Effects of Lubricating oil on the performance of a Four-Intersecting-Vane Rotary Expander

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Abstract. Lubrication plays an important aspect in machinery. It seals internal leakage gaps, reduces friction and cools components. The present investigation is intended to test the effects of lubrication to an expander’s performance. Polyester lubricating oil grades (Grade 48 and Grade 68) were tested at various operating conditions. The expander prototype is of the rotary mechanism with four vanes comprising two intersecting bars. An experimental test rig was designed and built. The pressure diagram as function of rotational angle was recorded by six pressure transducers arranged in the working chamber, based on which the features of the expansion process of the four-intersecting-vane rotary expander were analysed. To start with, the working fluid was compressed air, but the eventual intended application is in a vapour compression refrigeration system. The experimental study revealed that the rotary vane expander worked steadily at different operating conditions. The experimental results showed that the expander flow rate was generally lower with a thinner oil (POE grade 46), whereas the volumetric efficiency increased from 23.2% to 27.2%. The adiabatic efficiency decreased from 22.5% to 18.1% with a higher viscosity lubricating oil (POE grade 68).

Keywords: Rotary vane expander; Expansion; Oil, Friction, Efficiency

1. Introduction
In recent years reducing energy usage has become a key aspect in all applications and the trend will continue in the coming years, and this includes refrigeration systems [1]. This problem can be effectively solved by using more energy efficient systems and cleaner energy generation methods. One such method is to improve the energy efficiency of a refrigeration system by recovering the waste energy during the expansion process. Conventional throttle valves do not have the ability to produce work, and thus the idea to use an expander as a substitute has been considered. An expander can be categorised into volumetric machines, e.g. rotary, reciprocating piston, screw and scroll expanders, and dynamic machines, e.g. radial and axial turbines [2]. There have been a number of experimental investigations of different expander mechanisms, including screw [3], scroll [4], reciprocating [5, 6], swing piston [7], rolling piston [8], rotary vane [9], cross vane...
and revolving vane machines, all aimed at improving the expander efficiency. Among the potential concepts, the rotary mechanism is generally more compact, does not require suction/discharge valves, operates with less noise, has less vibration, but tends to suffer from high leakage and/or significant friction. Therefore, lubrication is crucial to achieve high performance and good reliability in a rotary machine.

Various studies are available in the literature on lubrication of compressors. Okoma and Onoda discussed various means of oil supply for pressure differential system, gas carry system, vane pump system and rotary compressors. In a pressure differential system, an oil cap encloses the crankshaft suction and an oil delivery pipe from the oil cap is hung over down to an oil sump. On the other end of the oil gallery inside the crankshaft it is connected to a suction pressure, and so oil from an oil sump under the pressure of discharge is drawn into the oil gallery inside the crankshaft through the oil delivery pipe. However, this system is sensitive to load changes, so the oil supply rate becomes unstable. De Paepe et al. studied the influence of oil atomization on the working process of a screw air compressor. They concluded that oil atomization considerably increased heat transfer. However, it did not significantly affect the compressor performance. It was observed that lowering the oil temperature improved the performance of the expander, while changing oil flow rate only gave small gains. Valenti et al. conducted an experimental study on the thermal effect of lubricating oil in a medium size rotary vane air compressor. The results showed that oil droplets with diameters on the order of 100 μm led to a large reduction in both temperature rise and compression work. It is to note that the lubrication design of an expander has its unique challenges. This is due to the low discharge pressure and temperature working conditions. For example, in a refrigeration system with R22 as the refrigerant, the expander’s discharge pressure and temperature are 3.5 to 4.1 bar (abs) and -5 to 1°C, respectively. This is different from a compressor, in which the discharge pressure and temperature are much higher. Therefore, the selection of the lubricant oil for the expander is important, which is based on the operating conditions such as pressure and temperature. It is to note that the oil viscosity increases with decrease in temperature. Excessive increase or decrease in the viscosity can result in permanent modifications of characteristics of oil, which can damage the lubricating system of the prototype. The survey above shows that compressor lubrication is relatively well-studied. However, to the best of the authors’ knowledge, no study has been conducted on expander lubrication. This study aims at filling the knowledge gap. Most expander mechanisms (scroll, screw, piston etc.) need lubrication but the study focuses on the influences of lubricating oil on a rotary expander with four vanes comprising two intersecting bars, which will be referred to as the four-intersecting-vane expander from here on. This prototype has been tested before in a refrigeration system. The mechanism is different from conventional rotary vane machines because the four vanes are formed by two intersecting long bars. Furthermore, while the rotor is circular like those in conventional rotary vane machines, the stator’s inner wall is not circular, as shown in Figure 1a. This configuration results in reduced frictional loss but may amplify the internal leakages. More detailed descriptions of the mechanism can be found in the survey. The lubricating oil is supplied to the inlet pipe of the expander and its influence on the performance of expander specifically volumetric efficiency, adiabatic efficiency and output torque are studied. The study provides researchers and engineers with useful information to choose the appropriate lubricating oil to optimize the expander performance.

2. The Expander Prototype

Figures 1 (a) & (b) show the expander prototype. It mainly consists of a circular rotor with four vanes. The vanes are made of two intersecting bars. The inner wall of the stator is not circular but is designed to minimize the gaps between the vane tips and the wall. The main geometric parameters of the expander are tabulated in Table 1. As shown in Figure 1, the four vanes and the line contact between the rotor and the stator separate the space into several working chambers. A suction port is located near the line contact between the rotor and the stator. High pressure fluid enters the expander through the suction port to the working chamber Volume 1. The volume of each working chamber changes as the rotor turns. The suction process ends, and the expansion process starts when the vane is at 90°. As the chamber volume increases
further, the fluid pressure decreases and the maximum chamber volume is reached when the vane is at 225°, the discharge process begins and continues up to 340°. The discharge port does not extend to 360° in order to decrease leakage flow from the suction port to the discharge. So, from 340° to 360°, the fluid might undergo a brief compression process. The developed expander prototype volume ratio is 8.3. If the volume ratio is too large for a given set of working conditions, the working fluid is trapped longer than necessary leading to a pressure drop below the outlet pressure (over-expansion). Conversely, a low volume ratio results in under-expansion. Both over- and under-expansion affect the expander’s efficiency.

| Item                      | Value   |
|---------------------------|---------|
| Radius of the rotor       | 27.7 mm |
| Radius of the stator      | 40.7 mm |
| Length of the cylinder    | 38.6 mm |
| Thickness of vanes        | 8 mm    |
| Angle range of inlet port | 10° – 90° |
| Angle range of outlet port| 225° – 340° |

3. **Experimental Setup**

Figure 2 shows the experimental test rig. The experiments were carried out to test the main characteristics of the expander, specifically the volumetric efficiency, adiabatic efficiency the instantaneous output torque at different pressures and rotational speeds. Compressed air was used as the working fluid. The maximum pressure available in the compressed air tank in the lab was 8 bar(g), hence, the expander was tested up to a suction pressure of 5 bar(g), to ensure a stable air flow. The expander suction pressure was controlled using a pressure regulator and the working fluid was discharged to the atmosphere. A dynamometer was used to consume power produced by the expander. The output torque was calculated by multiplying the measured force at the load cell with the length of the lever arm (i.e. 200 cm). The power output from the expander was calculated from the torque and rotational speed of the expander. The flow meter used in this experiment was the GSVT 54 float Fischer and Porter rotameter, and the speed was measured using an optical sensor. The accuracies of the measuring devices are tabulated in Table 2. The operating conditions of the experiments are listed in Table 3. All the measurement instruments were connected to a CR1000 measurement data logger designed and manufactured by Campbell Scientific. The specifications of the lubricating oils are given in Table 4.
Figure 2. The experimental rig

Table 2. Accuracy of the measuring devices

| Devices                        | Accuracies | Unit   |
|--------------------------------|------------|--------|
| Thermocouple (T-type)          | ± 0.5      | °C     |
| Load cell (HTC Sensor TAL220)  | ± 0.01     | g      |
| Frequency generator            | 0.02% ± 0.01% | Full scale (FS) |
| Pressure transducer            | ± 0.3%     | Full scale (FS) |
| Speed sensor (Optek/TT electronics OPB705WZ) | ± 5 | Revolutions per minute |

Table 3. Main testing parameters

| Item                | Value           |
|---------------------|-----------------|
| Inlet pressure      | 4-5 bar(g)      |
| Discharge pressure  | 0 bar(g)        |
| Operating speed     | 295-1350 rpm    |
Table 4. Typical physical characteristics of the lubricating oil [23, 24]

| ISO Fluid type | Kinematic viscosity (40°C) | Kinematic viscosity (100°C) | Density (kg/l) | Pour point (°C) |
|----------------|---------------------------|----------------------------|----------------|----------------|
| POE Grade 46   | 46.8                      | 7.3                        | 0.978^         | -48            |
| POE Grade 68   | 72.3                      | 9.8                        | 0.978^         | -39            |

^@20°C

3.1. Definition of parameters

The main parameters to evaluate the expander performance were power output, isentropic efficiency and volumetric efficiency.

Volumetric efficiency is the ratio of the theoretical mass flow rate to the actual mass flow rate, which is defined according to equation 1.

$$\eta_v = \frac{m_{th}}{m_{act}}$$  \hspace{1cm} (1)

The theoretical mass flow rate is calculated using equation 2

$$m_{th} = \rho_{suction} \times V_{suction} \times N \times n$$  \hspace{1cm} (2)

where $m_{th}$ is the theoretical flow rate if there is no leakage, and $m_{act}$ is the net volume flow rate into the expander under the actual experimental conditions, $V_{suction}$ is the suction volume, which is the volume of the chamber at 90°, $N$ is the rotational speed of the expander, $n$ is the number of vanes and $\rho_{suction}$ is the suction fluid density.

The adiabatic efficiency of the expander was calculated using equation 3:

$$\eta_{ad} = \frac{h_{in} - h_{out}}{h_{in} - h_{out,is}}$$  \hspace{1cm} (3)

where $h_{in}$ is the inlet enthalpy, $h_{out}$ is the outlet enthalpy and $h_{out,is}$ is the outlet enthalpy for isentropic expansion to the same outlet pressure. Thermophysical properties of the working fluid were obtained using the CoolProp database [18].

Every experimental study includes uncertainties coming from the inaccuracies in the measurement devices. To compute the uncertainty in this study the following equation 4 was used.

$$\delta R = \sqrt{\sum_i \left(\frac{\partial R}{\partial X_i} \delta X_i\right)^2}$$  \hspace{1cm} (4)

where $\partial R$ is the error propagated on generic indirect measured parameters and $\delta X_i$ is the error of the directly measured parameter relevant to $\partial R$. 
4. Results and discussions

To study the internal working processes of the four-intersecting-vane rotary expander, six pressure sensors were arranged in the prototype to measure the chamber pressures at different positions as tabulated in Table 5. Figure 3 shows the variation of pressure in the working chambers of the prototype corresponding to the rotational angle when the expander rotational speed was 1120 rpm. The ideal pressure-angle diagram if the pressure was constant during suction and discharge and the expansion process was isentropic is also shown and called the “ideal” case. It can be observed that the actual process of the expander differed from the ideal case. In the experiments, the pressure dropped during suction showing that the suction flow was not able to cope with the increase in volume. This might indicate that the suction port was too small. In the expansion stage, the pressure dropped less rapidly than that in the ideal/isentropic case. This was mainly because of internal leakages and heat transfer into the expanding fluid. At the end of the discharge process, the pressure rose slightly because the discharge groove only extended out to 340°. Therefore from 340° to 360°, the fluid was slightly compressed.

| Sensor | Position (°)     |
|--------|------------------|
| P1     | 0°-30°           |
| P2     | 30°-120°         |
| P3     | 120°-225°        |
| P4     | 225°-330°        |
| P5     | 330°-360°        |
| P6     | Suction port (10°) |
| T1     | Suction port     |
| T2     | Discharge port   |

To study the performance of the expander and how lubricant affects it, experiments were conducted using the test rig and measurement devices described in section 3. The experimental results are plotted in Figures 4 to 6. Figure 4 shows the volumetric efficiency of the expander at various suction pressures, operational speeds and oil grades. In this study, the average uncertainty of the volumetric efficiency data is ±0.16%. The larger inlet pressure increased pressure difference across the internal leakage paths. This results in the increased leakage flow rates and decrease in the volumetric efficiency [21]. As shown in Figure 4, the volumetric efficiency increases with an increase in the rotational speed of the expander. The leakage rate contribution to the total expander flow rate reduced with increasing operational speed, so a higher rotational speed led to a higher volumetric efficiency [19]. A more viscous oil decreased the leakage flow rate and improved the volumetric efficiency. The average oil flow rate for POE grades 48 and 68 lubricants at suction pressures of 4 bar(g) and 5 bar(g) were 0.28 g/s and 0.32 g/s, and 0.34 g/s and 0.37 g/s, respectively. The highest volumetric efficiency measured was 27.5% for POE 68 oil at a suction pressure of 4 bar (g). For all the cases, it is evident that a higher volumetric efficiency was achieved at higher operational speeds. This was because the leakage flow rate was mostly affected by the pressure difference and gap size, but was relatively independent of the rotational speed. However, the increase in operational speed increased the expander flow rate, hence, the leakage became less significant as the speed increased, which was also observed by other researchers [8]. The relatively low volumetric efficiency values suggest the severity of internal leakages in the expander and the need for a proper lubrication system. This could be accomplished using an active lubrication system, such as with an oil pump. The lubricating oil pump is used to circulate an amount of oil from a reservoir to the oil injector.
The variation of the expander’s adiabatic efficiency with suction pressure, rotational speed and different oil grades is presented in Figures 5 (a) & (b). In this study, the average uncertainty of the adiabatic efficiency data is ±1%. It can be observed that the increase in rotational speed improved the adiabatic efficiency. From Figures 5 (a) & (b), the highest adiabatic efficiency of 28% was obtained with POE 48 oil at an inlet pressure of 4 bar(g). A more viscous oil resulted in more frictional losses. It is interesting to note that expanders operate at lower temperatures as compared to compressors. Therefore, the actual operating viscosity of the oil is higher compared to that of compressors. Since the expander was tested under an open system arrangement, the exit pressure of the fluid was equalised with the atmospheric pressure. Another factor that
affected the adiabatic efficiency was the fact that the expander’s expansion process was fixed from 90° to 225°, which corresponds to an isentropic expansion from 19 bar to 1 bar. This meant over or under-expansion processes may happen as the inlet pressure was varied. Adiabatic efficiency was also influenced by the internal leakages in the expander, which were quite significant in this study as discussed above. It is noted that the maximum adiabatic efficiency measured in this study is relatively higher compared to all the expanders operating with compressed air as the working fluid in the open literature [16]. The results also show that oil viscosity has contradicting effects to the expander efficiency. A more viscous oil increased the volumetric efficiency but decreased the adiabatic efficiency.

Figure 6 shows the variation of average output torque with suction pressure, operational speed and oil grades. The average output torque increased with the increase in suction pressure. This is because a higher inlet pressure led to more force applied on the vane to rotate the expander. However, the increase in rotational speed decreased the average output torque. This was due to the high frictional losses at higher rotational speeds as a result of the higher rubbing velocity and the larger frictional forces [20]. A higher oil POE grade 68 lubricant reduced the output torque because more friction losses were generated. Overall, leakage and frictional losses were found to be the main factors for the low performance of the expander and hence further improvement in the prototype is needed to improve its performance.

5. Conclusions
An experimental study has been conducted to determine the performance of a rotary vane expander with four intersecting vanes using compressed air/oil mixture as the working fluid. The study was carried out at suction pressures of 4-5 bar (g), a discharge pressure of 0 bar (g) and rotational speeds of 295-1350 rpm. The expansion process of the expander was analysed using six pressure sensors. The effects of the lubricating oil and working conditions on the expander efficiencies were analysed and the following conclusion were drawn:

1) The volumetric efficiency increases with increasing the operational speed of the expander but decreases with increase in suction pressure. Higher viscosity oil increased the volumetric efficiency. The highest volumetric efficiency was obtained for lubricating POE oil with grade 68. When the rotational speed was 295 rpm, the volumetric efficiency of the expander was around 8.4%. Further increasing the rotational speed improved the volumetric efficiency of the expander to 27.5% at the rotational speed of 1350 rpm.

2) The expander adiabatic efficiency of the expander depended on the viscosity of oil, rotational speed and inlet pressures. The highest adiabatic efficiency was 22.4% for the POE 48 at an inlet pressure of 4 bar (g). The adiabatic efficiencies varied from 7.8% to 22.4% under the tested conditions.
3) The output torque produced by the four-intersecting-vane expander prototype is sensitivity to the viscosity of the lubricant oil used. The expander torque increased with operational speed and increased with increasing suction pressure but decreased with more viscous oils. When POE oil grade 48 is used, the highest average torque readings is 1.1 Nm for 295 rpm at a suction pressure of 4 bar (g).

4) A more viscous oil reduced internal leakage and increased volumetric efficiency, but it increased frictional losses and so decreased the adiabatic efficiency and output torque.

In summary, the current study provides an insight into the performance of the prototype four intersecting vane expander. Leakage and frictional losses should be reduced to further improve the performance of the expander. This can be achieved by the proper selection of lubricating oil. Moreover, it was shown that the effect of lubricant could be significant and needs further investigations. The expander prototype will be installed into a refrigeration system and experimented with in the future.

Nomenclatures

| Symbol | Description       | Unit |
|--------|-------------------|------|
| $h$    | Enthalpy          | (kJ/kg) |
| $m$    | Mass flow rate    | (kg/s) |
| $p$    | Pressure          | (bar) |
| $P$    | Power             | (W)   |
| $V$    | Volume flow rate  | (m$^3$/s) |
| $T$    | Temperature       | (K)   |

Greek

| Symbol | Description       | Unit |
|--------|-------------------|------|
| $\omega$ | Operation speed           | (rad/s) |
| $\rho$      | Density            | (kg/m$^3$) |
| $\eta$     | Efficiency         | -    |
| $\theta$  | Angle              | (deg) |
| $\partial F_i / \partial x_i$ | Derivative of the calculated data with respect to the independent measured variable | - |
| $\Delta x_i$ | Uncertainty of the directly measured parameter relevant to the calculated data | - |
| $\Delta y$ | Uncertainty of the calculated data | - |

Subscript

| Symbol | Description       |
|--------|-------------------|
| in     | Inlet             |
| is     | Isentropic        |
| out    | Outlet            |
| th     | Theoretical       |

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