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ABSTRACT

Lubrication plays a key role in assuring the reliability and efficiency of compressors and thus it has been the subject of many studies. However, the available investigations do not consider the presence of refrigerant and its effects on the oil supply. This paper reports the development of a novel experimental rig to measure the oil flow rate provided by the screw pump of the oil supply system of a hermetic variable capacity reciprocating compressor considering the presence of the refrigerant in contact with the lubricating oil. Experimental tests were carried out to investigate the effects brought about by the interactions between the AB ISO VG 5 oil and both air and R-600a as working fluids, considering that the oil and air are immiscible. The oil flow rate was measured under different operating conditions to assess the influence of the evaporating temperature and compressor speed on the oil flow rate. The results show that when compared to measurements with air, the presence of refrigerant gas decreases the oil flow rate, and this effect seems to be slightly potentialized by the increase of the evaporating pressure.

1. INTRODUCTION

Lubrication is one of the main aspects in determining compressor reliability. The adequate design of the oil supply system of reciprocating compressors is critical to assure that enough oil is provided to the compressor moving parts in a sensible time after startup. In fact, the need for efficient lubrication in compressors has increased in recent years, especially due to the increasing popularity of variable capacity compressors. In such devices, the threshold for decreasing the rotational speed is mainly determined by the oil pumping system ability to sustain sufficient oil supply to the compressor bearings. To fulfill these requirements, the complex oil-refrigerant mixture two-phase flow must be thoroughly studied, assessing the most important phenomena affecting the oil flow rate delivered by the pumping system.

In hermetic compressors, the oil and refrigerant gas coexist inside the compressor shell, and the interactions between these fluids, in the form of mass, heat and momentum exchange, may affect the performance of the oil supply system. For instance, when the gas is dissolved in the oil, the resulting mixture viscosity is lower in comparison with pure oil, which is detrimental to oil lubricity. Furthermore, when subjected to a pressure drop the dissolved refrigerant may come out of solution, giving rise to gas bubbles that may impair the oil flow rate delivered by the pumping system.

Because of its relevance, the oil supply systems of hermetic compressor has been the subject of many studies, employing different approaches. For instance, Kim *et al*. (2002) modeled the oil flow by means of an electrical circuit analogy applied to reciprocating compressors, while Kim and Lancey (2003) considered the same approach for rolling piston compressors. Other works were able to apply Computational Fluid Dynamics (CFD) to simulate numerically the oil supply system (Luckmann *et al*., 2000; Alves *et al*., 2011; Ozsipahi *et al*., 2014). These high-cost simulation models assumed similar hypotheses such as isothermal flow, no miscibility between the refrigerant gas and oil, and both fluids with constant thermophysical properties. Braga *et al*. (2022) presented the only numerical model that considered the solubility between the oil and refrigerant, estimating the mass transfer between the phases by means of a cavitation model. However, only the screw pump was considered in their model and the numerical results were not validated.

Some authors have experimentally investigated the oil supply system of hermetic compressors, reporting results regarding the oil flow rate delivered by the pump and the oil climbing time, which is defined as the time required for the first drop of oil to reach the top of the oil supply system. Tada *et al*. (2014) and Posch *et al*. (2018) developed experimental rigs to measure the oil flow delivered by the pumping system of a reciprocating compressor. The rig was basically composed of the compressor mechanical kit, an oil reservoir, and a graduated cylinder, with the measurements being carried out with the mechanical kit exposed to the atmospheric air. The tests considered different pump immersion depths and compressor rotation speeds. Kerpicci *et al.* (2013) and Zhu *et al.* (2020) focused on the visualization of the oil flow to measure the oil climbing time with a high-speed camera, but using the atmospheric air as the working fluid.

Despite the experimental investigations of the oil supply system of hermetic compressors, none of them have included the presence of refrigerant gas and the possible interactions that it entails when in contact with the lubricating oil. The experimental rigs available in the open literature provide measurements of oil flow rate and oil climbing time for oil and air, meaning that refrigerant absorption and outgassing from the mixture are neglected. However, these phenomena may give rise to relevant effects, such as outgassing and foam formation, that can significantly affect the oil supply for the compressor bearings.

This paper reports the development of a novel experimental rig to measure the oil flow rate provided by the oil supply system of a hermetic variable capacity reciprocating compressor considering the presence of the refrigerant gas in contact with the lubricating oil. The experiments are carried out with both air and isobutane as working fluids in order to investigate the effects brought about by the interactions between the oil and the refrigerant, since the interactions between oil and air are negligible. The oil flow rate is measured under different operating conditions to assess the influence of the evaporating temperature, compressor speed and oil temperature on the oil flow rate.

2. EXPERIMENTAL RIG

In a typical hermetic reciprocating compressor oil supply system, the oil is pumped from the oil sump at the bottom of the compressor shell to the bearings via the rotation of the main shaft. The main components of the pumping system are the partially submersed oil pump that admits the lubricant into the interior of the shaft and the helical groove (feeding channel), machined on the outer surface of the shaft, which transports oil to the bearings. After leaving the oil pumping system, the lubricant is delivered to the other compressor components such as the piston-cylinder clearance and the electric motor, returning then to the sump. The compressor model analyzed in this paper employs a screw pump, which works like a screw extruder and is comprised by a static central pin and a moving external barrel coupled to the shaft. The viscous force is the main driving force for pumping the oil upwards.

The experimental rig adopts a modified compressor assembly to allow the measurement of the oil flow rate delivered by the screw pump, as well as the visualization of the flow in key points of the compressor oil pumping system. These changes are illustrated in Figure 1. The upper wall of the compressor shell was replaced by an acrylic transparent lid to allow the visualization of the oil outlet at the top of the compressor mechanical kit. In addition to that, a rectangular viewing window was machined on the lateral wall of the compressor shell, allowing the optical access to the compressor interior environment. The shaft was shortened by 10 mm and then coupled with a transparent acrylic connector piece to visualize the oil flow inside the shaft. Additionally, a collecting tray made of polyurethane was introduced to collect the oil delivered by the pumping system, as shown in Figure 1, in which the pumped oil is represented by the yellow arrows. In the collecting tray, a cone-shaped hole was machined to conduct the oil for measurement, represented by the red arrows in Figure 1. Additionally, some pressure equalization holes were machined through the tray to avoid a pressure difference between the two environments inside the compressor housing.

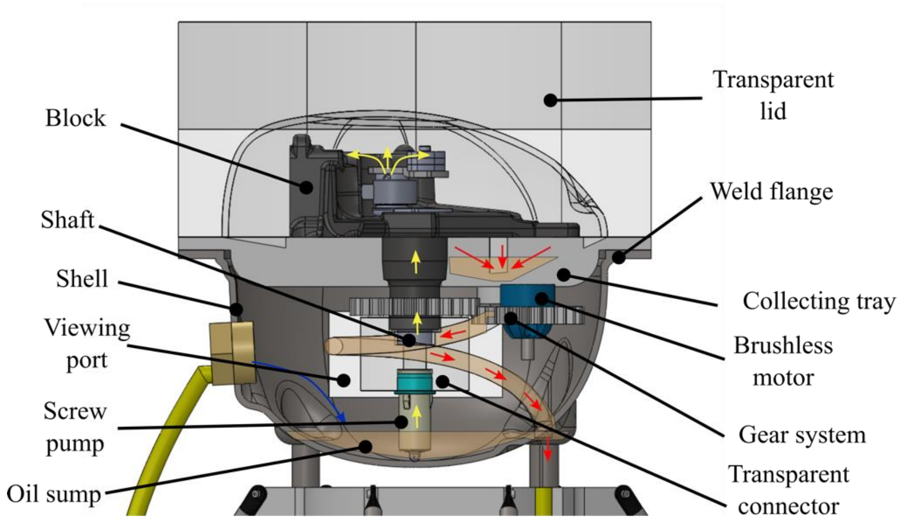

Figure 1: Test section components with oil flow direction.

Another important modification is required concerning the compressor electric motor, since it almost completely covers the drive shaft, preventing the visualization of the oil flow. To circumvent this difficulty, the electric motor was replaced by a brushless auxiliary motor of 770 rpm/V with a maximum speed of 13860 rpm, coupled to the shaft by means of a gear transmission system. The modifications made for visualization purposes are not explored in this paper and will be the subject of future works.

The modified compressor identified as the test section of the experimental rig, whose complete scheme is shown in Figure 2. The oil circulation and measuring system is comprised by the plastic hoses, an oil reservoir, an auxiliary pump, and a flowmeter. The direction of the oil circulation is represented arrows. Accordingly, the oil collected in the test section is directed to a reservoir (red arrows) and then fed back to the test section (blue arrows), driven by the auxiliary pump. Before returning to the test section, the oil passes through a flowmeter.

During the experiments, the oil temperature, pressure inside the compressor shell and rotation speed of the auxiliary motor are monitored and controlled. The oil temperature is controlled by two electric resistances, one located in the lower external part of the compressor shell (near the oil sump), and the second in the oil reservoir. To monitor the oil temperature, T-type thermocouples are installed inside the test section. The pressure inside the test section is measured by a Wika P30 pressure transducer, with a measurement range from 0 to 10 bar and a uncertainty of 0.05% of the full-scale value. The pressure inside the test section is controlled by a calorimeter, which in turn operates with an auxiliary compressor of the same model used in the test section. The brushless motor rotation speed is controlled by a magnet located in the shaft that transmits an electromagnetic pulse each time it passes through a receiver attached to the collecting tray. The time between pulses is communicated to a 30A ESC control driver, which is responsible to set the speed of the brushless motor.

The oil flow rate is measured by a flowmeter manufactured by Emerson, model Micro Motion 1700, and installed as shown in Figure 2. The equipment consists of a measuring unit, which applies the Coriolis principle, and an electronic unit, responsible for converting the electric signal into flow rate in L/min, with a maximum error of ±0.5% of the measured value.

To ensure that the oil flow rate in the pumping system is equal to the oil flow rate in the experimental rig provided by the auxiliary pump, the oil reservoir level is kept constant to ensure the system is in steady state. The auxiliary pump used on the bench is a Flexbimec gear pump, which produces a maximum flow rate of 1 L/min and a maximum pressure difference of 4 bar. The oil reservoir level control is carried out by means of a webcam that measures the number of pixels between the oil free surface and the top of the reservoir.

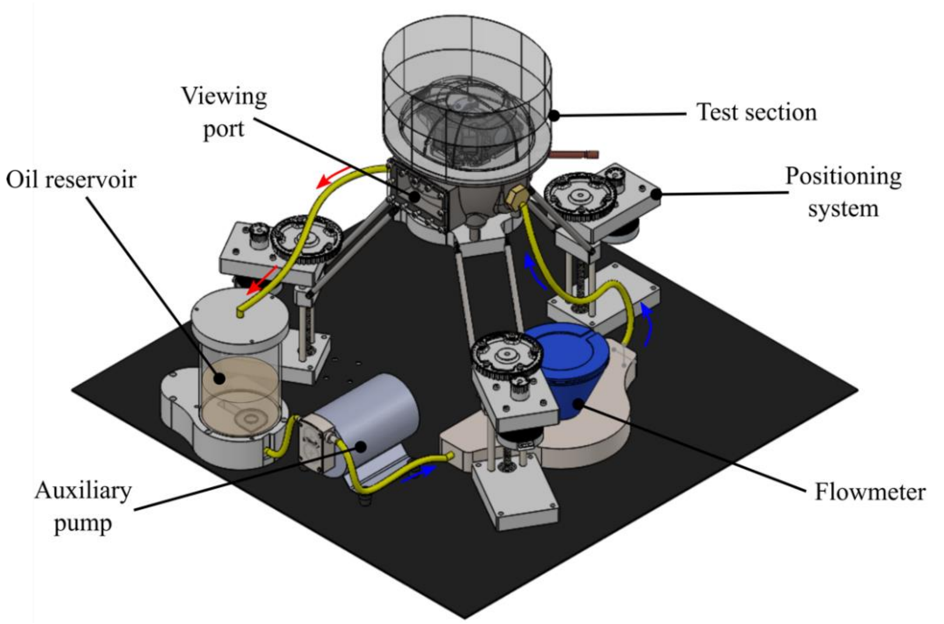

Figure 2: Experimental rig components with oil flow direction.

3. EXPERIMENTAL RESULTS

3.1 Experiments with air as the working fluid

Table 1 and Figure 3 show the measured oil flow rate (Q) in the test section operating with atmospheric air for different oil temperatures (T_{oil}) and compressor speeds (n). Typically, low speed values (1200 rpm, 1500 rpm and 3000 rpm) are chosen, as these values impose the most critical conditions for the oil pumping system. Six repetitions were carried out to establish each measurement, alternating the oil temperature between consecutive measurements.

The results in Figure 3 and Table 1 indicate a directly proportional and approximately linear dependence between the oil flow rate delivered by the pump and compressor speed. In addition, the increase in oil temperature causes a reduction in the oil flow. This effect was expected since the working principle of the screw pump is based on momentum transfer to the oil through viscous effects, and oil viscosity decreases with the increasing temperature.

Table 1: Oil flow rate for different oil temperature and compressor speed with air as working fluid.

n [rpm]	Q [mL/min]		
	T_{oil} = 50ºC	T_{oil} = 60ºC	T_{oil} = 70ºC
1200	53,4 ± 2,5	48,2 ± 1,9	45,3 ± 2,0
1500	69,1 ± 1,7	65,1 ± 1,7	61,6 ± 1,1
3000	135,7 ± 3,6	134,0 ± 4,4	130,1 ± 4,0

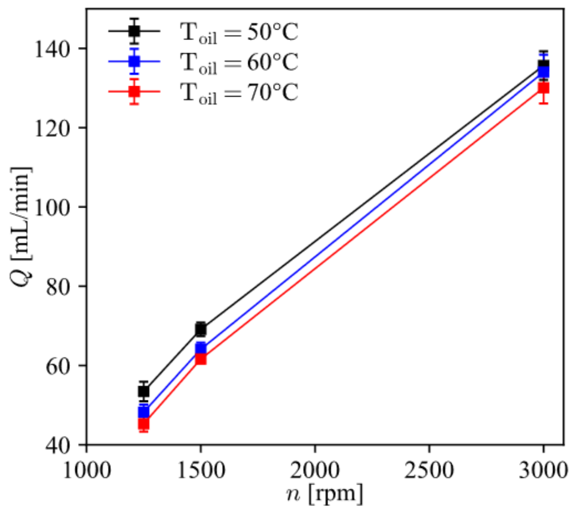

Figure 3: Oil flow rate for different oil temperature and compressor speed with air as working fluid.

3.1 Experiments with isobutane as the working fluid

After introducing the refrigerant into the experimental rig, a calorimeter was used to carry out experiments with different evaporating and condensing temperatures to assess the influence of these operating conditions on the measured oil flow rate. Typical operating conditions adopted for assessment of efficiency (-25/40ºC and -23.3/54.4ºC) and reliability (-10/ 90°C and -5/70°C) were considered.

Table 2 and Figure 4 shows the measured oil flow rate for each investigated operating condition for three rotational speeds. For comparison, the results obtained with air are also shown in Figure 4. All results were obtained with the oil temperature adjusted to 50°C. Each measurement represents the mean value of 6 repetitions, with the operating condition being changed between consecutive measurements. It should also be noted that the repetitions were performed on different days and by different operators.

The results in Figure 4 show that, regardless the operating condition, the oil flow rate when in presence of the refrigerant is smaller than that resulting when the test section operates with air, revealing that the solubility between the oil and the refrigerant is detrimental to the performance of the oil pumping system. One possible explanation for this reduction is the fact that outgassing takes place within the oil supply system, especially in the screw pump. The greater the amount of outgassing, the lower the average viscosity of the mixture, impairing the performance of the screw pump, whose efficiency strongly depends on the viscosity of the pumped fluid. In addition, increasing the amount of free gas can generate two-phase flow inside the pump and helical groove (feeding channel), which could also impair oil pumping due to flow intermittency brought about by the presence of gas in the pump.

Additionally, Figure 4 and Table 2 show that the increase of the evaporating temperature tends to decrease the oil flow rate. This effect may be related to the fact that as the pressure builds up inside the test section, the solubility between oil and isobutane increases, thus increasing the potential for outgassing in the helical pump. In fact, the only condition that diverges significantly from this trend is at -5ºC /70ºC with the rotation speed at 3000 rpm.

Due to this divergent result, additional tests were carried out for a more in-depth evaluation of the influence of evaporating temperature on the oil flow rate. This time, the condensing temperature was kept fixed at 40ºC. Figure 5 shows that, for both rotational speeds considered, the average oil flow rate slightly decreases with the increase in the evaporating temperature, reproducing the effect observed in Figure 4. However, it should be noted that this effect is not statistically conclusive, since the difference in the oil flow rate between the evaporation conditions is within the measurement uncertainty range of the points represented in Figure 5.

Table 2: Oil flow rate for different operating conditions and compressor speed with isobutane as working fluid.

n [rpm]	Q [mL/min]			
	-25ºC /40ºC	-23.3 ºC /54.4ºC	-10ºC /90ºC	-5 ºC /70ºC
1200	47,46 ± 0,91	46,8 ± 1,5	45,3 ± 2,6	45,2 ± 3,2
1500	61,6 ± 4,2	62,0 ± 3,2	56,9 ± 1,4	57,8 ± 2,1
3000	132,8 ± 6,2	131,4 ± 3,5	117,9 ± 4,0	127,5 ± 2,5

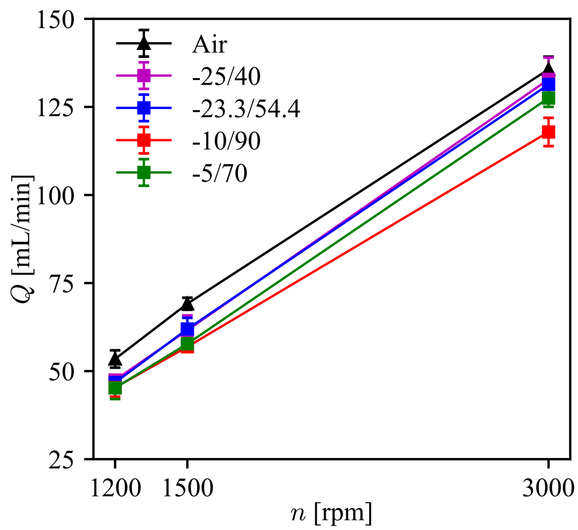

Figure 4: Oil flow rate for different compressor speeds for the test section with air as working fluid and oil flow rate for different operating conditions and compressor speed with isobutane as working fluid.

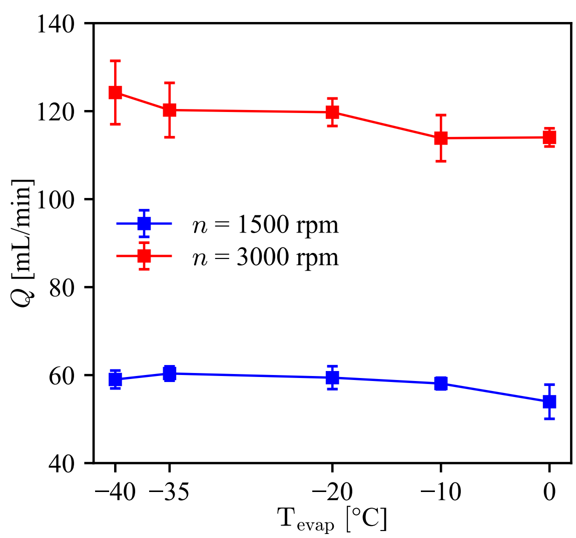

Figure 5: Oil flow rate measured as a function of the evaporating temperature, for rotation speeds of 1500 rpm and 3000 rpm, with a fixed condensing temperature of 40ºC.

4. CONCLUSIONS

The present paper reported the development of an innovative experimental rig to measure the oil flow rate delivered by the oil supply system of a hermetic reciprocating compressor in the presence of the refrigerant fluid. The results revealed that the oil flow rate is considerably decreased in the presence of refrigerant, for all the different operating conditions considered in the study, when compared with experiments carried out with atmospheric air as working fluid. The results indicates that the solubility between the oil and the refrigerant is a parameter of great importance, impairing the performance of the pumping system. A possible cause for this detrimental effect is that outgassing occurs in the oil supply system, especially in the screw pump, reducing the average viscosity of the mixture. Furthermore, the measurements showed that the reduction in the oil flow rate in the presence of refrigerant is more pronounced in higher evaporating temperatures, which may be related to the increase in refrigerant solubility in oil caused by the increase in the operating pressure.

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