Chapter 3

Expanders for Organic Rankine Cycle Technology

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Additional information is available at the end of the chapter

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Abstract

The overall power conversion efficiency of organic Rankine cycle (ORC) systems is highly sensitive to the isentropic efficiency of expansion machines. No expansion machine type is universally ideal as every machine has its own advantages and disadvantages and is suitable for a comparatively narrow range of operations of the highest efficiency. Therefore, an optimum selection of an expansion machine type is important for a financially viable ORC implementation. This chapter presents the mode of operation, technical feasibility, and challenges in the application of turbo-expanders (radial inflow, radial outflow, and axial machines) and volumetric expansion machines (scroll, screw, piston, and vane) for use in ORC systems. It can be concluded that different machines are suitable for a different range of power output in commercial applications. In general, volumetric machines are suitable for 50 kWe and below but turbomachines are more suitable for power outputs higher than 50 kWe.

Keywords: turbomachines, volumetric expanders, organic Rankine cycle, expansion machines, isentropic efficiency

1. Introduction

Organic Rankine (ORC) cycle-based systems have gained popularity in the last 2 decades for heat to power conversion in various applications. In comparison with the traditional Rankine cycle, the ORC-based power systems allow the flexibility to choose working fluids and expansion machines, as an additional degree of freedom, allowing optimal configurations both from the thermodynamic as well as techno-economic aspects. ORC systems can be designed to optimally convert waste heat from internal combustion engine (ICE) exhaust, geothermal heat sources, biomass applications, solar thermal applications, and other thermal gradient-based sources like ocean thermal energy conversion (OTEC). The maintenance-free automated operation and relatively smaller installation and operational costs compared to Rankine cycles make them ideal for commercial use specifically in <10 MW-scale applications.

ORC power systems generally comprise four major components, namely: evaporator, condenser, pump, and expander. The evaporator and condenser are primarily heat exchangers, appropriately sized for a certain duty to operate with specified fluids at specific operating conditions. The current state of the art can be considered sufficient enough to support the technological requirements to ensure the availability of heat exchangers to be used as evaporators and condensers. The pumps have also been well developed and can be bought off the shelf to fulfil the requirements of the ORC system. However, the expander can be considered as the most technically advanced component of an ORC system. The expander is the machine which extracts the energy from the expansion of high-pressure vapour resulting in low pressure while passing through its inlet to the outlet port and converts fluid energy to mechanical power (rotational or reciprocating), which is then often converted to electrical power via direct or indirect coupling to a generator. Organic Rankine cycles, in general, have low thermodynamic efficiency due to limited temperature differences between the heat source and heat sink streams. Therefore, the efficiency of the overall cycle is highly sensitive to the efficiency of the expansion machine [1]. Therefore, the selection of an appropriate expander for a certain ORC application is of great importance to avoid further efficiency reductions and for commercial viability. Depending on the application, operating conditions (temperature, pressure, and mass flow rate), working fluid, and power levels, different types of expansion machines can be used.

2. Primary classification of expanders

In general, the expansion machines are classified based on the nature of their operation. They are broadly classified as either turbo-machine or volumetric-type machines. The turbo-machines in this case refer to turbines of the dynamic or velocity type. They convert the dynamic pressure or high-velocity fluid momentum into mechanical energy while passing through a series of blades. The leaving fluid has generally low pressure and an overall enthalpy drop occurs while passing through machines. Turbomachines are more commonly used for medium to large-scale applications and are well known for their higher efficiency. For smaller power output (<50 kWe), volumetric machines are frequently the preferred choice.

The volumetric-type machines are also known as positive displacement machines. They operate on a principle of force application on a movable mechanical component to extract power. The pressurized fluid is introduced into a chamber and the chamber volume is increased as a net force is applied by a compressed fluid. When the chamber reaches its maximum expansion volume, the low-pressure fluid is released out of expander. The volumetric machines are often equipped with valves to control the inlet and outlet flow of fluid and synchronization with expanding chamber. The volumetric expanders are suitable for smaller power output and often derived from heating, ventilation, air-conditioning and cooling (HVAC) compressors modified to operate in reverse. Both turbomachines and volumetric expanders have their own advantages and disadvantages along with several types available for each main category.

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2.1. Turbo-expanders

In the turbo-expander operation, a high-pressure fluid is directed from the evaporator outlet to the turbine inlet, where the high static pressure of the fluid is converted into high-flow velocity when it passes through nozzles. The high-velocity fluid then transfers its momentum to an array of moving blades, while passing through them. The moving blades are attached to a shaft which is connected to a generator to convert the mechanical energy into electrical energy.

Turbines used in ORC application are generally different from expansion machines used for air, steam, and other gases because, in steam cycles, the enthalpy drop is much higher than that in ORCs. Thus, fewer turbine stages are required in ORCs; therefore, cheaper and lighter turbines are the result. However, the dense vapour properties vary largely from ideal gas behaviour and the speed of sound is much lower than in lighter gases or steam, which influences the nozzle design [2]. The low speed of sound in dense molecule fluids often causes turbines to operate in transonic and supersonic modes. As a result, a highly dissipative system of shockwaves is common in these machines which complicate the design and sacrifice performance specifically during off-design operations [3].

Turbo-expanders have two main categories: axial turbines and radial turbines, as differentiated in **Figure 1** (adapted from [4]). The main difference between the two categories is the way the working fluid flows in relation to the shaft. In axial turbines, the flow of the working fluid is parallel to the shaft, whereas, in radial turbines, it is radial to the shaft at the inlet converting to axial at the outlet of the turbine.

Selection of the suitable turbine (axial or radial) depends mainly on the operating conditions and corresponding enthalpy drop required. At low mass flow rates, the blades of the axial turbine become very small which results in a significant efficiency drop due to the difficulty of maintaining small tip clearance between the blades and the casing. Therefore, axial turbines are always preferred in a large-scale application where the mass flow rate is high and pressure ratio is small. In contrast, radial turbines are employed with applications of low mass

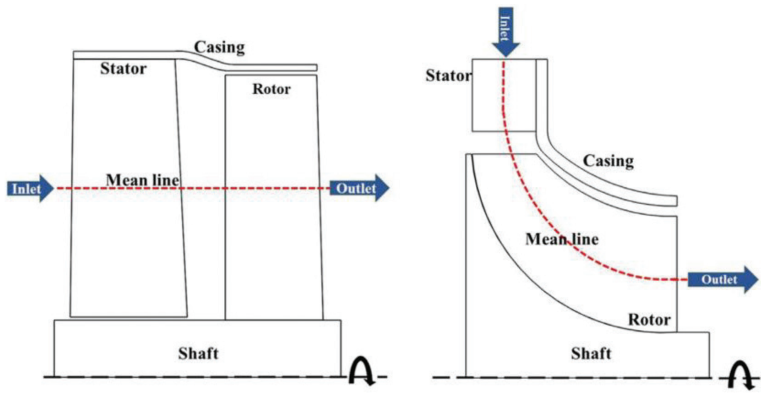

Figure 1. Schematic of axial flow (left) and radial inflow (right) turbines.

flow rate and high-pressure ratios such as turbochargers and ORC systems. For small flow rates, the radial turbines present more efficient performance due to their lower sensitivity to the blade profile than the axial ones. In addition, ORC applications employ high-density fluids which necessitate a more robust turbine due to the increased blade loading. In such conditions, radial turbines are favourable as the blades are rigidly attached to the hub. Due to the radius reduction from rotor inlet to the exit, radial turbines can handle a single-stage expansion ratio of 9:1, while axial turbines require at least two stages. Therefore, radial turbines are also favourable when the system size is taken into account.

However, generally speaking, axial turbines offer better performance at off-design conditions. Moreover, axial turbines present higher efficiency than radial turbines in large-scale applications such large gas turbines, due to the elimination of the flow turning in the meridional plane. In addition, the disc of the axial turbine is protected at high temperatures since only the blades are exposed to the heat. In radial turbines, on the other hand, both the blades and the disc are exposed to the heat since the expansion takes place at both inducer and exducer of the impeller. However, it is worth mentioning that ORC turbines operate usually at low temperatures where concerns about high temperature are significantly lower compared to other applications. The selection of the optimum configuration had been often related to two dimensionless parameters, that is, specific speed and specific diameter, which are based on the volumetric flow and enthalpy drop. **Figure 2** (adapted from [5] commonly known as Balje diagram) presents the selection map and suggests the use of axial flow machines for large specific speeds which correspond to larger flow rate. However, these diagrams should be used with caution as they were essentially developed for incompressible flow. Despite limitations, they are useful to provide initial information which can be further cross-checked by high fidelity models at later stages.

The radial turbines generally involved radial inflow configuration but recent advances in turbine development have also made the use of radial outflow turbines available in ORC technology which is discussed separately, below.

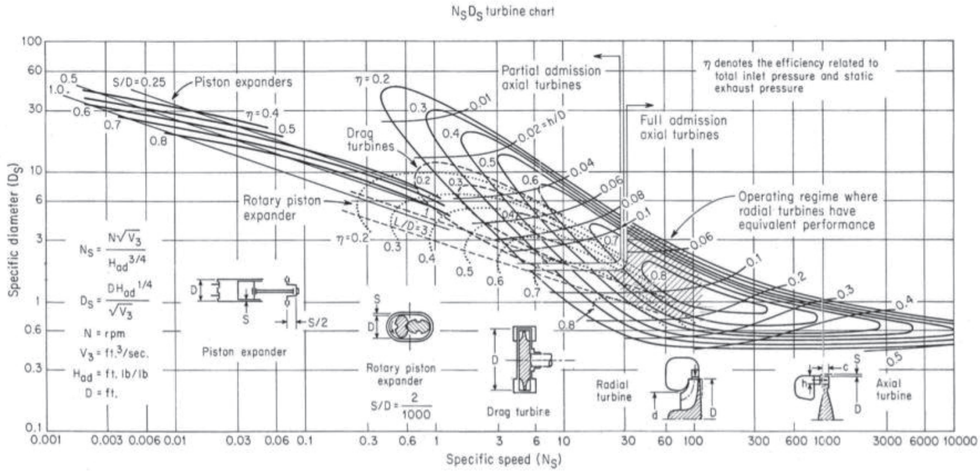

Figure 2. Turbines selection maps based on performance with respect to specific speed and specific diameter.

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2.1.1. Radial inflow turbines

Figure 3 presents the meridional view and overall architecture of the turbine stage in a radial inflow turbine (RIT), which are sometimes also referred to as inward flow radial (IFR 90). As it can be observed, the high-pressure fluid enters the casing (volute) inlet and initial flow direction is primarily radial which is converted to the tangential direction circumferentially at the rotor inlet stage where it, also, contains both radial and tangential components. While passing through the rotor the flow loses its tangential component. The leaving flow must have minimum swirl flow at rotor exit. Furthermore, the direction of flow is converted from radial at the inlet to axial at rotor exit where ideally, there is no radial component. The geometric parameters mentioned in **Figure 3** are obtained from the design process and they are defined as follows: r_1 as volute inlet radius, r_2 as stator inlet radius, r_3 as stator exit radius, r_4 as rotor inlet radius, r_{5t} as rotor exit tip radius, r_{5h} as rotor exit hub radius, b_2 as nozzle inlet diameter, b_3 as nozzle exit diameter, D_4 as rotor inlet tip diameter and ξ as clearance.

The design procedure of radial inflow turbines involves, in simplified form, the steps illustrated in **Figure 4**, which presents a typical path followed. Although there is no single correct procedure, various designers use their own, bespoke techniques.

The design process of turbines (not only limited to the radial machines) often utilize the concept of mean-line flow which provides a preliminary or baseline one-dimensional design. The techniques assume the properties and parameters to be lumped and focus mainly on the inlet and outlet of cascades. The flow is assumed to be uniform and unidirectional and estimation at the centre line of flow can provide average flow characteristics of the fluid. The outcomes of a mean-line model are linked with blade design often based on experiences from NACA databases and statistical models. The overall design is transformed to a computer-aided design (CAD) model which can be used for three-dimensional computational fluid dynamic (CFD) analysis based on Reynolds average Navier-Stokes (RANS) calculations. The resulting analysis helps in the final tweaks of the fluid dynamic design and to achieve optimum performance. The detailed design and optimization techniques can be found in [6–8] for understanding the radial turbine performance. The authors in [4, 8] compiled a list of advantages of using radial turbines specifically for small-scale units over axial machines listed below:

1. Radial inflow machines are often manufactured as single piece cast or forged whereas axial machines often require separate blade and rotor manufacturing.
2. Single-piece rotors are more robust, stiff, and have enhanced rotor-dynamic stability which can help to reduce the overall cost.
3. RITs can offer better off-design performance when variable geometry nozzles are used.
4. Downsizing the axial machines for small-scale ORC applications requires blades to be very small and numerous, which increases the wetted area and frictional losses and blade blockage effects.
5. The running clearance necessary between rotor tip and casing becomes a significant fraction of the blade height which means higher proportionate leakage losses.
6. RITs support larger pressure ratios in a single stage (up to 10 is common for RIT but the axial might need three stages).

2.1.2. Radial outflow turbines

The flow direction in a radial outflow turbine (ROT) is opposite to that of the radial inflow machine. The flow enters the ROT at the centre, near the axis of rotation, axially, and then it travels outwards in the radial direction while passing through arrays of rotor and stator blades. **Figure 5**, adapted from the works of [9], presents the schematic of radial outflow turbines which are also known as centrifugal turbines.

The low speed of sound in organic fluids requires supersonic or at least transonic flows which lead to losses due to shock formation and interaction. The large volumetric expansion of organic fluids requires larger areas at the exit of turbines for the reduction in losses. The ROTs can have the inherent feature of enlargement in the area as the flow moves in a radial direction which means supersonic flows can be avoided and losses can be reduced to end up with high-efficiency turbines.

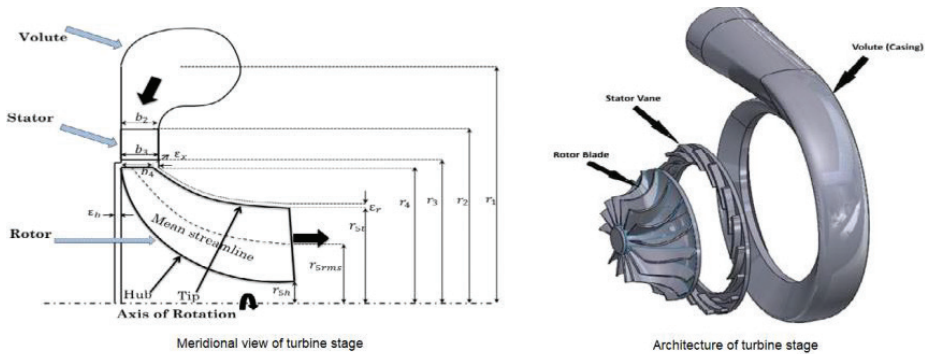

Figure 3. Meridional view (left) and architecture of turbine stage (right).

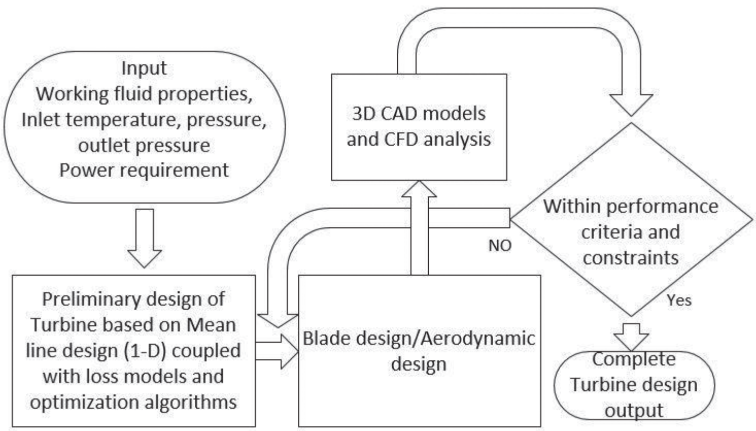

Figure 4. The fluid dynamic design process of turbine design.

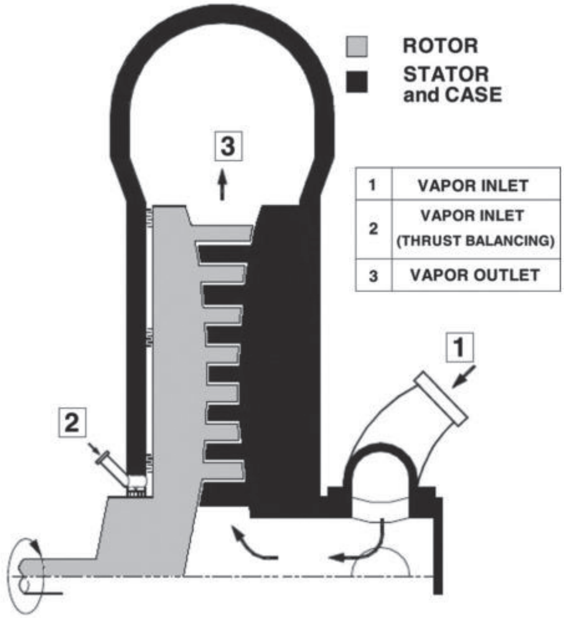

Figure 5. Schematic of radial outflow turbine.

The ROTs allow the adoption of multi-stator-rotor ring arrangements in the radial direction maintaining low peripheral speeds, resulting in low mechanical stresses, lower bearing losses, and simple connections of generator and grid. Furthermore, full-admission inlet stages can be adapted. The simplicity of multistage assembly allows tighter tolerances and thus losses can be reduced. The detailed design and analysis of ROTs are presented in [9, 10].

The disadvantages of ROTs include slightly lower efficiencies compared to RITs as large a surface area is in contact with fluid during flow. Furthermore, for heavy-/large-molecule working fluids, the first stage often has insufficient flow passage area due to the inherent square root proportionality of radius to area, therefore, limiting the turbine application for high-temperature applications. ROTs are more suitable for small-scale applications compared to micro-/mini-ORC applications.

2.1.3. Axial turbines

The axial turbines are characterized by the primary flow of the working fluid which is in axial direction and parallel to the rotational axis. Axial machines are more suitable for larger flow rates, which mean having larger specific speeds as per the Balje diagram (**Figure 2**). In ORC technology, these machines are often suitable for medium-to-large power outputs in single or multistage configurations ranging from one to five. The isentropic efficiencies of axial machines in nominal operations range from 80 to 90% [2]. The axial machines are the most commonly used turbomachine for power production, approximately 70% of power generated is based on these machines as the preferred expander type of large power units. One of the limitations of axial machines is that considering large-stage expansion ratios, the axial channel undergoes span-wise enlargements with a negative impact on performance. Furthermore, highly supersonic conditions may be found at stator exit and converging-diverging nozzle arrangement, which may not be conducive for good off-design performance. Despite their few limitations, axial machines are adapted in large-scale applications in power plants using steam or Brayton cycles and they are also popular in nuclear power applications along with megawatt-scale power output in ORC applications.

Axial machines are adaptable and a list is populated by [11] to highlight the reasons for their flexibility.

1. Pressure can be as high as 300 bar (supercritical cycle) or too low (few hundredths of a bar, last stages of the steam cycle).
2. Overall pressure ratio could be as high as several thousand or as low as 1.0002 in wind turbines.
3. The diameters could be ranging from few centimetres to 100 m in wind turbine applications.

The design and performance evaluation of axial machines is performed in a similar manner as for radial machines and details of which can be found in works of [11]. However, the simple correlations of efficiency prediction for gas and steam turbines are not strictly valid for ORC applications.

2.2. Volumetric expanders

Volumetric expanders can be classified into four main categories: scroll, screw, piston, and rotary vane. Unlike turbo-expanders where the fluid movement is continuous, in volumetric expanders, it is cyclic. An inherent characteristic of this type of expanders is the fixed volume expansion ratio. They operate by trapping a fixed volume of the fluid and displacing this volume into the discharge of the machine, resulting in mechanical work due to the pressure drop. Therefore, they are also called displacement expanders.

Contrary to turbines, some volumetric expanders may have valves at inlet and outlet ports. The compressed fluid is fed into a chamber and inlet valves are closed, the expansion process starts, and at the end of the expansion, the outlet valve is opened to release the low-pressure fluid. These might be useful to control the timing and flow through an expander but it incurs significant losses. Piston-type expanders often have valves, scroll machines may also have these valves, but in general, screw, scroll, and vane type-expanders operate without valves.

Another peculiarity of volumetric expanders is related to their lubrication requirement. As they operate on the principle of changing volumetric capacity, there must exist some parts which are moving in contact with other surfaces to increase the volume for expansion. The contact movement adds friction which increases the wear, tear, and heat of the surfaces. Lubricant oil is often circulated especially in scroll- and screw-type expanders, which reduce the friction and also help seal the clearances and reduce leakage losses. The lubricant oils used

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are often soluble in working fluid and can be circulated in the complete ORC cycle or only in the expander by means of separation mechanisms where oil is removed and re-circulated in the expander. The oil separation systems incur the cost of extra equipment and add to system complexity. The complete circulation might have detrimental effects on heat exchanger performance. In order to mitigate these issues, working fluids with good lubricant properties are preferred. Oil-free expanders have also been developed and the lubrication for their bearings is done by the application of grease [12].

Figure 6 adapted from [13] presents the under- and over-expansion losses. The volumetric machines have fixed volumetric ratio, so the thermodynamic cycle must be designed for optimum expansion ratio. It might be possible that a higher pressure ratio may theoretically lead to higher efficiency but the over-expansion losses will limit the overall performance so the close match between cycle and machine expansion ratio must exist. In recent years, volumetric expanders have received a great deal of attention in small-scale systems due to their good off-design performance.

2.2.1. Scroll expanders

Scroll expanders consist of two spirals: an orbiting scroll and fixed scroll as presented in **Figure 7** adapted from [14]. The orbiting scroll moves along with the fixed scroll within tight tolerances. The working fluid moves in from the centre and moves outwards inside the chamber between the orbiting and moving scroll. They are widely used as they can be derived from a scroll compressor, thereby, reducing the machine cost. Scroll expander can be either compliant or constrained. In the former, a lubrication system is required in order to reduce the friction between the contacting sidewalls. In the latter, lubrication is not required due to the existence of the linking mechanism between the rotating and fixed scrolls. In addition, there is no need for exhaust valves which results in noise reduction.

Scroll expanders usually operate at low power output applications (<10 kWe) due to their limited speed. In addition, they are preferred in small-scale applications due to their low parts count, which reduces the level of noise, increases the reliability, and makes them more cost

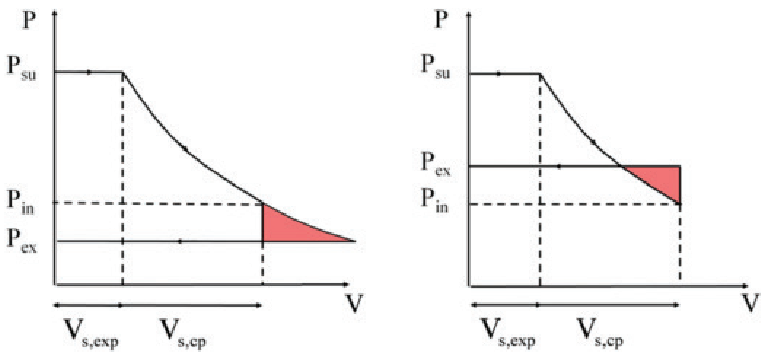

Figure 6. The under expansion (left) and over expansion (right) losses.

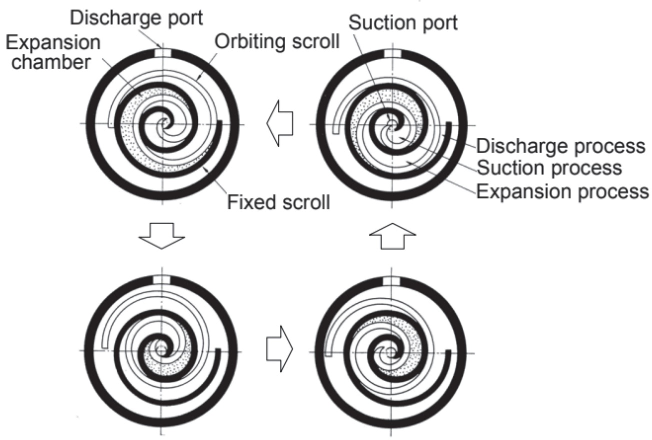

Figure 7. Operation of the scroll expander.

effective. Scroll expanders have a volumetric ratio between 1.5 and 5 and maximum power output reported of 12 kW [15]. Moreover, scroll expanders can have high efficiency as 80% at different operating conditions. Furthermore, the off-design operation of scroll machines was presented in the works of [16] and **Figure 8** adapted from [15] presents the accounting of losses of scroll machine when operated at various pressure ratios.

The results suggest that the highest efficiency is achieved when the scroll machine is operating at a built-in pressure ratio of 4–7. Furthermore, it can be inferred that decline in performance is more rapid for underexpansion when compared with overexpansion. This suggests that a slightly oversized machine will be a better choice for varying load applications. The primary operation range for scroll machines is from 0.5 kWe to 10 kWe output power range. Scroll machines are primarily derived from refrigeration and HVAC compressor units. They could be in various sealing conditions, for instance, hermetic, semi-hermetic, and open-drive configurations. Hermetic-type machines contain electric machines sealed in a single casing along with the compressor/expander. The working fluids may come in contact with electric coils and help to cool down the electrical systems. The machine is not supposed to be opened for services. The semi-hermetic configuration allows the machine to be dismantled for servicing and open-drive systems have generators/motors completely separate from the expander/compressor. The generator/motor is connected to expansion machine by a belt or magnetic coupling allowing the sealing to be limited to expansion machine components only. Technological enhancements to increase the volumetric ratio are being pursued; one of the ways to increase the built-in volume ratio is by utilizing variable thickness walls. However, no such commercial unit is available yet. The operating speeds in general for scroll machines are around 3600 RPM so the generators can be directly coupled to them.

2.2.2. Screw expanders

Screw expanders are composed of two helical rotors designed with an accurate profile to trap the required amount of the working fluid. **Figure 9** (adapted from [17]) presents a schematic

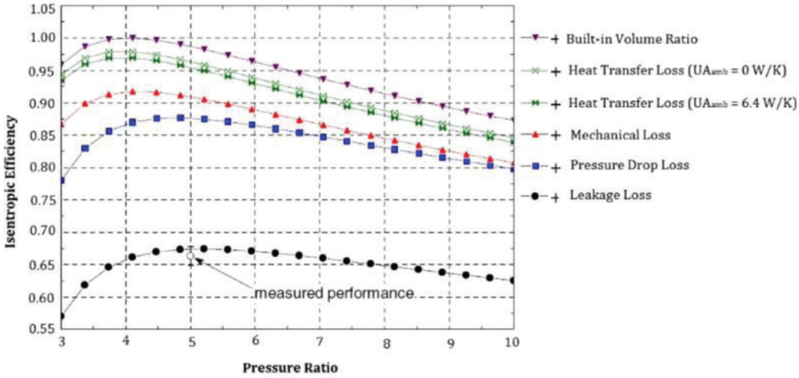

Figure 8. Variation in efficiency as a function of pressure ratio.

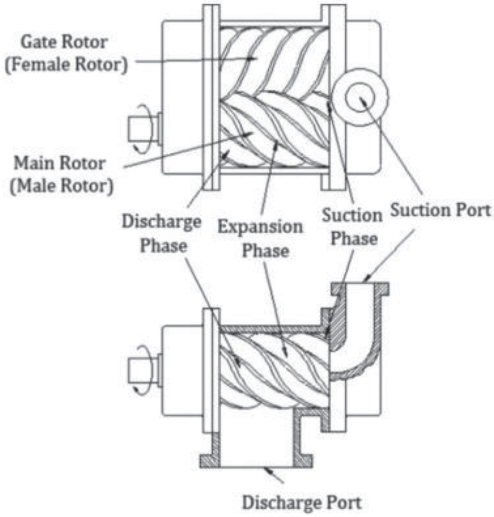

Figure 9. Schematic of a twin screw expander.

of twin screw expander. The synchronized movement of intermeshing rotors generates volume profiles that originate at one end of the rotor and terminate at another end. The working fluid is expanded in that meshed chamber. Screw expanders can be applied in systems with power outputs up to 1 MW. Lubrication is required in screw expanders due to the direct contact between the rotors. However, lubrication can be omitted if a fluid with lubrication specification is adopted. Like scroll expanders, screw expanders can operate with wet working fluids since they can accept large mass fractions.

The rotor clearance is below 50 μm so the leakage losses are comparatively small, thus reducing friction losses. Screw machines exhibit in general medium levels of noise and high costs.

The volumetric ratio can be from 2 to 8. Expander power output ranges from 1.5 kW up to 1 MW. The isentropic efficiencies have been reported to be as high as 70% [15]. They can operate at higher RPM configurations than scroll machines, and a gearbox may be required if the machine is operating above 5000 RPM, which is not uncommon for screw machines. In general screw, machines are suitable for power applications from 5 to 50 kW in ORC applications. **Figure 10** (adapted from [18]) presents the selection maps of working fluids for operation with screw machines based on evaporation and condensing temperatures.

2.2.3. Piston expanders

The working fluid enters the piston expander when the piston is around the top dead centre (TDC) and the inlet valve is then closed. The fluid expands as the piston is pushed by the internal pressure; the energy is transferred to the central crankshaft by connecting rod. The exit valve is opened up at bottom dead centre and expanded working fluid starts moving out of the chamber as the piston moves back to TDC as shown in **Figure 11**.

The piston expanders can have a single piston or multiple piston-cylinder arrangements. The designs are also not limited to the piston-connecting rod and crank-based systems. Linear piston expanders are gaining popularity where a single piston may oscillate in a cylinder and operate in two volume chambers at opposite ends. Apart from the aforementioned, axial configuration, rolling pistons, and swash plates are some of the different types of piston expanders.

Pistons expanders are known to have lower isentropic efficiencies compared to corresponding turbomachines, for example. The maximum reported efficiency is 76% and the average is reported to be around 50% [15]. Piston expanders are characterized by relatively large pressure ratios of 6–14. Due to the low power outputs, such types of volumetric expanders are preferred in small- and micro-scale applications. In general, the output of the expander around 2 kW is reported but one of the works has reported 18.6 kW with steam as the working fluid [15]. Unlike the previous categories of volumetric expanders, piston expanders are adopted with inlet and discharge valves to control the suction and discharge processes. However, for the latter process, exhaust ports can be applied instead of valves which lead to larger work and

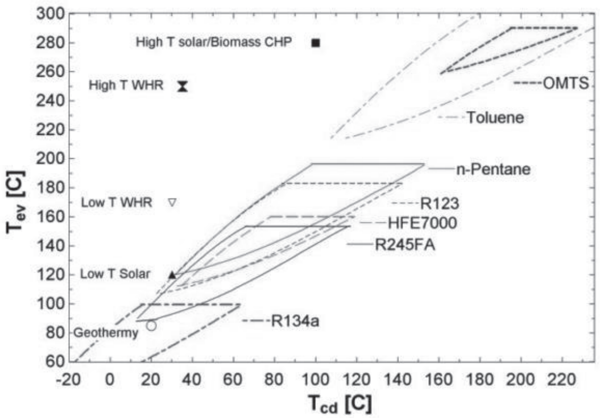

Figure 10. Selection maps for screw expander application.

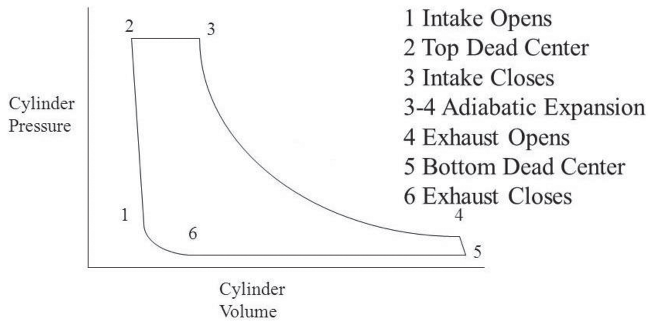

Figure 11. Pressure-volume diagram of piston expander.

lower mass flow rates. Piston expanders can operate under two-phase conditions of the working fluid. However, they are heavy and suffer from noise and vibration. However, like some volumetric expanders, lubrication is required in piston expanders but entails the difficulty of implementation because the oil should be mixed with the working fluid, which reduces the efficiency of the cycle. The piston expanders mainly suffer from the requirement for weight balancing, torque impulse, heavy weight, precise valve operation, and a large number of parts [4] but they have mature manufacturing technology available.

2.2.4. Rotary vane expanders

Rotary vane expanders are operated based on Wankel concept. **Figure 12** adapted from [19] presents the working of a vane expander. Working fluid enters the expander at the location having small clearance. A rotor with moveable vanes is attached to a rotor which is in close proximity to the casing in asymmetric orientation. The rotation of the rotor allows the vanes to move outwards while trapping working fluid, as the rotation angle increases the volume bound by consecutive vanes increases and expansion of working fluid occurs.

Their reported power output ranges from few watts to 2.2 kW. As some volumetric expanders, they can be directly attached to the generator due to their low rotational speeds. They are usually preferred to reduce the system costs because of their simple design and low manufacturing costs, higher torque, and higher volumetric efficiency. In addition, they are mechanically simple and available commercially. In addition, they are characterized by small vibration, low acoustic impact, and simple and reliable structure. However, they exhibit lower isentropic efficiencies compared to other volumetric expanders due to leakages and higher friction losses. Furthermore, the machine must be lubricated to minimize wear and enhance sealing. **Figure 13** presents a picture of actual vane expander and the size can be estimated from the figure. Despite their popularity for micro-scale applications, they have certain technical limitations. The volumetric ratio is limited with commonly reported values of 3–7. The maximum temperature at the inlet is also limited to around 140°C. To ensure the vane movement in the groove and to still minimize the leakage, a tight tolerance is maintained. If a very high-temperature working fluid is passing through it, the expansion of vanes might cause them to stick and the machine will cease its operation. This limits the use of vane expanders for high-temperature applications [12].

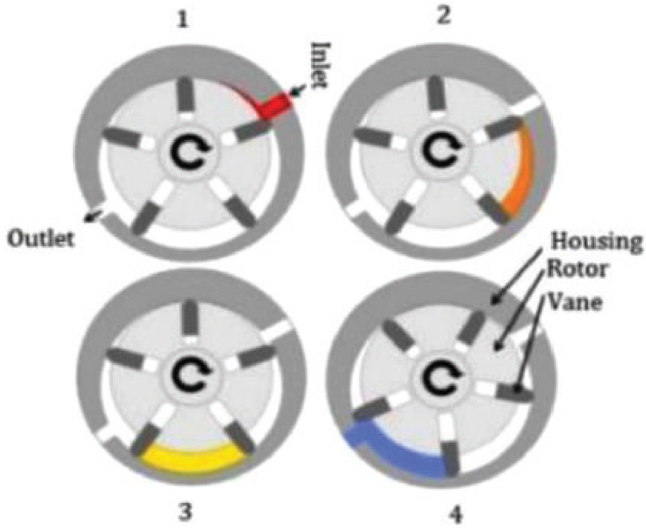

Figure 12. Operation of a rotary vane expander.

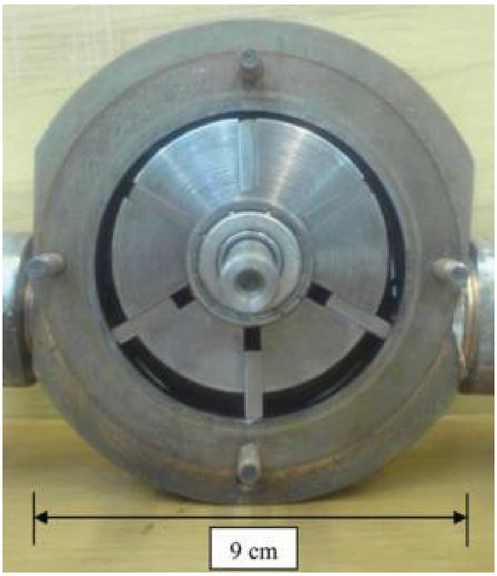

Figure 13. Photograph of the actual vane expander from the works of [20].

3. Multiphase expansion capability

Working fluids with a positive temperature-entropy slope or at least isentropic expansion capability are preferred for organic Rankine cycle applications to avoid the formation of liquid droplets during the expansion process which can lead to erosion problems. ORC systems utilizing variable heat source conditions (flow rate or temperature) are specifically prone to two-phase fluid admittance in expansion machines. Although the degree of superheating is continuously monitored by control systems, under severe fluctuating heat source conditions, which are typical for an automobile engine operating in an urban driving cycle, the tight control of the degree of superheating becomes difficult even for state-of-the-art control schemes. The influx of the liquid phase becomes unavoidable in such cases. Furthermore, the trilateral flash cycles are considered thermodynamically beneficial [21, 22] but they need expansion machines which are capable of handling multiphase fluids.

The turbo-expanders have very limited capability to handle multiphase flows. Major erosion problems arise when liquid droplets strike the rotor blades. Thin liquid films may form on the stator and then larger droplets are shed which accelerate along the high-velocity gas molecules towards the rotor blades. The velocity of droplets is not immensely high due to their large inertia but when fast-moving rotor blades are hit the change in momentum can cause tremendous localized forces and cause the erosion of rotor blades specifically at leading edges and near the tip due to higher tangential velocities [23]. Some coatings can slow down the erosion process but increase the expander cost and complexity. Water drainage systems and blade heating systems to prevent liquid flow are very difficult to implement for small-scale systems.

Volumetric machines are generally more tolerant to admission liquid phase primarily due to lower component velocities involved. Scroll and screw machines are well known for the capability of flooded expansion. Recent research [24] compared the experimental results of a 2 kW-class scroll machine for superheated expansion and flooded expansion. The results indicate that system can run smoothly for flooded expansion and even at conditions where both inlet and outlet of the expander have two-phase conditions. The net power output was observed to be lower as the cycle work output is reduced thermodynamically. However, it was not possible to report the isentropic efficiency in the multiphase expansion regime as instrumentation capability was limited. Furthermore, a significant quantity of oil was used to test the liquid flooded expansion in the scroll machine in the works of [25] and only a modest decrease in performance was observed. It was also discovered that liquid injection can reduce the leakage losses in scroll machines and can be a promising solution for trilateral flash-cycle expanders for the smaller scale [26].

Screw expanders are particularly popular for their potential use in trilateral flash cycles. Screw machines have been tested with flooded expansion in various studies. Up to 0.1 mass fraction of liquid oil was used to test the performance of the expander in [27] and it was reported that the presence of liquid oil helps to seal the clearance volumes at lower operational speeds and as throttling losses become dominant at higher rotational speeds. The authors concluded that further work with low-viscosity working fluids and impact of gases and liquid working fluid along with lubricants needs to be pursued. The authors in [28] compared dry expansion and water-flooded expansion and concluded that water-flooded machines are preferable to dry-running machines for all circumferential speeds but oil-flooded expansion was beneficial for lower speeds. It was also proposed in the literature [29] that screw expanders can be built no larger than current gas compressors to work as two-phase expanders with far higher efficiencies than were believed to be possible for trilateral flash cycles. If the liquid-flooded working fluid can also act as the lubricant instead of oil, the expander and cycle performance can increase substantially in organic Rankine cycle trilateral flash cycles [30]. The multiphase expansion capability of reciprocating expanders still needs to be investigated in detail with a focus on the throttling losses through the inlet valves.

4. Conclusion

It is evident that no expander technology is perfect; each type of machine has its own advantages and disadvantages. Based on the literature survey and experiences in commercial applications of ORCs, it is clear that efficiency and cost are a function of power output. In general, efficiency increases and cost reduces per kW as the power output is increased. In the context of the presented text in the chapter, it can be concluded that in general volumetric machines are suitable for smaller power output systems. Vane-type expanders are suitable for power outputs below 1.5 kWe, and scroll machines are suitable in the range of 1–5 kWe. Screw machines are suitable for the range of 5–50 kWe. Piston expanders are suitable for larger pressure ratios between 5 and 20 kWe. Turbomachines are suitable for larger-scale systems or if the cost of manufacturing is reduced then they can replace the volumetric machine with their single-stage expansion capability and high comparative efficiencies. In general, radial inflow machines are suitable from 30 to 500 kWe; recent advances in the industry have also successfully implemented radial machines in the MWe range. Axial machines are, however, the dominant type in the MW power range. Radial outflow machines appear to be competitive in the small-scale range below 50 kWe where radial inflow machines require higher rotational speeds.

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