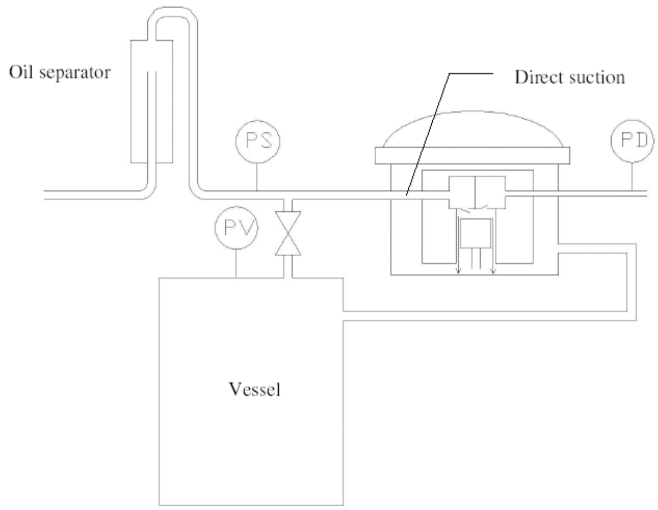
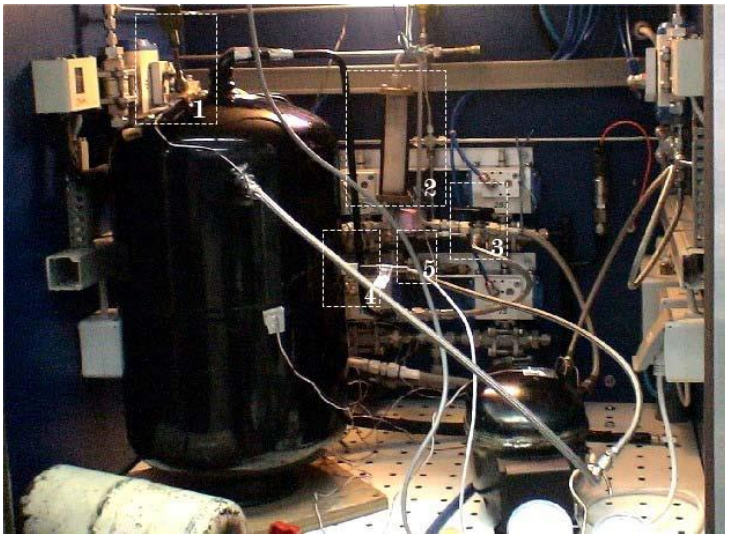



**Figure 1:** The principle of measuring piston leakages

In order to measure compressor piston leaks, the compressor must be prepared in the form of direct suction (direct suction). Therefore, the necessary changes have been made to the compresses that are not in the direct suction structure. Firstly, the compressor's muffler is cut to use the rest of the cylinder head. Cold welding (J-B Weld) is performed by placing a copper pipe in the section remaining in the cylinder head of the muffler. In addition to this, the suction pipe is extended into the compressor. A bellows was used to connect this copper pipe and the copper pipe in the compressor. It is extremely important that there are no leaks in the connections. For this purpose, a direct suction leakage test (7 min for a change of 0.8 -1 bar, 80 seconds for a change of 1.4-1.2 bar) is performed.
A certain time is required for the system to enter the regime (suction pressure, condenser temperature, compressor, and chamber temperatures are in equilibrium). During this time, the oil circulating in the system will remain in the bracket and will not go to the compressor. In order for the oil in the compressor not to decrease, the oil in the bracket should be returned to the compressor. For this, a connection has been established between the bracket and the compressor. This connection is closed with a valve during the experiment. This connection is shown in Figure 2. In order to measure the chamber temperature and pressure, holes were drilled on the chamber as shown in Figure 2 and the pressure sensor and RDT were placed. In addition to the pressure sensor installed in the system and a knee-mounted computer with a Hydra, Fluke data collector was used to add data from the RDT. In order for the received data to be consistent with each other, the Fluke and the calorimeter computer have been synchronized.




1: Pressure Transducer, 2: Oil Separator, 3: Suction pipe & chamber valve, 4: RTD, 5: Oil Pipe & Compressor Valve

**Figure 2:** Compressor gas leaks test assembly

### 2.1 Calculation of Piston Leaks

In order for piston leaks to be found, it is necessary to process the data obtained experimentally. During this data processing, the ideal gas equation (1) was used. According to the equation, it is necessary to know the temperature, pressure, and volume values in order to find the mass of the gas in the compressor and chamber. The temperature and pressure values were obtained with the placed sensors, while the volume of the compressor was obtained in different ways. It is proposed to use Boyle's Law (P*V=constant) to determine the compressor volume. But the volume is determined by the method of filling water into the compressor, which is easier. the gas volume of the tested compressor is 1.55 liters. R600a refrigerant gas was used in the measured compressors. The molar mass of this gas is 58.08g/mol.

$$P * V = n * R * T \tag{1}$$

$$M = n * Molar\ Mass = \frac{P * V}{R * T} \tag{2}$$

In the piston leak test, the gas mass in the compressor and the chamber is measured by taking the data before and after the system is turned off. The gas mass difference between them comes from the piston leaks. Piston leakage flow rate was determined by dividing by time. The ratio of the piston leakage flow rate to the flow rate in the system determines the piston leakage percentage.

## 3. RESULTS AND DISCUSSION

In order to determine the gas leaks that occur during the operation of the compressor, first of all, necessary changes were made in the calorimeter test system and the system was arranged to measure the gas leaks. Leakages of the tested compressor 1 are determined under ASHRAE conditions. Then, measurements were made for the selected tested compressor 2 for comparison, and piston leakages were determined under different evaporation and condensation conditions of the tested compressor 2, except for ASHRAE conditions. After the compressors are made to make direct suction, they undergo internal leakage and bellows leakage tests. Although the leaks seen during internal leakage are generally from the gap between the piston and the cylinder hole, they can also be from the gasket and connection points. However, it has been observed that the most dominant part of the leaks is around the piston.
The gas leakage flow rate determined in the experiments performed with compressor 1 and 2 are given in Figure 3 below. Measurements were carried out under ASHRAE conditions. The mean leakage flow rate is 0.00445 g/s and 0.0054 g/s for compressor 1 and 2 respectively. When the results are examined, it is seen that the gas leakage flow rate in compressor 1 is higher than compressor 2. Since the piston geometries are different in compressor 1 and 2 compared, leaks are also changing. In order to make a more accurate comparison, the mass flow rates of the leaks were examined. The capacity of compressor 2 is about twice that of compressor 1. In this case, it is seen that the flow rate of leaks in compressor 2 is twice that of compressor 1.

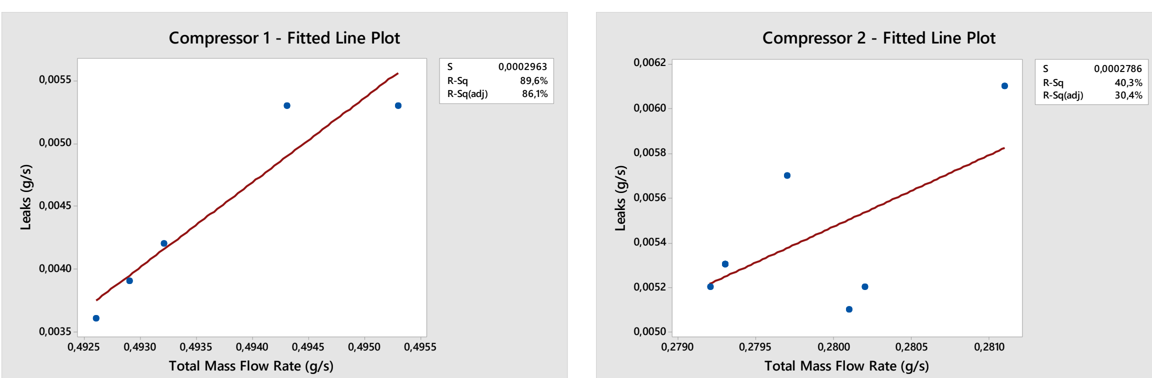



**Figure 3:** Compressor gas leaks comparison

As seen in Figure 4, piston leakages increase depending on the increasing evaporation and condensation temperature. It is seen that the piston leakages increase linearly depending on the increasing evaporation temperature for the same condensing temperature. At constant evaporation temperatures, gas leaks increase depending on the pressure due to the increasing condensation temperature. Due to the increasing condensing temperature, the pressure and the compressor outlet pressure increase, and accordingly, an increase in gas leaks is determined. In experiments performed at constant condensation temperature, the change in evaporation temperature affects the inlet pressure to the compressor. On the other hand, the change in the condensation temperature affects the compressor outlet pressure due to the condensation pressure. Gas leaks are more affected by the compressor outlet pressure, that is, by the condensation pressure. Therefore, in the measurements made, it is seen that the increase in gas leaks due to the increase in the condensation pressure is higher than the increase in the gas leaks due to the increase in the evaporation temperature.

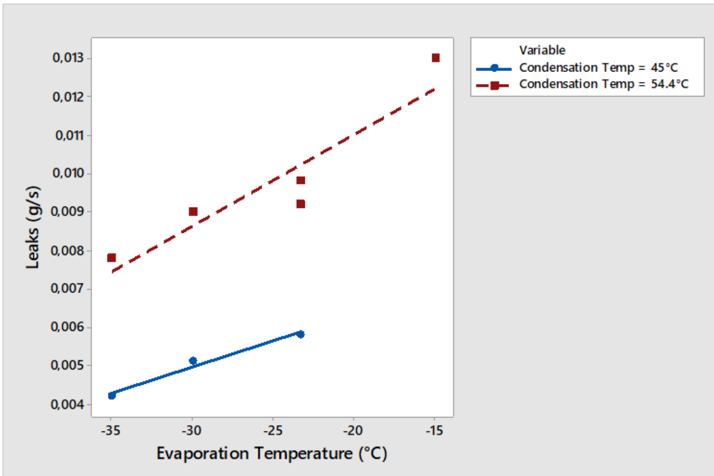



**Figure 4:** Variation of leaks according to evaporation temperature

Compressor 2 was first tested for gas leaks with Suniso 1GSA (10 cSt) oil and then Shell WE 7125 (7 cSt) oil. Gas leakage measurement results with oils of different viscosities are given in Table 1 below. When the results are examined, it is seen that the leaks increase by 35% in the measurements made with low viscosity Shell oil. The viscosity difference between the oils in question will be greatly reduced during the compression process. In this case, there is no significant change in piston dynamics depending on oil viscosity. On the other hand, it is thought that the increase in gas leaks will occur in the non-load-bearing parts of the piston. In addition, changing oil viscosity is also effective in lubricating the piston. In the case of using low viscosity Shell oil, the inability of the oil to reach the piston-cylinder bearing sufficiently may cause the piston-cylinder bearing to operate under more severe conditions. However, the number of measurements should be increased in order to interpret the change in piston leakages with the use of low-viscosity Shell oil.

**Table 1:** Gas leak tests with oils of different viscosities

| Compressor 1 | Leaks (g/s) | Mass Flow Rate (g/s) | Leaks Ratio (%) |
|---|---|---|---|
| Shell 7125 (7 cSt) | 0,00817 | 0,274 | 3 |
| | 0,00641 | 0,273 | 2,3 |
| | 0,0068 | 0,273 | 2,5 |
| | 0,00714 | 0,273 | 2,6 |
| | 0,00791 | 0,273 | 2,9 |
| Suniso 1GSA (10 cSt) | 0,00606 | 0,275 | 2,2 |
| | 0,00515 | 0,274 | 1,9 |
| | 0,00534 | 0,274 | 1,9 |
| | 0,00521 | 0,275 | 1,9 |
| | 0,00526 | 0,274 | 1,9 |
| | 0,00511 | 0,275 | 1,9 |
| | 0,00569 | 0,274 | 2,1 |
| | 0,00534 | 0,274 | 1,9 |

If the pressure ratios are approximately 1.7 in the flow in a narrowing section channel, the fluid velocity reaches the speed of sound at the narrowest part of the flow section and then creates a shock wave. In case the said pressure ratio is exceeded, further increase in the pressure ratio will not increase the flow through the channel. This situation is called “choked flow” in the literature. It has been determined that "choked flow" will occur in the calculations made by considering the pressure values during the compression work in the leakage region shown in the Figure 5 below, and that the flow value that will emerge in this case is in the same order as the leakage flow values obtained during the measurements.

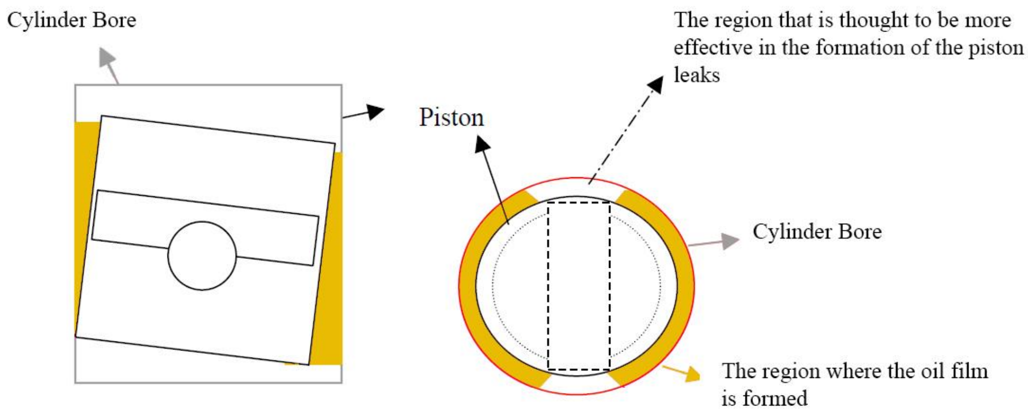



**Figure 5:** Representation of piston movement and oil film in the cylinder

It is the oil film that fills between the piston and the cylinder, which prevents the gas from escaping from the cylinder cavity into the compressor housing during the compression process. Since there is no load acting on the non-load-bearing region of the piston, there is not enough oil film to prevent gas escape. In this case, the escape of gas from this shadow is much easier.

Compressor 1 was first tested for gas leaks with 10 cSt oil and then with 7 and 5 cSt oils. The viscosity difference between the oils in question will be greatly reduced as the cylinder temperature rises during the compression process. In this case, it is expected that there will be no significant change in piston dynamics depending on oil viscosity. However, when the results were examined, it was seen that the viscosity and piston leakages changed with the oil viscosity and this change was not a linear relationship.

## 5. CONCLUSIONS

In this study, the second movement of the piston, which determines the mechanical losses, piston gas leaks, and noise of the crank-connecting mechanism, has been studied. By creating a dynamic model, the parameters of the crank-connecting rod mechanism were investigated. Then, piston gas leaks, which occur as a result of the second movement of the piston, were measured using the experimental setup in the literature.

In the dynamic model, connecting rod length, crank length, crank misalignment, pin position, and the position of the piston center of gravity were parametrically investigated. The most effective ones among these parameters are the distance of the pin from the piston center of gravity and the misalignment. In addition, eccentricity also creates a change on the second movement, but since the change of this parameter will also cause changes in other parameters such as capacity, compression ratio, speed, a more comprehensive examination will be required.

During the experimental studies, parameters such as oil viscosity, evaporation, and condensation temperatures were investigated. It has been observed that these parameters change the load on the piston and the piston dynamics. While oil viscosity did not create a linear change, piston leakage increased linearly depending on the increase in condensation and evaporation temperatures. While the density changes with the change of evaporation temperature, it influences the leaks, while the condensation temperatures change the outlet pressure and increase the loads on the piston, causing an increase in the leaks. Condensation temperatures are more effective on leaks than evaporation temperatures.

Studies show that piston secondary movement is affected by many parameters. It is an optimization problem to determine all these parameters to create minimum secondary movement in line with the desired capacity and efficiency.

## NOMENCLATURE

| | | |
|---|---|---|
| M | Mass | kg |
| n | Amount of gas | mol |
| P | pressure | kPa |
| R | Gas constant | kPa.L/mol. K |
| T | Temperature | K |
| V | volume | L |

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## ACKNOWLEDGEMENT

The present study was developed as part of a research reports of Arçelik A.Ş. The authors are also grateful for the support of Arçelik A.Ş.