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A Review on Expanders and their Performance in Vapour Compression Refrigeration Systems

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ABSTRACT

This paper reviews progress reported in the open literature of the use of expanders to recover expansion power to improve the energy efficiency of vapour compression refrigeration systems. Pioneering works in the field are first discussed, and then a variety of expander mechanisms, including reciprocating piston, rolling piston, rotary vane, scroll, screw and turbine are reviewed together with their reported performance. Most of the reported works have been for transcritical CO_2 refrigeration systems, which have reported improvements in the coefficient of performance (COP) of up to 30%. In a non-CO_2 system, the maximum reported increase in the COP was 10%. The maximum reported expander efficiency (i.e. the ratio of measured power to the power available if the expansion process is isentropic) was 83%, obtained with a scroll expander in a CO_2 refrigeration cycle. Other expander issues including heat transfer, expansion process, internal leakage, irreversibility, control strategy, installation issues and economic analysis are also reviewed. It is noted that there are very limited studies in these areas. Finally, challenges and recommendations for future studies in the area of refrigeration expander technology were presented.

Keywords: refrigeration system; expander; energy efficiency; sustainability

1. INTRODUCTION

Refrigeration, air conditioning, and heat pumps have contributed significantly to the growth of various industrial sectors, including food, building, medicine, and manufacturing. Furthermore, refrigeration has improved the general well-being of humanity by providing better food and pharmaceutical storage systems, while air conditioning has allowed regions with high air temperature and humidity to be more habitable and, consequently, undergo economic development. The International Institute of Refrigeration (IIR) (Coulomb et al., 2015) have estimated the total number of refrigeration, air conditioning, and heat pump systems in operation worldwide as 3 billion units and the global annual sales of related equipment at 300 billion USD. The report also estimated that 17% of the global electricity consumptions is used for refrigeration and air conditioning. Unfortunately, due to the world's dependence on fossil fuels to produce energy, this means that the sector contributes significantly to the total greenhouse gas emissions into the atmosphere.

In addition to their energy consumption, refrigeration systems also impact the environment when their working fluids, called refrigerants, are released or escape into the atmosphere. Until the 1980s, halogenated chlorofluorocarbons (CFCs) and hydrochlorofluorocarbons (HCFCs) were widely used. When scientists discovered their ozone depleting properties, the world agreed to ban or phase out these Ozone Depleting Substances (ODS) through the Montreal Protocol (Breidenich et al., 1998). Another damaging impact of these refrigerants is their Global Warming Potential (GWP). In recent years these two factors have driven the re-introduction of natural refrigerants, such as CO_2, ammonia, water and hydrocarbons (such as butane, propane, and cyclopentane), in the recent years (Bolaji and Huan, 2013; Riffat et al., 1997; Lorentzen, 1995). Natural refrigerants generally do not affect the stratospheric ozone layer and have low GWPs. Among others, carbon dioxide (CO_2) has arguably received the most attention as it has good thermophysical properties, is non-corrosive, non-flammable and non-toxic (Fukuta et al., 2010; Bansal, 2012).

There are a number of thermodynamic cycles that can be adopted for refrigeration. However, by far, the most popular is the vapor compression refrigeration (VCR) cycle. It includes four main components, a compressor, a condenser, an expansion device, and an evaporator. The refrigerant is compressed in the compressor to a higher temperature and pressure. It then rejects its thermal energy in the condenser to the ambient (in refrigerators and air conditioners) or the heated space (in heat pumps). The condensed refrigerant then undergoes expansion in an expansion device to a lower temperature and pressure. It then enters the evaporator to absorb heat from the cooled space (in refrigerators and air conditioners) or from a thermal reservoir (in heat pumps). The refrigerant then flows back into the compressor, and the cycle is continued.

The expansion process is traditionally carried out by throttling in a capillary tube or an expansion valve. The expansion process is traditionally carried out by throttling in a capillary tube or an expansion valve. Previous studies have shown that this process is one of the major sources of irreversibility in a vapor compression refrigeration system (Ahamed et al., 2011). Therefore, it is desirable to find solutions to reduce this loss. It is important to note that the performance of a VCR system is typically quantified by its Coefficient of Performance (COP), which is defined as the ratio between the heating or cooling capacity (depending on the application) and the power consumption. Compressor power typically dominates the overall power consumption. Thus, the COP can be expressed as Equation (1).

$$COP_{heating\ or\ cooling} = \frac{Q_{heating\ or\ cooling}}{W_{compressor}} \tag{1}$$

The traditional expansion device is a capillary tube. More sophisticated alternatives include a thermostatic expansion valve (TEV) and an electronic expansion valve (EEV), which allow for a more precise expansion control system that will give an improved performance (Cao et al., 2016). However, even with an EEV, the expansion process is still highly irreversible, and the available expansion work is still lost. Hence, there have been proposals to modify the expansion process to one that is isentropic, and to simultaneously recover the expansion work. This can be achieved by using two alternative technologies, an ejector or an expander. An ejector utilizes the expansion energy to lift the refrigerant pressure at the compressor suction line to reduce the compressor's expansion ratio, which will result in an increased COP. Recent reviews on ejector technologies are available in the literature (Elbel and Lawrence, 2016; Besagni, et al., 2016). In general, ejectors are simpler to build than expanders. However, ejectors require a more extensive modification to the cycle. In contrast, expanders are easier to install and are especially suitable for retrofitting purposes. Moreover, expanders are generally more efficient than ejectors (Elbel and Lawrence, 2016).

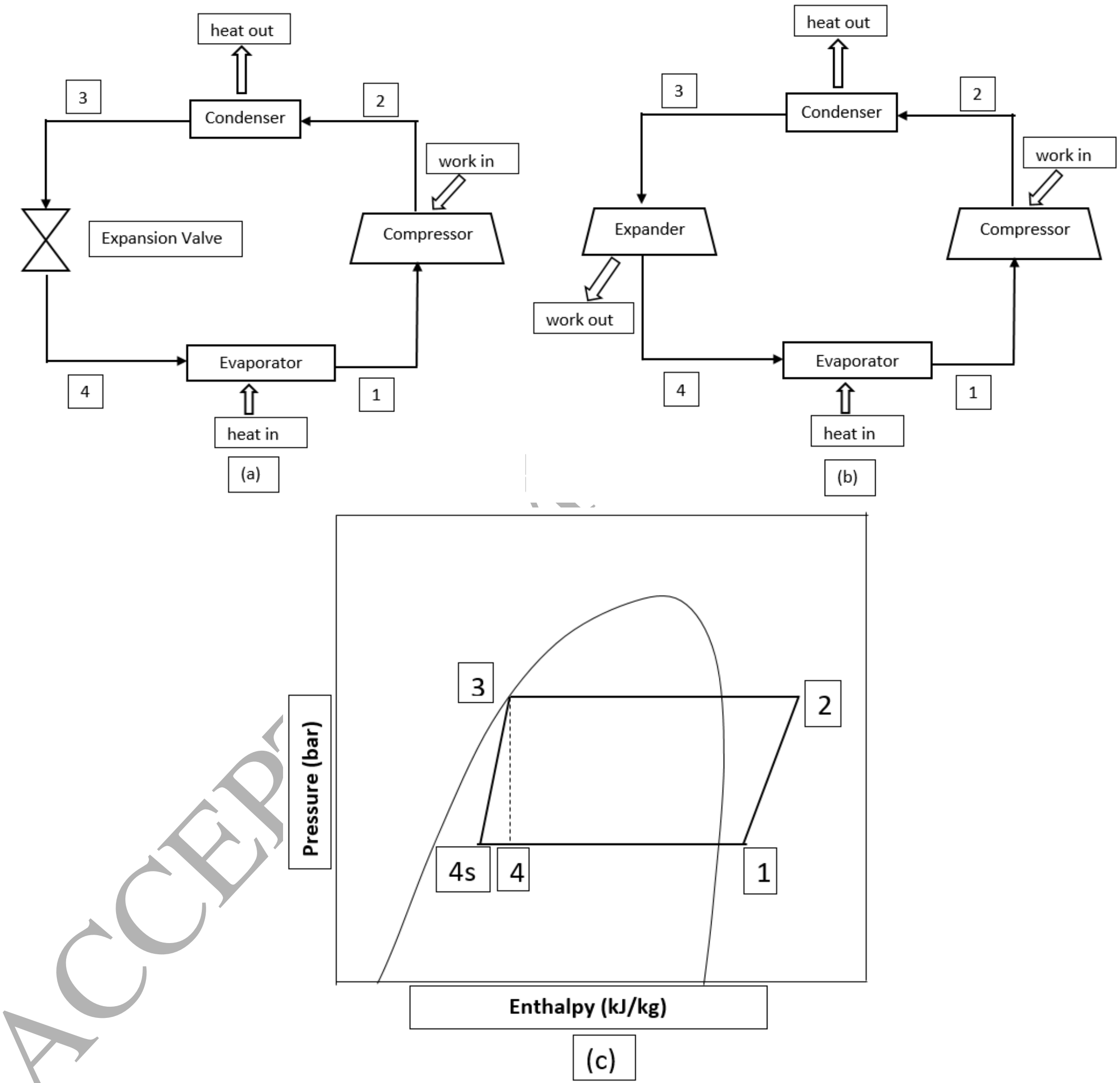

Figure 1. Schematics and idealized pressure-enthalpy diagrams of a vapor compression refrigeration cycle with an expansion valve and an expander

Schematics and pressure-enthalpy diagrams of vapor compression refrigeration systems with an expansion valve and with an expander are shown in Figure 1. The ideal expansion process in an expander is isentropic, unlike that in a throttling device which is an isenthaplic process. This results in more cooling capacity without changing the heating capacity. At the same time, the available expansion power is captured. Therefore, the COP of a VCR system with an expander can be expressed as in Equations (2) and (3).

$$COP_{cooling} = \frac{Q_{cooling} + \Delta Q_{expander}}{W_{compressor} - W_{expander}} \tag{2}$$

$$COP_{heating} = \frac{Q_{heating}}{W_{compressor} - W_{expander}} \tag{3}$$

An expander can be thought of as a compressor operating in reverse. It takes in high-pressure fluid and expands the fluid gradually to a lower pressure to produce energy. The energy is typically in the form of torque, which can be used to reduce the load of the compressor motor or to generate electricity through a generator. The concept of using one in a refrigeration system has been around since the early 1900s and they are sometimes mentioned in basic thermodynamic textbooks, but their use has usually not been considered to make economic sence (Horst, 1911; Plank, 1912). They were considered more seriously with the introduction of the CO_2 transcritical refrigeration system, which suffers a large throttling loss due to the extremely large operating pressure difference (Lorentzen and Pettersen, 1993; Lorentzen, 1994). Since then, the concept has gained more attention. This is visible in recent refrigeration and compressor conferences, where there has been an increasing number of presentations on expanders. However, there is currently only one review paper on expanders for refrigeration systems in the literature, which was published in 2013, which focused on CO_2 transcritical refrigeration systems (Zhang et al., 2013). Since then research has been published including the application of expanders to non-CO_2 refrigeration systems. Therefore, a review the status of expander developments in refrigeration systems is overdue. This paper will provide such a review, with an emphasis on the reported performances of expanders and their systems. Other aspects that affect an expander's performance including heat transfer, expansion process, irrerversibility, lubrication, control and installation strategies, are also reviewed. Practical and economic aspects, challenges, and recommendations for future development of the technology are discussed. Emphasis will be given to journal papers, but conference contributions and other methods of publications will be mentioned whenever relevant.

2. SIMULATION WORKS OF VCR SYSTEMS WITH EXPANDERS

Simulation is heavily relied on for the understanding of expanders and their systems. In this section, simulation studies (without experiments) on the performance of vapor compression refrigeration (VCR) systems with expanders are reviewed. They provide a general idea on how an expander affects VCR systems. Most of the reported works have been focused on system configuration and parametric studies.

Robinson and Groll (1998) were among the first to study how an expander affects the performance of VCR cycles, and its interaction with an internal heat exchanger (IHX), using a simulation model for with and without an expander. The systems studied included a conventional

R-22 and a transcritical CO_2 cycle. They found that an expander with an isentropic efficiency of 60% could reduce the contribution of the expansion device to the overall irreversibility by 35%. The heat sink temperature for all cycles, the condensing temperature of R-22 and the outlet temperature of the CO_2 gas cooler were assumed constant at 35°C, 50°C and 40°C, respectively. The evaporating temperatures were varied from -40°C to 5°C to determine the optimum heat rejection pressure and COP for the cycles. They found that without an IHX, the expander improved the cycle's COP by 25% compared to that of a transcritical CO_2 cycle with a throttling valve. Interestingly, the COP of the CO_2 cycle decreased by up to 8% when an IHX was used in conjunction with the expander, while it increased the COP by up to 7% in a traditional CO_2 cycle. When applied to an R-22 cycle, an expander with a 60% isentropic efficiency could improve the COP by more than 10% as compared to that of an R-22 cycle with a throttling valve. Salim (2009) also compared combinations of expansion valves, expanders and IHXs by simulating three different automotive CO_2 air-conditioning system configurations: 1) a baseline cycle with an internal heat exchanger (IHX), 2) an expander cycle, which was simply a baseline cycle without an IHX and with a turbine replacing the expansion valve, and 3) an expander cycle with an IHX. In addition, an R-134a cycle with an expander was also simulated. It was found from the study that the CO_2 expander cycle without an IHX offered the highest COP with up to 45% improvement as compared to the baseline cycle, in agreement with the earlier study of Robinson and Groll (1998). However, this came at the cost of a lower cooling capacity. On the other hand, the CO_2 cycle with an IHX and expander gave the highest capacity. They also found that up to 57% of the compressor work could be recovered in the CO_2 expander cycle without an IHX. In the CO_2 cycle with an expander and an IHX, only 30% recovery was obtained. When an expander was used in an R-134a system, around 18% of the compressor work could be recovered. Another simulation study on the effects of expanders and IHXs was reported by Ghazizade-Ahsaee and Ameri (2018) who developed a simulation model to analyze the energy and exergy of various transcritical CO_2 geothermal heat pump cycles, including: 1) an expansion valve, 2) an expander, 3) a valve and an internal heat exchanger (IHX), and 4) with an expander and an IHX. It was observed that without the IHX, the expander cycle had a higher COP and exergy efficiency as compared to that of the expansion valve. An IHX improved the performance of the cycle with an expansion valve, but had no significant effect to the cycle with an expander. Again, this is consistent with the results from previous studies (Robinson and Groll, 1998; Salim, 2009).

A parametric study of a transcritical CO_2 system with a throttling valve or an expander was performed by Yang et al. (2005, 2014) to determine the optimal heat rejection pressure. It was found that the expander cycle not only had a better COP, but it had a lower operating pressure as compared to the throttling valve cycle. The gas cooler outlet temperature, evaporation temperature, compressor efficiency, and expander efficiency were related to the optimal heat rejection pressure. The study found that the gas cooler outlet temperature had the most significant impact. Yang et al. (2006a) simulated a CO_2 water heat pump system with an expander. The study found that inlet temperature and flow rate of cooling water influence the performance of the system and optimal rejection pressure.

Goncalves & Parise (2008) modelled an expander in an R-134a refrigeration system. The simulation assumed condensing temperatures of 40-50°C and an evaporating temperature of 0°C. The expander isentropic efficiency was varied from 0.2 to 0.8. They showed that the expander was able to improve the COP by up to 12%, depending on the expander's efficiency, expansion ratio and condensing temperature.

The characteristics of multi-stage VCR systems with expanders have also been studied. Yang et al. (2007) investigated three different variations of transcritical CO_2 two-stage compression cycles with an expander and: 1) two-stage compression at optimal intermediate pressure (TCOP) cycle, 2) two-stage compression with expander driving high-pressure stage (TCDH) cycle, and 3) two-stage compression with expander driving low-pressure stage (TCDL) cycle. It was found that the performance of the TCDH cycle was superior as compared to the other cycles. Its COP was more than 11% higher than that of the TCDL cycle, more than 9% higher than that of the single-stage compression with expander cycle and approximately 1% higher than the TCOP cycle. Zhang et al. (2017a) developed thermodynamic models to analyse the performance of six transcritical CO_2 refrigeration cycles: (a) a single-stage cycle with a throttle valve (SC), (b) a single-stage cycle with suction line heat exchanger (SCSLHX), (c) a two-stage cycle with an internal gas cooler (TCIGC), (d) a single cycle with expander (SCE), (e) a two-stage cycle with an intercooler (TCIC), (f) a two-stage cycle with flash gas bypass (TCFGB). They found that the single-stage expander cycle (SCE) had the highest COP among all the cycles. However, the TCIC cycle had the least exergy loss. Another work was reported by Zhang et al. (2018). They developed a simulation model to study the parametric sensitivity of three transcritical CO_2 cycles with expanders: 1) a conventional single-stage system with an expander, 2) a two-stage compression with an intercooler and the expander drives the second-stage compressor, and 3) a two-stage compression with an intercooler and the expander driving the first-stage compressor. The parameters studied include the inlet pressure of gas cooler, the temperatures at evaporator inlet and gas cooler outlet, the inter-stage pressure and the isentropic efficiency of expander. It was observed that maximum COP increases linearly with the expander isentropic efficiency.

To conclude this section, it can be seen that expanders are generally able to increase the COP of VCR systems, both in single and multi-stage configurations. The improvement is most significant in transcritical CO_2 systems, which involve very high operating pressures. It is for this reason that most expander studies have focused on transcritical CO_2 refrigeration systems. However, a moderate COP improvement is potentially achievable even in non-CO_2 VCR systems. It is noted that expanders should not be used in conjunction with IHXs.

3. STUDIES ON VARIOUS TYPES OF EXPANDERS

There are various types of refrigeration expander mechanisms. Traditionally, these have been modified compressor mechanisms. However, in recent years novel mechanisms have been built solely for expander applications. This section will review each of the existing expander mechanisms in the open literature. The focus will be on the reported experimental performance data.

It is noted that different authors use different parameters and terminologies to measure performance. In general, the main parameters are internal leakages, mechanical losses and irreversibility. Volumetric and mechanical efficiencies are commonly used to measure the first two parameters. Volumetric efficiency is defined as the ratio of ideal mass flow rate to the measured mass flow rate (see Equation (4)) and mechanical efficiency is the ratio between the measured nett output work to the available work assuming no mechanical losses (see Equation (5)). To measure irreversibility, various names and definitions are used in the literature, but we will use the term “indicated efficiency”, which is defined as the ratio of the measured enthalpy change to that of an isentropic one ideal one with the incomplete expansion process , as expressed in Equation (6). Another efficiency that often appears in the literature is that called “expander efficiency”. It is defined as the ratio of the measured nett output power to that if the expansion process is isentropic, as expressed in Equation (7). Reviewed data in the literature will be converted to these four efficiencies in this section to provide a fair comparison.

$$\eta_{volumetric} = \frac{\dot{m}_{ideal}}{\dot{m}_{measured}} \tag{4}$$

$$\eta_{mechanical} = \frac{W_{measured}}{W_{no\_mech\_losses}} \tag{5}$$

$$\eta_{indicated} = \frac{\Delta h_{measured}}{\Delta h_{isentropic}} \tag{6}$$

$$\eta_{expander} = \eta_{indicated} \times \eta_{mechanical} = \frac{W_{measured}}{\dot{m}_{measured} \Delta h_{isentropic}} \tag{7}$$

3.1. Reciprocating Piston Expanders

3.1.1. Piston Cylinder Expander

The piston-cylinder mechanism is among the oldest machine designs available and remains popular for compressor and engine applications. While the mechanism is reliable, it is bulky and tends to vibrate more than other mechanisms. A schematic diagram of the mechanism is shown in Figure 2. Baek et al. (2002) experimented with a piston-cylinder expansion device in a transcritical CO_2 refrigeration cycle. To control the expansion process, fast-acting solenoid valves were used as intake and exhaust valves. The design was based on a commercially available two-piston engine with a capacity of 2 × 13 cm^3. The crankshaft revolution speed was limited to a low 120 rpm because of the time lag between the input signal and the opening and closing of the solenoid valves. The experiment showed that the expander increased the system’s COP by up to 10%. In their follow-up study, Baek et al. (2005a; 2005b) developed a simulation model of the expander and carried out further experiments. The main power loss was attributed to the friction of the piston rings and the expander efficiency was 11%. Internal leakage was noted as a major issue in the study.

A three-stage reciprocating piston expander was developed and tested by Nickl et al. (2005) in a CO_2 refrigeration system. The developed prototype was tested to directly power the second stage compressor. The study found that there was a 40% COP improvement compared to a similar cycle with throttling valve. The indicated efficiency ranged from 65-70%. The expander also incorporated a new feature called a liquid-vapour separator vessel between the second and third stage of expansion, which helped to reduce the losses which mainly occurred during the final expansion process.

While the piston cylinder mechanism is very popular in engine and compressor applications, to our knowledge, there are no other publications on piston cylinder refrigeration expanders in the open literature other than those reviewed above. One of the major reasons for its unpopularity is due to the challenging control strategy required for the valves. This issue can be overcome by modifying the reciprocating piston design or by employing other types of mechanisms, as shall be seen in the subsequent discussions.

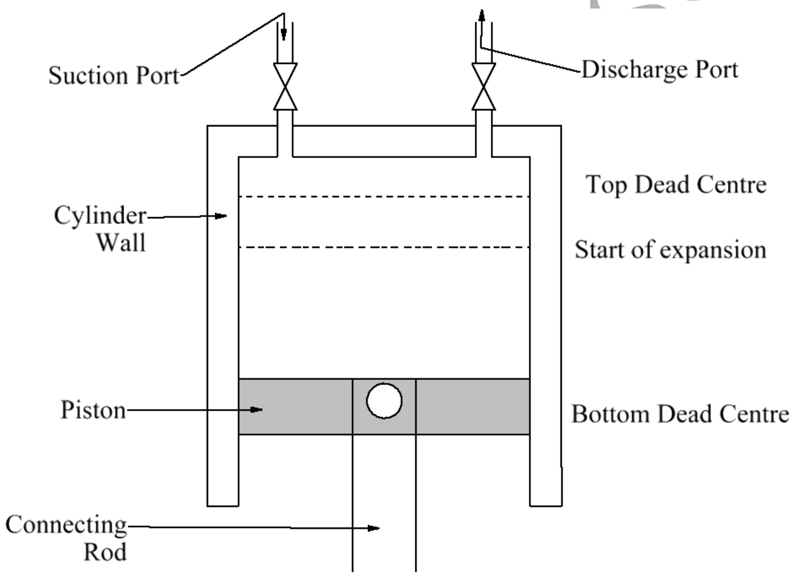

Figure 2: Schematic diagram of a piston cylinder mechanism

3.1.2. Free Piston Expander

The valve control issue of the piston cylinder expander mechanism can be overcome by employing a free-piston mechanism. Heyl and Quack (1999) were among the first to experiment with a free-piston machine in a CO_2 refrigeration system. The machine functioned as both a compressor and an expander. It was shown that the machine improved the thermodynamic efficiency of the system by utilizing 87% of the available expansion power to reduce the compression work. Throttle losses through the inlet and outlet valves were negligible.

The team of Peng et al. (2006) built a free piston expander and tested it with air and CO_2 as the working fluids. The experiments showed that the prototype could function steadily. However, high pressure was required for the expander to start and a significant pressure loss occurred in the expander-compressor unit. It was also observed that during the filling process of the expander, a large pressure drop occurred, and this was due to the pulsation at the expander inlet. Following the work of Peng et al. (2006) above, Zhang et al. (2006) further studied the slider-based free piston expander. Among other things they found that due to the use of a mechanical spring, the expander could only work well for a small range of pressure conditions. To overcome this limitation, a double acting free piston expander mechanism was proposed and built. The new prototype had issues with vibration due to unbalanced forces. In the following year, Zhang et al. (2007) reported on their progress with the prototype. Experiments were carried out with air as the working fluid, and the prototype was found to function steadily with little vibration. The pressure-temperature diagrams for the expander indicated that the slider-based inlet/outlet control scheme was successful in controlling the fluid flows. However, the inlet port was unable to function well at high frequencies, which resulted in an insufficient gas flow at the inlet and a corresponding decrease in the expander efficiency. An expander efficiency of 62% and indicated efficiency of 77.4% were measured with high and low pressures of 7.8 bar and 3.3 bar, respectively.

Wenzel and Ullrich (2012) developed a free-piston expander-compressor unit for subcritical and transcritical CO_2 cycles. They reported an increase in the coefficient of performance by 9.9%, as well as a drop in the discharge temperature and heat rejection pressure by 6 K and 4.7 bar, respectively. Guo et al. (2006) theoretically investigated a free piston expander-compressor unit in a transcritical CO_2 refrigeration system. Their study found that the indicated efficiency and expander efficiency of the prototype were 69% and 52%, respectively.

3.1.3. Radial Piston Expander

Another variation of the reciprocating piston mechanism is that of the radial piston configuration. Fukuta et al. (2014) built a radial piston reciprocating expander for a small CO_2 refrigeration system, suitable for a vending machine. Internal leakage is an important issue in such applications. Their prototype exhibited a volumetric efficiency of 70% to 90%, depending on operating speed. The prototype had a built-in a volume ratio of unity. Hence, not all of the available expansion work could be recovered. The experiments found that the mechanical, indicated and expander efficiencies of the prototype were 63%, 89% and 44% at the rotational speed of 300 rpm. The main mechanical loss was found to be caused by the O-ring seals.

Another study on a radial piston expander was reported by Ferrara et al. (2016) who manufactured a radial piston expander from a hydraulic motor and carried out experiments using CO_2 as the working fluid. The study found that friction had a great influence on the performance of the expander. The main friction losses were in the bearings and the frontal seal. The prototype exhibited volumetric and expander efficiencies of 55%-65% and 5%-19%, respectively. Based on these data, a maximum COP potential improvement of 7.4% was expected.

3.2 Rotary Piston Expanders

3.2.1. Rolling Piston Expander

Rolling piston mechanism is popular in small refrigeration and air conditioning compressor applications. It is compact and exhibits a better vibration characteristics as compared to the reciprocating piston mechanism. It generally comprises a cylinder, a rolling piston (rotor/roller), a vane and a crankshaft with an eccentric. A schematic representation of the mechanism is shown in Figure 3. When used for expanders, the mechanism needs no discharge valve, although a suction valve is still required. This makes the mechanism less practical and hence, there is only a limited number experimental studies in the literature.

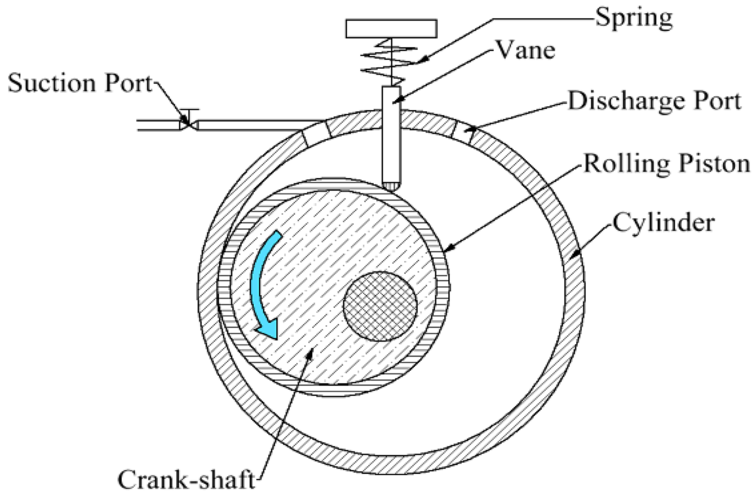

Figure 3: Schematic of a rolling piston expander

Zha et al. (2003) were among the first to study the rolling piston expander. The machine was designed for a transcritical CO_2 cycle. The study found that cavitation, sealing, high pressure differences and lubrication have significant influence on the expander's efficiency. An expander efficiency of 50% was computed.

Tian et al. (2010) built a rolling-piston expander to recover the expansion power in a transcritical CO_2 heat pump cycle. A parameter called "recovery ratio" was introduced, which was defined as the ratio of recovered power to the input power of the compressor. The study found that the recovery ratio s proportional to the expander efficiency and outlet temperature of the gas cooler. In this study, the rotational speed was an independent variable without being influenced by the inlet temperature, the inlet pressure or the mass flow rate. At the designed rotational speed of 1480 rpm, the maximum recovery ratio was 14.5% and the expander efficiency was 45%. These correspond to a COP (heating) improvement of more than 40% when compared to a system with a throttle valve. In the following year, Tian et al. (2011) reported their study on the effect of a non-condensable gas on the expander performance. Their intention was to use the non-condensable gas to accelerate the phase change process and improve the expander efficiency. Experiments were carried out using nitrogen as the non-condensable gas because of its stable properties. An average increase of 25% in expander efficiency was observed in the study. Subsequently, Tian et al. (2012) investigated the leakages losses in a rolling piston expander. Their study found that lubricant film played an important role in an expander leakage.

Li et al. (2011) experimented with a rolling piston expander in a CO_2 refrigeration system. The prototype consisted of two cylinders and a special suction mechanical control system. A cam mechanism was used for the suction control system. The prototype exhibited an expander efficiency of between 23%-58% and a COP improvement of up to 19%. It is noted that the improvement in the system's cooling capacity was insignificant due to the heat generated by the expander during expansion. Recently, Giuffrida et al. (2018) introduced a novel balanced rolling piston expander for a small scale power generation unit. The study found that although the expander could operate smoothly under the operating conditions studied, it had issues with vane acceleration, which was relatively high. This, however, could be effectively reduced by correct adjustment of the loading springs.

3.2.2. Two-Rolling Piston Expander

As mentioned above, one of the main drawbacks of the rolling piston expander is its challenging suction control strategy. To overcome this drawback, the two-rolling piston expander was introduced. It is constructed with two sets of expanders connected in series. The first expander is smaller than the second machine. Suction from the condenser is done by the first expander, while discharge into the evaporator is done by the second. This configuration requires no valve in its operation. However, it is bulky and costly as two mechanisms are required.

Yang et al. (2006b, 2010) were among the first to build and test a two-stage rolling piston expander. The two machines were connected by a common crankshaft, which consisted of two eccentrics rotating in different directions. The experimental results showed that the expander could operate smoothly. Matsui et al. (2008, 2009) investigated a two-stage rotary expander in a transcritical CO_2 system. Their prototype was integrated with a commercial scroll compressor. A dynamic analysis was carried out. The study found that the eccentricity and cylinder height of the expander have a considerable influence on its efficiency. Their expander achieved an efficiency of 54%. Jiang et al. (2013) built and tested a two rolling piston expander for a small scale transcritical CO_2 water heat pump system. They noted that the expander is more suitable for systems with larger mass flow rates and traditional synthetic refrigerants with larger expansion ratios as compared to that of CO_2. To reduce the clearance volume, the channel connecting the first and the second cylinder should be as short as possible. Their prototype exhibited an indicated efficiency of 28%-33% when operated at rotational speeds of 850-1000 rpm. Another study was carried out by Zhao et al. (2014) who simulated a two-rolling piston expander in an R-22 VCR system. The expansion ratio of this expander was around 13. A two-stage expander was chosen to cater for the large expansion ratios of low-pressure refrigerants. The study demonstrated that the expander could potentially recover around 43% of the available expansion work. Friction losses were significant, accounting for more than 78% of the total losses and leakage losses were about 16%. The main friction losses were between the vane slides and the slide grooves in both cylinders. Most of the leakage losses happened in the second cylinder at the gap between the slide and the slide grooves and the radial gap.

3.2.3. Swing Piston Expander

As can be seen from the papers reviewed in the previous sections, one of the issues with the rolling piston mechanism is the internal leakage. This can be effectively reduced by fixing the vane to the roller, which results in a mechanism known as the swing piston expander. A swing bush is then adopted to cater for the swivelling motion of the vane.

Sakitani et al. (2005) built and tested a two-swing piston expander for CO_2 heat pumps. As with the two-rolling piston expander, no valve is required for operation. The experiments were carried out according to the Japanese standard heating mode. It was able to reduce the compressor input power by 12%. The reported volumetric, mechanical and expander efficiencies of the prototype were 98%, 65% and 59%, respectively. Another study in the open literature is that from Guan et al. (2006) who tested a swing piston expander for CO_2 heat pumps. The power recovered from the expander was converted into electricity using a generator. Their experimental results showed that the expander efficiency was in the range of 28% to 44%. The maximum recovery load could be obtained by applying an optimal load to the CO_2 swing expander. The study also revealed that the expander efficiency depended on the cooling water temperature.

3.3. Vane Expander

3.3.1. Rotary Vane Expander

The rotary vane mechanism is widely used in hydraulic pumps, automobiles and air-conditioning systems. It is usually more compact than reciprocating mechanisms. In addition, it does not require a valve to control the suction and discharge processes. Unfortunately, the mechanism usually suffers from high internal leakage and friction losses. Therefore, many studies in the literature focus on reducing these losses.

The mechanism typically has a rotor with longitudinal slots, which house individual sliding vanes. There is an offset between the rotor and the stator/cylinder, creating the chamber space for the working fluid. As the rotor rotates, the vanes are pushed against the stator by centrifugal force and the chambers vary in volumes. A schematic of the mechanism is shown in Figure 4. Badr et al. (1984) were among the first to investigate the use of a rotary vane machine to capture expansion energy, although not for a refrigeration system. The mechanism typically has a built-in volume ratio of around 2-8 (Fukuta et al., 2003; Wang et al., 2012).

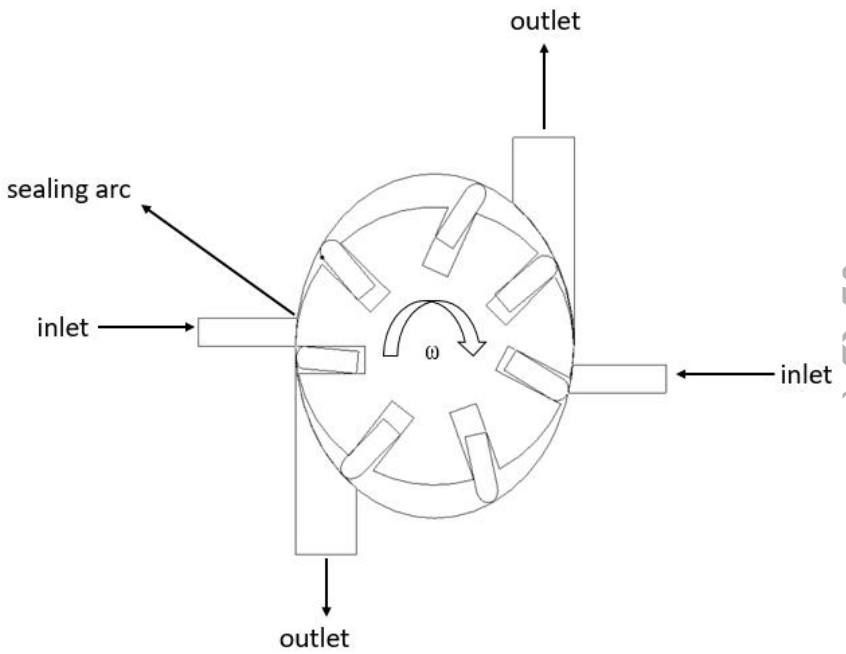

Figure 4: Schematic of a rotary vane expander

Jia et al. (2009) built a mathematical model of a vane expander. A prototype was manufactured to validate their model (Yang et al., 2009b). It was found that installing a spring in the vane slot significantly improved the performance of the expander. Their experimental results showed that these measures improved the prototype's expander and volumetric efficiencies to 23% and 30%, respectively. It was observed that the vane gap clearance had no impact on the expander efficiencies. In another paper, Jia et al. (2011) built a rotary vane expander for a CO_2 transcritical refrigeration system. The prototype introduced high-pressure gas into the vane slots to reduce internal leakage. The experimental results showed that the developed prototype exhibit a performance improvement, with expander and volumetric efficiencies of up to 45% and 35% respectively. Overall, the cycle performance of the system was improved by up to 27.2%, when compared to a throttle valve cycle working under the same operating conditions.

Rotary vane expanders have also been investigated for conventional refrigeration systems, particularly with R-410A as the refrigerant. Wang et al. (2012) simulated a rotary vane expander with two continuous internal expansion stages for a domestic R-410A air conditioner. The expander had a built-in volumetric ratio of 7.6, mechanical efficiency of 47-88%, volumetric efficiency of 37-76% and expander efficiency of 36-62%. These are higher than for conventional single-stage vane-type expanders. From their simulation, it was found that the expander could improve the system's COP improvement by 6-22%, depending on the condensation temperature and the expander's inlet degree of subcooling. A year later, Wang et al. (2013) reported further a further study, comparing modeling data with and experimental. This study found that even with R-410A as the refrigerant, the expander could improve the COP by 10.3% as compared to a system with a throttle valve. It was observed that leakage could be reduced by operating the expander at higher rotational speeds, thereby improving the volumetric efficiency. However, it also generated higher friction that eventually deteriorated the performance of the expander; however, this could be overcome by increasing the inlet pressure. The maximum volumetric efficiency and maximum expander efficiency were 55% and 31.5%, measured with rotational speeds of 1800 rpm and 1404 rpm respectively. Xia et al. (2013) conducted an experimental study of a rotary vane expander in a refrigeration system with R-410A as the refrigerant. The prototype structure was improved by installing springs in the vane grooves. The study found that lubrication of the vane reduced leakage by creating a hydrodynamic lubricating layer, which further reduced the noise and frictional losses. The experimental results showed these measures improved the prototype's expander and volumetric efficiencies by 61.1% and 57.8%, respectively, at a rotational speed of 1200 rpm. It was also observed that the developed prototype showed a COP improvement of almost 9% compared to a conventional refrigeration system.

As mentioned above, one of disadvantages of rotary vane machines is their relatively high internal leakage. A number of studies have been dedicated to studying this phenomenon. It is useful to note that leakage in an expander behave rather differently from those in a compressor because the working fluid in an expander is usually not a gas.

Fukuta et al. (2003) studied the internal leakages of a rotary vane expander and how it affected the performance of a transcritical CO_2 cycle. It was observed that the leakage could be reduced by operating the expander at higher rotational speeds. The developed prototype was found to have expander and volumetric efficiencies of approximately 43% and 64%, respectively. Later, Fukuta et al. (2006b) studied the performance of a combined vane expander-compressor unit in a two-stage CO_2 transcritical cycle with an intercooler between the two compressors. The expander was used to drive the second-stage compressor. The performance of the combined expander/compressor unit could be optimized by controlling the heat rejection pressure, which could be achieved by either pre-expansion or by using a bypass valve. The experiments found that the efficiency of the combined unit was 70%, leading to more than 51% improvement in the system's COP as compared with a normal CO_2 cycle. Fukuta et al. (2009) further investigated the rotary vane expander and found that the vane slot clearances were the major source of internal leakages in the expander. The prototype was improved by altering the vane back-pressure supply. The expander was reported to have a maximum volumetric efficiency of 70%, a mechanical efficiency of 94% and an indicated efficiency of 91% at a rotational speed of 2000 rpm. An expander efficiency of 59% was observed at the rotational speed of 2000 rpm for the improved expander.

Yang et al. (2006c) designed a rotary vane expander for a transcritical refrigeration cycle. An electric generator was used to extract the work generated by the expander during the expansion process. The experimental results showed that the prototype could function steadily for different operating conditions but internal leakages was found to be a major issue. In a following study, Yang et al. (2008) conducted a simulation and experimental investigation of the internal leakage in a rotary vane expander for a CO_2 refrigeration system. The simulation focused on the influence of the gap and the length of the seal arc using a commercial CFD software. Their prototype showed that leakages occurred mainly in the gaps between the vanes and vane slots in the rotor, the seal arc, the clearance between the rotor and the end cover and the clearance between the end surface of the vane and the end cover to the low-pressure chamber. The results showed that the main leakage was observed at the seal arc between the cylinder and the rotor, which accounted for up to 41% of the total leakage. The leakage from the end cover was about 32% and other leakages were up to 27%. The effects of these leakages reduced at higher rotational speeds. The experimental results showed that volumetric efficiency of the rotary expander was 10% to 60% at rotational speeds of 300 – 3000 rpm.

Yang et al. (2009a) conducted an experimental investigation of the internal leakage of a rotary vane expander in a transcritical CO_2 cycle. The investigation was carried out by installing and removing springs from the vane slots. It was found that with the springs, the expander and volumetric efficiencies were increased from 17% to 30% at a rotational speed of 800 rpm. The reported maximum expander efficiency was around 6%. The study also concluded that leakage affected performance more than friction. In another paper, Yang et al. (2009b) reported another study with their rotary vane expander in a CO_2 transcritical cycle. Experimental results showed that the expander and volumetric efficiencies were increased from 9% to 23% and 17% to 30%, respectively. They showed that the expander was able to improve the COP by up to 14.2%, compared with the throttling cycle. Fukuta et al. (2010) studied transcritical leakage flows in a CO_2 rotary vane expander. There was a good agreement between the numerical modeling and experimental mass flow rate with a difference of $\pm$10%. The study found that friction losses were dominant in the two phase and supercritical regions. It was observed that careful mixing of the lubricating oil reduced the refrigerant leakage. However, the sealing effect of the oil on the transcritical leakage flow was weak. Mahmoud et al. (2010) also investigated the internal leakages of rotary vane expanders. They found that internal leakage losses mainly depended on operational speeds and geometrical parameters of the expander, including the curvature of the vane tip, the number of vanes and inaccuracy in the gap size.

3.3.2. Revolving Vane Expander

As mentioned above, rotary vane machines generally have more leakage and friction losses which reduces the efficiency. One of the methods to overcome these limitations is to modify the design such that the cylinder rotates together with the rotor during operation, which results in a reduced relative velocity between the rubbing surfaces. This is called the revolving vane mechanism. A schematic representation of the mechanism is shown in Figure 5. Subiantoro and Ooi (2009) were the first to study the mechanism for expander applications. Their simulation predicted a mechanical efficiency of up to 94%, which is higher than other rotary mechanisms.

There are variations of the mechanism design, including the various vane slot design and vane-rotor-cylinder arrangement. Subiantoro and Ooi (2012a) built a revolving vane expander where the vane head was located at the cylinder and the vane is allowed to swivel relative to the cylinder as it slides in and out of the straight slot at the rotor. A simulation model was developed for the machine (Subiantoro and Ooi, 2012b). The study was carried out with compressed air as the working fluid. The efficiency of the expander was found to depend on the rotational speed but, interestingly, not on the operating pressure. This was due to the unique vane-rotor arrangement of revolving vane mechanisms. Volumetric and expander efficiencies of 55% and 9%, respectively, were measured. The main source of frictional loss was at the journal bearings. The study also found that major leakage paths were at the lip seal and radial clearances. The leakage through the radial clearance was dominant, contributing 67% - 93% of the total leakage.

A year later, Subiantoro et al. (2013) built and studied another revolving vane expander prototype using a different configuration where the vane was rigidly fixed to the rotor and a specially designed vane slot was located at the cylinder. The prototype was tested with compressed air as the working fluid. The experiments showed that the prototype had volumetric and expander efficiencies of up to 57% and 8%, respectively. Subiantoro and Ooi (2010a, 2010b, 2014) simulated the performance of four different revolving vane expander configurations. They concluded that the vane should be rigidly fixed to the driving component in order to reduce friction losses.

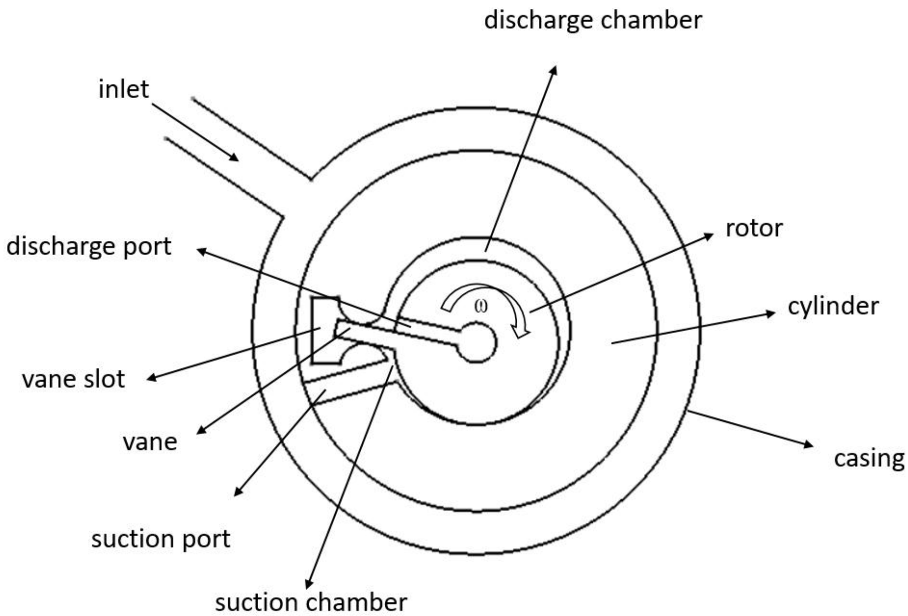

Figure 5: Schematic of a revolving vane expander

3.3.3. Cross Vane Expander Compressor Unit

Another design evolution of the rotary vane mechanism is the Cross-Vane Expander-Compressor (CVEC) unit introduced by Yap et al. (2014a). The advantage of this mechanism is it functions both as a compressor and an expander simultaneously. Therefore, it is more compact when compared to the conventional compressor-expander arrangement. A theoretical investigation was conducted using R-134a as the working fluid (Yap et al., 2014b). It was observed that, because of the combined expander-compressor unit, the mechanical losses associated with CVEC was less than other types of rotary expander-compressor systems. The mechanical efficiency of the CVEC model was around 96.5% and an energy saving of up to 18.0% was calculated. Subsequently, Lim et al. (2017) developed a numerical model to study the main design parameters. They concluded that proper optimization of the main design parameters could reduce the friction power losses by up to 50% and improve the COP by 58%. Recently, Yap et al. (2018) built a CVEC prototype and tested it with compressed air. The experiments showed that the developed prototype could function steadily. Frictional losses at the endface were found to account for 81.2% of the total losses. Their simulation predicted a COP improvement of 36.6% when the prototype was used in a transcritical CO_2 refrigeration system.

3.4. Screw Expanders

Screw mechanisms are popular for large-scale applications. They have excellent performance but are complex to manufacture and costly to maintain. The profile of the rotor plays an important role in the reduction of frictional losses and leakage and overall cycle performance of the expander (Stosic et al., 1997). In one of the designs, a pair of male and female rotors are used. The male rotor is always connected to the power source, acting as the driving rotor, while the female rotor is driven by the male. When the two rotors rotate, the volumes in their grooves varies.

Kovacevic et al. (2006) conducted a simulation of a combined screw compressor-expander machines for use in CO_2 refrigeration systems. Oil was injected into the system to seal, lubricate and cool the high-pressure gas. Gears were not used to synchronize the rotors. They found that these measures could reduce the radial loads on the bearings by 20%, since the pressure loads in the compressor and the expander were located almost opposite to each other near the center of the rotors. In the design of CO_2 systems, reduction of bearing load was of importance.

Wang et al. (2011) tested a single screw expander prototype with wet air as the working fluid. The reported peak value of the shaft power was 5 kW at a rotational speed of 2850 rpm. Their experiments showed that the prototype exhibited maximum indicated and expander efficiencies of 59% and 35.4% respectively. Lu et al. (2013) built a single screw expander and tested it in a compressed air refrigeration system. The performance showed little variation with inlet pressure. However, as the rotational speed increased, the performance of the system was improved. An indicated efficiency of 65% was measured. Wu et al. (2017) proposed a simplified model to calculate the torque characteristics of a single screw expander. It was found that the torque ratio does not depend on the initial pressure levels. The study found that there was a good agreement between the numerical modeling and experiment, with the output power showing a small difference of 2.07 kW to 2.37 kW. More recently, Shen et al. (2018) conducted an experiment and simulation to understand the characteristics of a single screw expander. The study found that higher volumetric efficiencies and power outputs could be achieved at higher rotational speeds. The prototype exhibited a maximum volumetric efficiency and power output of 88.53% and 3.37 kW at a rotational speed of 3000 rev/min. In another study, Li et al. (2018a) studied the effect of varying the rotational speed and inlet pressure on the performance of a high-pressure single screw expander. The experimental results showed that the maximum power output, expander, volumetric and indicated efficiencies were 56.55 kW, 50.96%, 80.57% and 62.55%, respectively, for an inlet pressure 5.0 MPa. They concluded that leakage could be reduced by operating the expander at higher rotational speeds.

Kovacevic et al. (2013) performed a CFD analysis of a twin-screw expander using compressed air as the working fluid. Comparison with measurement data showed that the model's accuracy improved at higher rotational speeds. The study found that performance of the expander mainly depended on the clearances of the interlobe, axial and radial gaps.

3.4.1. Integrated Unit of a Screw Expander and a Compressor

As compared to other expander mechanisms, attempts to integrate screw expanders with compressors have a long history. Smith and Stosic (1995) who introduced a combined expander-compressor unit and called it an expressor. It was built for power recovery in a refrigeration system. Later, Stosic et al. (2002) simulated a twin-screw combined compressor and expander for a CO_2 refrigeration cycle. They showed that the machine could balance its axial loads to minimize the radial bearing load by 20%. This solved the issue of high radial loads in high operating pressure conditions, which could lead to an excessive deflection to the rotor and cause a high wear in the bearings. It was noted that balancing the bearing forces relative to the pressure distribution within the machines was challenging, though. Kovacevic et al. (2006) further studied the combined screw compressor-expander machines in high-pressure refrigeration systems with CO_2 as the working fluid. The investigation was carried out using a three-dimensional Computational Continuum Mechanics simulation. The results show that the compressor-expander model reduced the radial and axial forces acting on the rotors and also increased the COP of the system. This enabled the bearing losses to be reduced and the maximum permissible operating pressure difference for the twin screw compressors to be increased.

3.5. Scroll Expander

The scroll mechanism is widely used for compressor applications in air conditioning systems. The mechanism consists of a fixed and a rotating or orbiting scroll. The advantage of a scroll machine is that it has few moving parts and their operation produces less vibration and noise compared to other mechanisms. Furthermore, they do not require any valves and thus, it is relatively simple to convert a scroll compressors into an expander. A schematic of the mechanism is shown in Figure 6.

A simulation model considering the effects of heat transfer, internal leakage and valve losses was developed to estimate the performance of compression and expansion processes in a scroll compressors and expanders by Huff et al. (2002). The COP and capacity of the system were improved by 40-70% and 5%-15%, respectively. Westphalen (2004) designed a scroll expander for use in a military environmental control unit based on a CO_2 refrigeration system. The prototype was shown to have an overall efficiency of 72%. Following on the study, Westphalen and Dieckmann (2006) integrated the scroll expander with a dual-stage rotary (rolling piston) compressor for use in a CO_2 cycle. The geometrical parameters of the scroll expander were optimized for high ambient (51.7°C) and evaporator air return conditions (32.2°C). A static leakage test was carried out to observe the leakage of the fluid through the expander, which was found out to be only around 10%.

Fukuta et al. (2006a) investigated the performance of a scroll expander in a CO_2 refrigeration cycle. The volumetric efficiency of the expander was approximately 80% and was not significantly affected by the operating speed. The maximum expander efficiency of the expander was approximately 55% at 3500 rpm. Kohsokabe et al. (2008a) also studied the performance of a scroll expander in a CO_2 refrigeration cycle. The prototype had a built-in volume ratio of 2.0 and an inlet volume of 2800 mm^3. The experimental results showed that maximum expander efficiency was 83% and the optimal speed was between 2200 to 3400 rpm. It was also observed that the optimal pressure ratio of the expander was slightly higher than the ideal pressure ratio. The study found that volumetric and isotropic efficiencies were a function of the mass flow rate of the refrigerant. Hiwata et al. (2008) experimented with axial and radial force control for their CO_2 scroll expander prototype. Characteristics of the radial force and oil film pressure were analysed and the results were validated using their experimental data. The prototype was found to have an expander efficiency of 62%. Subsequently, Hiwata et al. (2009) further developed and studied their prototype. They redesigned the gap profile to exploit overexpansion and to control the axial force on the thrust bearings. The study found that two leakage paths were affecting the volumetric efficiency of the expander, namely the gap in the radial direction between the wraps, and the gap in the axis at the tip of the wrap. However, these problems could be overcome by using a high pressure in the slots and a 96% volumetric efficiency was obtained. The expander efficiency was measured as being up to 62%.

Zhang et al. (2017b) built a scroll expander and tested it using air as the working fluid. Their prototype exhibited a power output and expander efficiency of 1.2 kW and 30.5%, respectively. The study showed that temperature had no infuence to the expander's power output. The clearance gap sizes had a significant impact on the expander performance. When the clearance gap increased, the flowrate increased, the power output first increased, then decreased slightly and finally levelled off, while both the volumetric and expander efficiencies decreased. The maximum volumetric efficiency of the prototype was reported to be 44.2%. The study concluded that a favorable clearance value should not exceed 0.06 mm. Singh et al. (2017) conducted a CFD simulation of a scroll expander for a transcritical CO_2 refrigeration system. A dynamic mesh was adopted for the transient flow field with a geometry that changed with time. They found that the expansion process in the expander was significantly affected by the leakage path between the moving and the fixed scrolls. The fluid was observed to flow in a vortex in the expansion chamber, resulting in a periodic fluctuation in the fluid velocity.

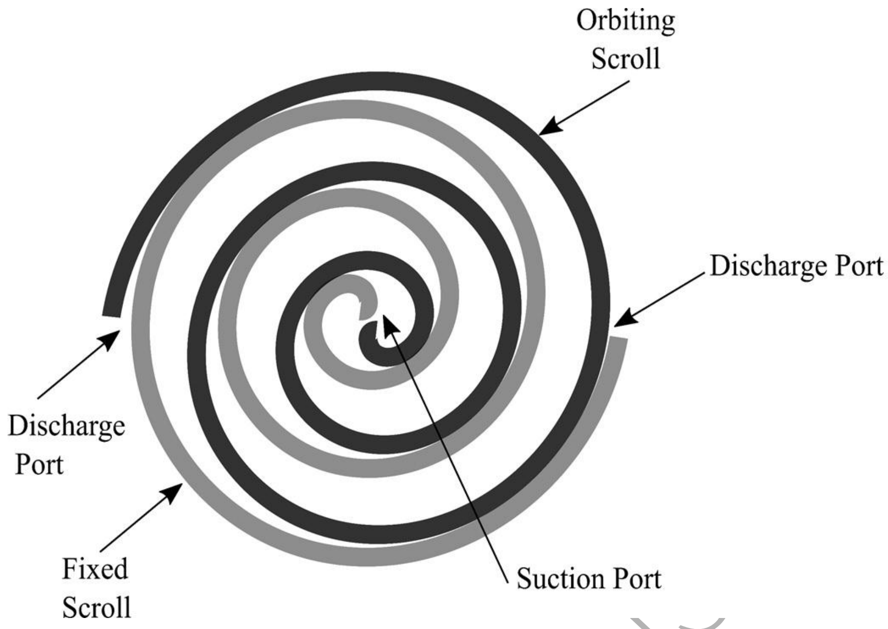

Figure 6: Schematic diagram of a scroll expander

3.5.1. Integrated Unit of a Scroll Expander and a Compressor

Kohsokabe et al. (2006) studied a combined scroll expander and a rolling piston type rotary sub-compressor for an air-cooled CO_2 chiller system. The expander and compressor were connected through their main shafts. The study found that the expander efficiency of the expander-compressor unit was 57%, leading to a more than 30% improvement in the system performance. Kim et al. (2008) simulated a combined scroll expander-compressor unit for a CO_2 transcritical system. Initially, the system implemented a two-stage compression arrangement with intercooling by making use of scroll compressors. The first stage compressor was directly attached to the expander shaft, which unfortunately resulted in a significant leakage. With the gas cooler exit temperature of 35°C and compressor inlet and outlet pressures of 35 bar and 100 bar, respectively, the overall efficiency of the expander was observed to be 54.4%. Their prototype exhibited an improvement in cooling capacity and performance of the system by 8.6% and 23.5% respectively. It was also observed that the expander efficiency was relatively stable for different operating pressure ratios, but the compressor efficiency significantly reduced as the inlet pressure increased. Kakuda et al. (2009) and Nagata et al. (2010) conducted an experimental investigation to understand the characteristics of a scroll type expander-compressor for a two-stage CO_2 refrigeration cycle. The expander-compressor unit was used for the second stage compression. The measurement showed a volumetric efficiency of 104% based on the state at the expander inlet and an up to 30% increase was observed in the COP. They concluded that heat leakage and pressure losses were significant for the cycle efficiency.

3.6. Turbo Expander

Turbines (or sometimes called turbo expanders) are the most widely used mechanism to capture expansion power in power plants. They are efficient and suitable for high flow rate with low pressure drop conditions. However, they are less efficient for high pressure drop conditions such as occur in refrigeration systems, and therefore there are not many studies of this mechanism.

Hays and Brasz (2004) developed an integrated turbine-compressor unit for a two-stage compression transcritical CO_2 refrigeration cycle. The integrated unit was for the first compression stage. The turbine's rotational speed was 110,000 rpm, the expander efficiency was 69% and the system's COP was improved by 39% when compared to a system with a throttling valve. Hou et al. (2014) investigated a turbo expander for a transcritical CO_2 refrigeration system with a refrigeration capacity of 15 kW. Based on the experimental data, a maximum COP improvement of 7% was obtained. Conversely, Zhang et al. (2015) studied a turbo expander in a subcritical CO_2 refrigeration system. They found that the operational speed and inlet parameters have a great influence on the expander efficiency. Their experiments found a maximum torque of 0.65 Nm and a maximum expander efficiency of about 10.4% at a rotating speed of 2000 rpm. It was also observed that the optimum velocity ratio shifts to higher values as the expander inlet temperature increases. Verma et al. (2015) conducted a CFD simulation of a turbo expander for cryogenic refrigeration and liquefaction cycles. They studied various design parameters and concluded that transient models were essential to understand the stator-rotor interaction and trailing edge vortices.

Tondell (2006) studied an impulse turbine for a CO_2 refrigeration system. The prototype showed low expander efficiency due to the high frictional losses exhibited. The losses could be reduced by using a smaller turbine wheel. He et al. (2009) designed and fabricated a Pelton turbine and integrated it with a generator for an R-410A domestic refrigeration system. The developed expander consisted of two nozzles and one simplified twin arc blade impeller. The prototype was found to function steadily under two-phase conditions with rotational speeds ranging from 5710 to 26500 rev/min. It was also observed that the maximum expander efficiency was 32.8%. The cooling capacity and COP increased by 6.5% and 5.4%, respectively, when the expander was integrated with an auxiliary centrifugal compressor in series to the main compressor. Recently, Li et al. (2018b) investigated the performance of a liquid turbine in a supercritical compressed air energy storage system. The developed prototype was tested under a range of operating conditions. The experimental results showed that the prototype exhibited a maximum expander efficiency and power output of 75.16% and 30.44 kW respectively.

Cho et al. (2008) experimented with an axial turbine for an R134a air conditioner. The prototype exhibited an indicated efficiency of 15.7%. Experiments were carried out and it was observed that the indicated efficiency was directly proportional to subcooling temperature or pressure ratio. The maximum efficiency at the specified output increased steadily as the enthalpy difference increased. However, they decreased when the mass flow rate increased. Zhang and

Tian (2014) used energy and exergy analysis to study an impulse turbo expander in an R134a refrigeration cycle. Interestingly, they also used the expander for economizer (vapour injection) purposes too. The study found a COP improvement of up to 21.6% over a basic cycle.

3.7. Summary

Various expander mechanisms have been studied over the years to recover the expansion energy in a refrigeration system. Most are based on existing compressor mechanisms. Each mechanism has its own strengths and weaknesses. Arguably the two most popular expander mechanisms are the rotary vane and scroll expanders, which need no valves to control the suction and discharge processes. Most of the studies used CO_2 as the refrigerant, mainly because of the high pressure conditions associated with CO_2 transcritical refrigeration cycles which results in a large irreversible loss in a throttling tube. Compressed air is also popular in expander studies, particularly for studying new mechanisms. Other refrigerants that have been studied include R-22, R134a, and R-410A, but most of the works are theoretical and there are only a few experimental studies in the literature using these conventional refrigerants.

The overall highest measured expander efficiency is 83%, reported by Kohsokabe et al. (2008a) in a transcritical CO_2 refrigeration system with a scroll expander. With a conventional refrigerant (R-410A), the maximum reported expander efficiency is 61% with a rotary vane expander (Xia et al., 2013). The highest COP improvement measurement reported for a CO_2 system was approximately 30% (Kohsokabe et al., 2006; Kakuda et al., 2009) and for a non-CO_2 system (R-410A) was 10.3% (Wang et al., 2013). A summary of various expander experimental data available in the literature is tabulated in Tables 1-3.

Table 1: Reported Performance of Expanders with CO_2 (in chronological order)

Authors	Expander type	Operating speed (rev/min)	Operating pressures (MPa)	Inlet temperature (°C)	Max. expander efficiency (%)	Max. mechanical efficiency (%)	Max indicated efficiency (%)	Max. volumetric efficiency (%)	Max. COP improvement (%)
Fukuta et al. (2003)	Rotary vane	500-2000	4.1-9.1	40	43	-	-	64	-
Hays and Brasz (2004)	Turbo	12800	-	-	69	-	-	-	-
Baek et al. (2005a)	Piston cylinder	-	3.4-10.2	26.0-31.4	11	-	-	-	10.5
Sakitani et al. (2005)	Swing piston	2400-5400	3.4-9.4	30	59	65	93	98	12
Fukuta et al. (2006a)	Scroll	2000-4000	4-10	40	55	-	-	80	-
Guan et al. (2006)	Swing piston	600-2000	4-10	29-42	44	-	-	-	-
Kohsokabe et al. (2006)	Scroll expander-Compressor	-	3.8-9	36	-	-	57	-	30
Zhang et al. (2007)	Free piston	-	0.33-0.78	-	62	-	77.4	-	-
Kohsokabe et al. (2008a)	Scroll	1000-4000	8.2-3.96	36	83	-	-	-	-
Yang et al. (2008)	Rotary vane	500-3000	-	-	-	-	-	60	-
Fukuta et al. (2009)	Rotary vane	1000-2500	4.1-9.1	35-45	59	94	91	70	-
Hiwata et al. (2009)	Scroll	3000	4.1-7.1	20	62	-	-	96	-
Kakuda et al. (2009)	Scroll	2760	1.30-3.19	-	-	-	-	104	30
Matsui et al. (2009)	Rotary vane	-	-	-	54	-	-	-	-
Yang et al. (2009a)	Rotary vane	300-1500	7.5-9	32.4-44.3	6.2*	-	-	30	-
Yang et al. (2009b)	Rotary vane	100-2000	2.0-8.5	-	23	-	-	30	14.2
Nagata et al. (2010)	Scroll	2760	1.69-4.16	-	-	-	-	104	-
Tian et al. (2010)	Rolling piston	500-1900	8-12	35	-	-	45	-	-
Yang et al. (2010)	Rolling piston	1300-3100	4.6-7.2	37	-	-	-	-	-

Jia et al. (2011)	Rotary vane	800-2000	3.5-9	34-42	45	-	-	35	27.2
Li et al. (2011)	Rolling piston	850-2500	-	35.8-38.0	58.7	-	-	-	19
Wenzel and Ullrich (2012)	Free piston	-	-	-	-	-	-	-	9.9
Jiang et al. (2013)	Rolling piston	850-1000	4-9	35	-	-	33	-	-
Fukuta et al. (2014)	Radial piston	180-400	3.2-8.5	25	44	63	89	80	-
Zhang et al. (2015)	Turbo	900-3200	0.5-1.7	5-8	10.4	-	-	-	10.9
Ferrara et al. (2016)	Radial piston	300-1000	-	-	19	-	-	65	7.4

*Re-calculated

Table 2: Reported Performance of Expanders with Other Refrigerants (in chronological order)

Authors	Expander type	Working fluid	Operating speed (rev/min)	Operating pressures (MPa)	Subcooling Temperature (°C)	Max. expander efficiency (%)	Max. mechanical efficiency (%)	Max indicated efficiency (%)	Max. volumetric efficiency (%)	Max. COP improvement (%)
Cho et al. (2008)	Turbo	R-134a	3600	0.6-1.9	2-15	-	-	15.7	-	-
He et al. (2009)	Turbo	R-410A	4000-28000	-	2-3	32.8	-	-	-	5.4
Wang et al. (2013)	Rotary vane	R-410A	800-1800	3-3.6	8.3	31.5	-	-	55	10.3
Xia et al. (2013)	Rotary vane	R-410A	800-1800	1-3.4	2-10	61.1	-	-	57.8	8.97

*Re-calculated

Table 3: Reported Performance of Expanders with Compressed Air (in chronological order)

Authors	Expander type	Operating speed (rev/min)	Operating pressures (MPa)	Inlet temperature (°C)	Max. expander efficiency (%)	Max. mechanical efficiency (%)	Max indicated efficiency (%)	Max. volumetric efficiency (%)
Wang et al. (2011)	Screw	400-3000	0.6-1	17	32.5*	60	59	-
Subiantoro and Ooi (2012a)	Revolving vane	100-900	0.2–0.6	-	9*	-	-	55
Subiantoro et al. (2013)	Revolving vane	100-600	0.3	-	8*	-	-	57
Lu et al. (2013)	Screw	1400-2800	0.8-1.6	23	-	-	65	-
Zhang et al. (2017b)	Scroll	300-1200	0.7-1	-	30.5	-	-	44.2
Li et al. (2018a)	Screw	1500-3000	1-5	-	50.9	-	62.5	80.5
Shen et al. (2018)	Screw	2000-3000	-	18.5-20.74	-	-	-	88.53

*Re-calculated

4. VARIOUS ASPECTS OF EXPANDERS

Many studies have reported on the general performance (including expander and volumetric efficiencies) of various types of expander as reviewed in the previous section. Recently, more detailed studies have considered other aspects of expanders. These include, among others, the heat transfer, leakage, expansion process, irreversibility, lubrication, control, integration with compressors, power utilisation strategy and economic aspects. This section will review these studies. However, as can be seen, the material is limited for most of these topics.

4.1. Heat Transfer

Heat transfer between the fluid and the solid components affects the isentropic efficiency of expanders. As the refrigerant is expanded, its temperature decreases. Meanwhile, the expander walls increase in temperature due to friction. Therefore, heat is typically transferred from the hot expander walls to the cold fluid. This is different from that in a compressor where the fluid's temperature rises as it is being compressed. In addition, the fluid phase in an expander usually contains liquid, while that in a compressor is entirely gaseous. However, few studies are available on the heat transfer processes in an expander. In most studies, either heat transfer is ignored or models from compressors or engines are used.

Huff and Radermacher (2003) modelled the expansion process of CO_2 in a reciprocating expander. The study predicted that heat transfer could be neglected from the expansion process, particularly when the expander operates at higher rotational speeds. Fukuta et al. (2003) developed an analytical model for a vane expander to observe the expansion process from a super-critical condition to a two phase one in a transcritical CO_2 system, including the corresponding heat transfer processes. The study found, among other things, that the influence of the heat transfer on the performance of the expander was small as compared to the influence of internal leakage. Later, Fukuta et al. (2013) used a single piston cylinder machine to study the expansion of CO_2. They reported that during expansion, a heat transfer coefficient of 2-7 kW/(m^2·K) was measured experimentally. This coefficient increased with rotational speed and increased slightly when oil was not present.

4.2. Expansion process of refrigerants

The expansion process in a refrigeration expander involves a liquid-vapour mixture. This is different from that in a compressor that only deals with gases. Fukuta et al. (2008) observed the expansion process of CO_2 in a piston-cylinder expander in a transcritical refrigeration cycle. While the expansion process moved into the two-phase region from the supercritical region on a pressure-enthalpy diagram, a phase transition is observed. In general, this transition occurred under non-equilibrium conditions. Therefore, the expansion process was convoluted and unclear. Later, Fukuta et al. (2013) used a high-speed camera to observe the expansion process of CO_2 in a single piston expander as it moved from the supercritical region to the two-phase region. As the fluid changed phase from liquid/supercritical-gas to liquid-vapour mixture, a blackout phenomenon caused by the generation of fine mist was observed. The study also found that during expansion, the enthalpy of the fluid increased. This suggests that the expansion is far from isentropic, since for an isentropic process the enthalpy of the fluid would be expected to decrease. The increase in enthalpy was related to the heat transfer that occurred from the expander to the working fluid.

4.3. Exergy

Exergy analysis is increasingly popular to study the losses in a system. This analysis provides a good indicator of the sources of irreversibility, which helps optimization of the system performance. Exergy analysis has been adopted for expanders, albeit to a limited degree.

Zhang and Tian (2014) used exergy analysis to study an R134a refrigeration cycle with an expander. Among other things, they found that the main irreversibilities were in the condenser and the compressor. However, the analysis was on the system's level and did not study the irreversibilities in the expander.

Subiantoro et al. (2016) applied exergy analysis to a revolving vane expander when used with compressed air. The analysis was carried out with benchmark conditions of 8 bar suction pressure, 1 bar discharge pressure and a rotational speed of 3000 rpm. It was observed that the exergy efficiency reduced with higher cylinder inertias and higher rotational speeds. The main irreversibilities were mechanical friction at the vane and heat transfer in the suction chamber.

4.4. Lubrication

Lubrication plays an important aspect in machinery. It seals leakage gaps, reduces friction and cools components. Lubrication design of an expander has its unique challenges because of the low-temperature and low-pressure conditions involved during operation. The low-temperature condition means oils intended to be used at high temperatures such as in compressors and internal combustion engines may not be suitable for use in expanders. Moreover, the working fluid usually is not in gaseous phase and thus, the oil miscibility must be properly checked. The low-pressure condition means the lubrication flow network must be properly designed to suit the pressure distribution. A lubrication pump can be employed. However, this requires additional cost and power requirements, and therefore using a lubrication pump is usually not desirable.

To the authors' knowledge, there are no in-depth studies on the lubrication of expanders. However, some notes are available on the effect of lubrication to the expander performance. For example, Subiantoro and Ooi (2012b) showed that in a revolving vane expander, contact between the tip of the vane and the piston results in vane tip frictional force. Lubricating the contact area or improving the surface finish will minimizes the frictional force. Wang et al. (2011) showed that inadequate lubrication affected the adiabatic efficiency of a screw expander. The lubricating oil was injected into the gate rotor room. The study found that the expander power output was significantly impacted by the friction of the lubricating oil on the gate rotor. Similarly, Jiang et al. (2013) showed that proper lubrication could mitigate the frictional loss that occurred due to movement between the two contact parts in a swing piston expander.

Another challenge regarding expander lubrication is that the oil is injected to the components inside the expander through the lubrication system, but this lubricating oil is not usually separated at the outlet of the expander. Therefore, the oil flowing out from the expander tends to depress heat transfer in the evaporator. This is an important issue which needs to be considered when designing the lubrication system for an expander.

4.5. Control Strategy

A control strategy is required for an expander to allow it to respond to the changes in the operating conditions of the system. This is important to optimise the expander's performance and to protect it and the system from damage. The two main strategies in the literature are by regulating the inlet flow or by controlling the operating speed of the expander.

Cleveland (1986, 1988) first studied the parameters governing expander performance and also the connection to the expander alternator systems. They found that, among other factors, the operating pressure has a significant impact on the power output from the generator. Following on from this work, Barmby and Cleveland (1990) developed a basic control system using an upstream valve to regulate the inlet pressure and flow rate for the expander. This provided protection against over speeding and a reasonably constant inlet pressure.

Zhang et al. (2007) investigated how expander speed affected their expander's performance. They found that a relationship between the expander speed and the inlet pressure. Their experiments showed an optimal frequency for their prototype of between 10 Hz to 17 Hz. When the expander was connected to a generator, the speed could be adjusted by controlling the electrical resistance load, as was done by Yang et al. (2009a, 2009b), Jia et al. (2011) and Wang et al. (2012). By correctly controlling the expander speed, the power output and expander efficiency could be optimised.

4.5. Installation Strategy and Issues

In general, there are two strategies to install an expander into a refrigeration system. The first strategy is to directly couple the expander and compressor so that the power generated by the expander can be used directly to reduce the compressor load. The second method is to separate the expander from the compressor and capturing the power generated by the expander using an alternator. The second method requires a simple compressor-expander control strategy, but it requires more components, is more costly and may generate more power losses as compared to the first.

Yang et al. (2007) compared the direct and indirect coupling of the integrated expander-compressor unit in a CO_2 cooling system with single and dual compression arrangements. The single stage compression systems were arranged with an expander or a conventional expansion valve. The two stage compression systems consisted of three configurations; (i) an expander indirectly driving the low-pressure compressor with optimized intermediate pressure (DCOP), (ii) an expander directly driving the high-pressure compressor (DCHP), or (iii) an expander directly driving the low-pressure compressor (DCLP). The study found that the DCHP system gave the optimum results. It was noted that the optimum inter stage pressure was higher than the geometric mean pressure, which is typically used to optimize the configuration of an expander in a subcritical two-stage compressor. Further, it was observed that there was a 9% COP improvement by using two stage compression with an expander.

Another important consideration, which has not been studied in detail to date, is torque matching between the compressor and expander. One study by Subiantoro and Ooi (2015) was for an integrated revolving vane compressor-expander unit. When properly matched, the integrated unit would minimize both peak and average torques. The study found that by properly optimizing the peak torque matching, the efficiency could be improved by 65% and the bearing loads reduced by 25%.

4.7. Economic Analysis

One of the most obvious practical challenges in the adoption of expanders in commercial refrigeration systems is the additional capital cost imposed. Hence, the main question is whether expanders are worth using or not? (Hwang, 2009). Over the years, there have been attempts to answer this question.

One of the main challenges in answering the question is that the cost of an expander when it is manufactured in large quantities is still unknown. Riha et al. (2006a, 2006b) predicted it would be on the order of a third of the cost of one main compressor. Henderson et al. (2000) studied the economics of a compressor/expander unit for an R-410A heat pump system. They concluded that when this heat pump is used to replace gas boilers, the payback time can be reduced by about 20% as compared to a conventional heat pump. Subiantoro and Ooi (2013) carried out a more comprehensive economic analysis of expanders in medium scale air conditioners with various refrigerants, including CO_2, R-22, R-134a, NH3, R-32, R-404A and R-1234yf. They assumed that the additional cost imposed by the expander is less than the price of a compressor due to expander's smaller physical size. It was also assumed that the system is in operation for 8 hours per day and 250 days annually. The study concluded that the payback times were generally less than 5 years when the expander efficiency was 50%. Other than for the CO_2 system, an expander was particularly attractive for R-404A systems where the payback time was less than 3 years. Expanders were most attractive for high ambient temperature applications.

An integrated compressor/expander unit may have shorter payback times since it might be expected to be cheaper than the traditional separate compressor and expander arrangements (Heyl et al., 1998).

5. CHALLENGES AND RECOMMENDED FUTURE WORK

Although much work has been done on expanders in refrigeration systems, particularly in transcritical CO_2 systems, a few key issues remain for further investigation. Some issues that still need to be further studied include the following:

1) Expanders in non-CO_2 systems have been shown to be able to give a COP improvement of around 10%. However, the studies in such systems are still very limited. Considering the fact that most commercial refrigeration systems do not use CO_2 as a refrigerant, more studies of expanders in such systems are required.
2) There is little information available on expander processes and characteristics, such as heat transfer, fluid-structure interaction, expansion, internal leakage and irreversibility, and more studies on these topics are required.
3) Further development of control strategies that allow expanders to dynamically react to changes in operating conditions is identified as a crucial step towards their commercialization.
4) One of the major limitations of existing expanders is their inability to adjust to different operating pressure conditions. This is because, unlike compressors, the expansion process is usually determined by geometry (volume ratio) and not by valve timing. A new expander design that allows such flexibility, or a valve design that is suited for expander applications would be beneficial for further expander development.
5) Further studies to determine the optimum way to utilize expander power in a refrigeration system are required. Expanders can be directly coupled to a compressor or connected to a generator to produce power. Each method has its own strengths and weaknesses. In general, directly coupling the expander to the compressor seems to be a logical option. Mechanisms that combine the expander with the compressor, such as the free piston

expander-compressor (Heyl and Quack, 1999) or the more recent cross-vane expander-compressor unit (Yap et al., 2014a) are promising.

It is worth noting that to the authors' knowledge, expanders have yet to be adopted in commercial refrigeration systems. It is to be hoped that further research will demonstrate their commercial feasibility.

6. CONCLUSIONS

Recovery of expansion energy using expanders is a promising method to improve energy efficiency of refrigeration systems. Many studies have reported their effectiveness, particularly for transcritical CO_2 refrigeration systems. A number of mechanisms have been studied, including the reciprocating piston, rolling piston, rotary vane, scroll, screw, and turbine expanders. The highest experimentally measured expander efficiency in the literature is 83% for a scroll expander in a CO_2 refrigeration cycle (Kohsokabe et al. 2008a), while that for a conventional refrigerant (R410A) is 61% for a rotary vane expander (Xia et al., 2013). Most of the studies have focused on small scale refrigeration systems with a capacity of less than 5 kW. The highest COP improvement reported for CO_2 systems was 30% (Kohsokabe et al., 2006; Kakuda et al., 2009) while that for non-CO_2 systems (R-410A) was 10.3% (Wang et al., 2013).

Issues relating to the application of expanders such heat transfer, expansion process, internal leakage and irreversibility have started to be studied, but more research is required. Control strategies that allow expanders to react dynamically to changes the operating conditions of a refrigeration system are required for commercialization of the technology. Further development and optimization of expander designs and integration with compressors are also identified as crucial aspects for future progress of the technology.

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