Small Scale Organic Rankine Cycle For Solar Applications

H.J. Lombard*[1], R.T. Dobson[1]

Department of Mechanical and Mechatronic Engineering, University of Stellenbosch, Private Bag X1, Matieland, Stellenbosch, 7602, South Africa

* Corresponding Author: rtd@sun.ac.za

Abstract

The global demand for energy has risen dramatically in recent years and the cost of electricity has risen considerably. Local electricity sales prices followed the global trend, increasing by an average of 25% per annum from 2008 to 2014. This upward trend has opened the market for new technologies, such as renewable energy alternatives, better use of waste streams and low temperature power generation.

The Organic Rankine Cycle is an example of a low temperature power generation cycle that can utilize a renewably energy source or recover waste heat. The Organic Rankine Cycle is in essence a Rankine Cycle that employs a different working fluid than water. The primary objective was to build a functioning ORC. The cycle utilized a low temperature (< 120°C) and pressure heat source to generate electricity. This temperature is easy achieved using solar collectors which will make the system suitable to operate using solar radiation as a heat source. To meet the temperature, pressure and global warming potential requirements, refrigerant R123 is used as a working fluid.

The system consisted of a rotary vane pump, some plate heat exchangers, a scroll expander and a natural convection condenser. The scroll expander was a modified Copeland scroll compressor. The system demonstrated a cycle efficiency of 13% compared to the Carnot cycle efficiency of 25% and a modified Carnot efficiency of 14%. Due to excessive heat loss and poor conversion from mechanical to electrical energy the electrical output was 40 W. The conversion efficiency could be greatly improved if the electric generator was changed to a DC generator.

Keywords: Organic Rankine Cycle, Scroll expander, Working fluid

1. Introduction

Many sources of energy for electricity generation exist ranging from gas and coal to renewable alternatives like solar and wind. The most common energy source in South Africa is coal which accounts for about 90% of electricity generated (Eskom Holdings SOC Limited, 2012). While coal is inexpensive it has two major drawbacks. Firstly the supply of coal is finite, which means that ultimately this fuel supply will have been depleted. Furthermore, global warming and environmental agencies apply increasingly more pressure to limit the expulsion of CO_2 into the atmosphere which requires even more sophisticated and more expensive sequestration technologies (Eskom Holdings SOC Limited, 2012).

South African electricity sales prices followed the global trend, increasing by as much as 30% in 2009 (Eskom Holdings SOC Limited, 2012). This upward trend has opened the market for new technologies such as renewable energy alternatives, better use of waste streams and low temperature power generation. Popular renewable energy technologies are photovoltaic cells, Stirling engines, wind turbines and organic Rankine cycle (ORC) systems.

An ORC is an example of a low temperature power generation cycle that can utilize a renewable energy source or recover waste heat. The ORC is in essence a Rankine cycle that employs a different working fluid than water to increase the cycle efficiency at the lower temperatures. The ORC uses high temperature heat to vaporize liquid which drives a turbine connected to a generator. The primary objective of this paper is to show how a simple functioning ORC was built, tested and its thermal performance evaluated. The cycle is to utilize a low temperature (< 120°C) and pressure heat source to generate electricity. The low temperature is readily achievable using existing solar collectors that are currently used as water heaters.

Nomenclature

h	Enthalpy, *J/kg*
l	Liquid
P	Pressure, *bar*
s	Entropy, *J/kgK*
s	*Isentropic exit condition*
T	Temperature, *°C*
v	Vapour

2. Organic Rankine Cycle

2.1 System Description

The system was designed to utilize a low temperature heat source between 80°C and 120°C to generate electricity. The most basic Rankine cycle consists of the following five components: boiler, expander, condenser, working fluid and a pump. In the current ORC a total of nine components are used. The additional components include a super-heater, a feed water heater (FWH), accumulator, oil pump and oil tank where oil is heated. Figure 1 shows a schematic representation of the system.

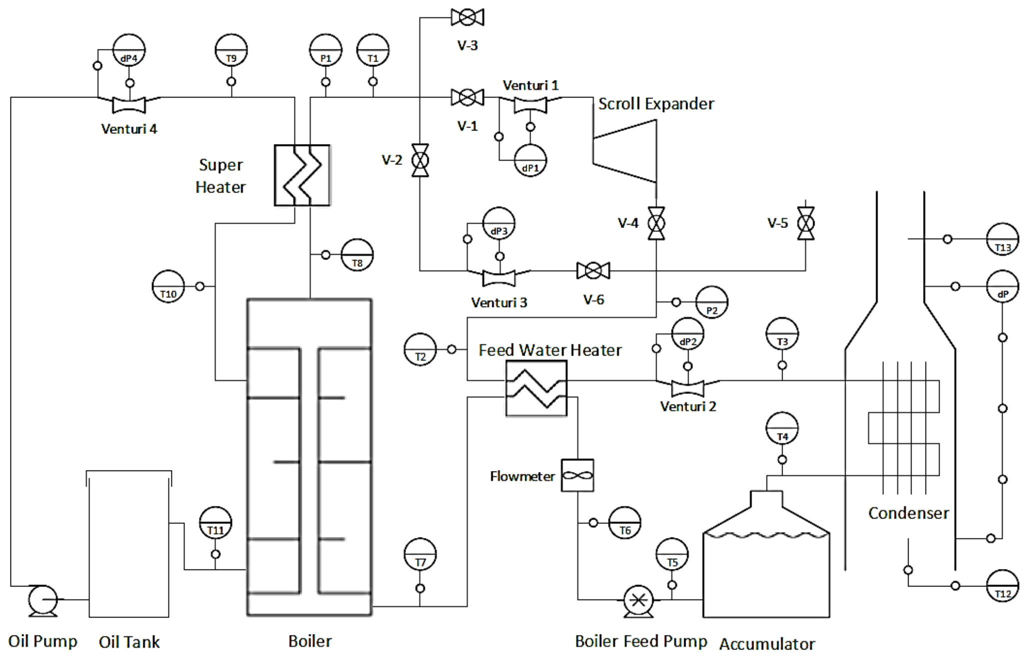

Figure 1: Schematic diagram of the organic Rankine cycle (ORC) system

The working fluid in liquid form is pumped from the accumulator to the FWH by the boiler feed pump. The liquid stream is heated by the vapour stream exiting the turbine in the FWH. Liquid then enters the boiler where thermal energy is transferred from the oil to the refrigerant. The outlet stream of the boiler then passes through the super-heater where additional thermal energy is transferred from the oil to the refrigerant to ensure that all the working fluid has changed from liquid to vapour before entering the expander.

The vapour stream then passes through the scroll expander where energy is extracted from the fluid and converted to mechanical energy. The mechanical energy is then converted to electrical energy by the electric motor attached to the scroll expander. A bypass valve (V-2) is installed to divert flow away from the scroll expander if needed or to control the rotational speed of the scroll expander.

The recombined flow then passes through the FWH to pre heat the working fluid in liquid form flowing to the boiler. The vapour then enters the natural convection condenser where the heat is transferred from the working fluid to the air flowing over the fins of the condenser. Liquid leaves the condenser and flows to the accumulator.

2.1.1 Expander Selection

A wide range of expanders exist for power generation (Yamamoto, 2001). Expanders convert vapour at a high temperature and pressure into mechanical work. The prevailing 5 types of expanders are turbines, reciprocating piston, rotary vane, rolling piston and scroll volute.

The most common type of expander is the turbine, although turbines are a proven technology with isentropic efficiencies reaching up to 85% these machines operate at very high speeds and are not commercially available in small scale (Yamamoto, 2001). Turbines do not work well with wet fluids and performance characteristics worsen drastically if they are not operated near the design conditions (Yagoub, et al., 2006).

Reciprocating pistons have been around since 1765 when the first steam engines were built and have since matured as a technology (Encyclopædia Britannica, 2014). One of the major drawbacks of this type of expander is high friction losses (Baek, et al., 2005). They also require significant balancing and valve timing to function properly. Reports of isentropic efficiencies as high as 62% have been published, but these have yet to be employed in ORC systems (Zhang, et al., 2007).

Reciprocating piston rotary vane devices also have high friction losses. Furthermore, their performance suffers due to leakage through the vanes and rotor end faces (Mohd.Tahir, et al., 2010). Rotary vane devices are simple, robust and can handle high pressures. Isentropic efficiencies of up to 48% in ORC systems have been reported but their usage thus far have been limited (Yang, et al., 2009).

Much the same as the rotary vane the rolling piston is a simple, robust design that can handle high pressures (Wang, et al., 2010). Some variations exist that reduce the amount of friction and leakage (Subiantoro & Ooi, 2010). The highest isentropic efficiency reported for rolling piston expanders is 45.2% (Haiqing, et al., 2006).

The scroll expander has been tested extensively in literature with isentropic efficiencies as high as 83% reported (Schuster, et al., 2009). Although the scroll expander has a complex geometry, scroll compressors are widely used and inexpensive to buy and modify (Mathias, et al., 2009). Scroll expanders have also shown good performance when conditions deviate from design conditions (Kim, et al., 2007).

Considering all the attributes of the aforementioned expanders a scroll expanders was chosen. The main favourable factors were the low cost, availability, the ease of modification and the fact that these devises have been extensively tested.

2.1.2 Working Fluid

In utility scale electricity plant water is used as the working fluid. When water is used as working fluid the minimum condenser temperature for a positive pressure is 100°C, but the aim of this project is to generate electricity from a low temperature heat source (80°C to 120°C). Thus a different working fluid is required with a boiling point much lower than that of water (100°C at 1 bar). Such fluids are normally used in the refrigeration and air-conditioning industries and are thus called refrigerants. Refrigerant boiling points vary from about -80°C for R23 to almost 27°C for R123 at 1 bar (NIST, 2013). The lower boiling point allows the system to maintain positive pressures, while operating at lower temperatures.

The properties of suitable working fluids are high densities, low cost, moderate pressures in heat exchangers, low ODP and GWP (Marion, et al., 2012). R11 was eliminated due to the large environmental impact and R134a requires the operation pressure to exceed the maximum allowable pressure of 10 bar. This leaves R123 which has a high density, moderate pressures in the heat exchangers, low ODP and GWP.

2.1.3 System design operating conditions

The design specified operating points are given below in Table 1. For design purposes the isentropic efficiency of the pump was take to be 60% and the isentropic efficiency of the scroll expander was taken to be 50%. The mass flow rate is 0,0833 kg/s at design conditions.

__Table 1: Design operation conditions for ORC__

Point Number	Temperature (K)	Temperature (°C)	Pressure (Bar)	Entropy (J/kg)	Enthalpy (kJ/kg)	Description
1	315.71	42.71	2	1.1463	243.12	*Pump Inlet, Condenser Outlet*
2s	315.5	42.5	8	1.1463	243.22	*Isentropic Pump Outlet*
2	315.5	42.5	8	1.1452	243.29	*Pump Outlet, FWH Cold Inlet*
3	342	69	8	1.2274	270.18	*FWH Cold Outlet, Boiler Inlet*
3,*l*	373.36	100.36	8	1.3278	306.01	*Saturated Liquid*
4	373.36	100.36	8	1.,6863	439.88	*Boiler Outlet, Super-heater Inlet*
5	383	110	8	1.7075	448.11	*Super-heater Outlet, Turbine Inlet*
6s	338	65	2	1.7075	422.86	*Isentropic Turbine Outlet*
6	354.5	81.5	2	1.7411	435.49	*Turbine Outlet, FWH Hot Inlet*
7	321.2	48.2	2	1.667	408.59	*FWH Hot Outlet, Condenser Inlet*
8	315.71	42.71	2	1.1463	243.12	*Condenser Outlet, Pump Inlet*

The design operating points are presented in the form of a *T-s* diagram in Figure 2.

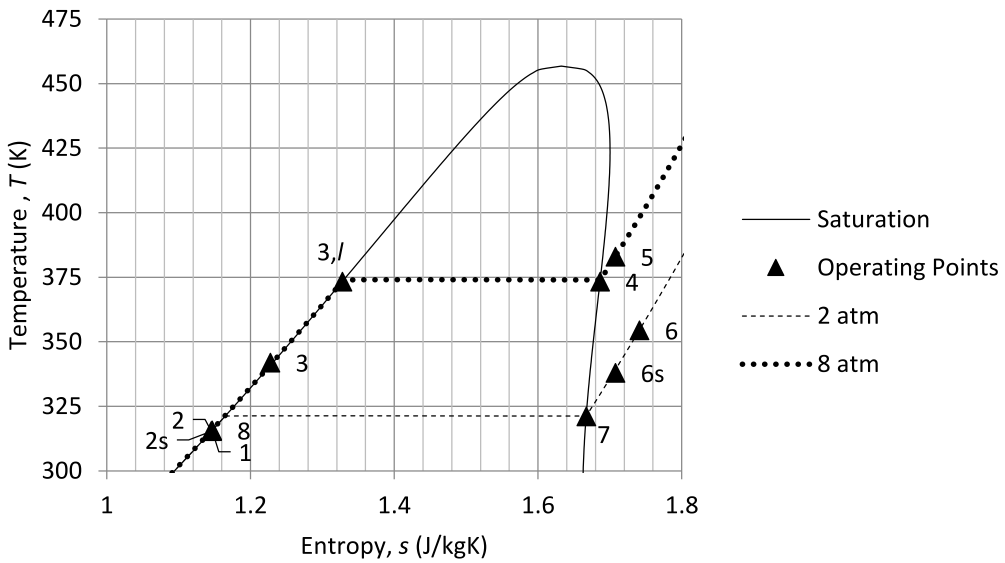

__Figure 2: T-s diagram indication the design operation conditions__

3. Experimental Results

The operating temperatures and pressures are given in Table 2 with the cycle shown on a *T-s* diagram in Figure 3. The scroll expander inlet pressure reached about 5.2 bar absolute and a temperature of almost 113 °C.

Table 2: System operation points at 120 °C

Point Number	Temperature, *T* (K)	Enthalpy, *h (kJ/kg)*	Entropy, *s* (kJ/kg*K)	Pressure, *P* (Bar Absolute)	Description
1	297.216	222.990	1.080	1.895	*Pump Inlet, Condenser Outlet*
2	299.474	226.140	1.090	5.961	*Pump Outlet, FWH Cold Inlet*
2s	297.500	223.080	1.080	5.961	*Isentropic Pump Outlet*
3	339.743	268.510	1.223	5.957	*FWH Cold Outlet, Boiler Inlet*
4	384.172	393.020	1.743	5.372	*Boiler Outlet, Super-heater Inlet*
5	385.947	454.290	1.746	5.186	*Super-heater Outlet, Turbine Inlet*
6	368.300	446.468	1.770	2.302	*Turbine Outlet, FWH Hot Inlet*
6s	356.000	437.055	1.746	2.302	*Isentropic Turbine Outlet*
7	315.219	388.410	1.599	1.992	*FWH Hot Outlet, Condenser Inlet*
8	297.216	222.990	1.080	1.895	*Condenser Outlet, Pump Inlet*

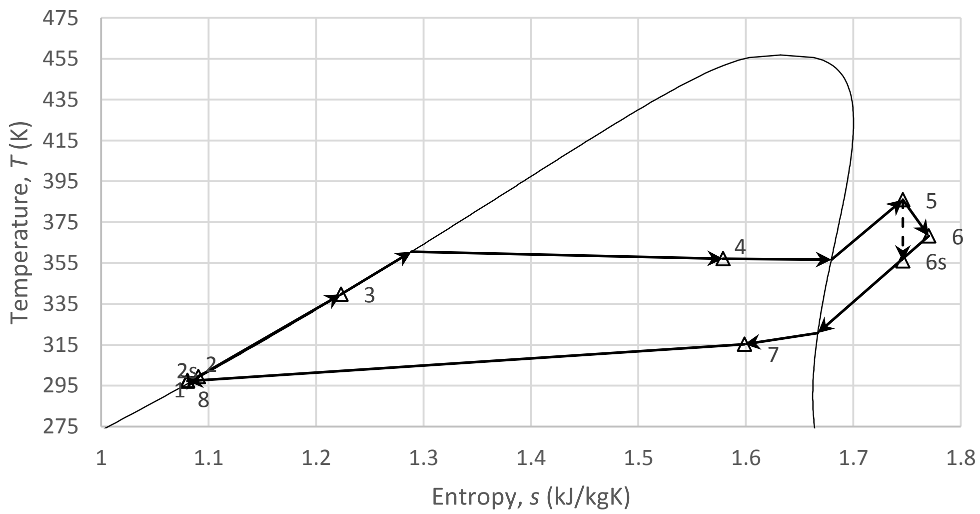

Figure 3: System T-s diagram for operation at 120 °C

The *T-s* diagram indicates the operation of the system in its entirety. Though the experimental *T-s* diagram is similar to the design specification the position of point 4 is expected to be located on the saturation curve. Therefore it is evident that the boiler performance is not satisfactory. The expectation is that point 7 should fall on the saturation curve. The shift indicates that the FWH performance is exceeding initial expectations. The performance of the super-heater and the condenser seem to be satisfactory as points 5 and 8 are where expected. From the *T-s* diagram the isentropic efficiency of the pump appears to be lower than expected.

The typical inlet temperature of the boiler feed pump was 24 °C and the outlet temperature was about 26.3 °C. The change in pressure over the pump was measured to be 7.1 bar at a measured flow rate of 0.067 kg/s. The power consumption of the pump was measured as 240 W. The boiler managed to transfer almost 8.2 kW of thermal energy from the oil to the refrigerant. With the oil inlet at 112 °C and the outlet temperature at 111 °C the mass flow rate was 1.08 kg/s while the liquid R123 entered at a temperature of 66.6 °C. The 8.2 kW was enough to vaporize 75% of the fluid at a saturation temperature of 84 °C.

The super-heater transferred 4.1 kW of thermal energy from the oil to the refrigerant. At the super-heater exit all the liquid had been vaporized and the temperature recorded was 112.8 °C. The oil entered at 113.8 °C while at the super-heater exit the temperature had decreased to 112 °C. The working fluid entered the scroll expander at a temperature of 112.8 °C. At the turbine exit the temperature measured was 75.8 °C. Taking into account the heat loss of 875 W through the housing of scroll expander the energy absorbed by the scroll expander was 566 W and the exit temperature was 95.2 °C. The isentropic efficiency was calculated as 47%. The no-load voltage reached up to 120 V. As the load was applied the current drawn increased to 0.38 A, but the voltage decreased to 104 V. This brings the electric power to 39.5 W.

The FWH recovered 2.8 kW from the refrigerant after it had passed through the expander and heated the liquid refrigerant between the pump and the boiler. The vapour temperature was 75.8 °C at the FWH inlet and 42 °C at the outlet to the condenser while the liquid entered at 26.3 °C and exited at 66.6 °C. It was noted that the vapour started to condense in the FWH. The condenser extracted 11 kW of thermal energy from the working fluid and transferred the energy to the air flowing over the fins. The R123 entered the condenser at a vapour quality of 0.86 and a temperature of 42 °C. At the condenser outlet the vapour quality was zero and the temperature had decreased to 24 °C. The air entered the sheet metal plenum at 17 °C and a relative humidity of 27%. At the outlet of the plenum the air temperature was measured to be 36.9 °C.

4. Conclusions

The pumps performed well, but the O-rings hardened over time causing the pump to start leaking. It is therefore recommended that the O-rings on the pump be replaced with some made of a more suitable material such as Viton.

The boiler heat transfer capacity was considerably less than the design capacity due to the two-phase flow in the narrow passages between the plates. A heat exchanger that is more suitable for the 2-phase flow, with inclined tubes instead of narrow vertical plates, was recommended but the purposefully designed and built unit would have cost triple. There are multiple ways to increase the capacity, three of which are given here:

1. Raising the heating oil temperature would increase the boiler capacity, but this would require a different pump and thermostats.
2. If the oil is changed to water the heat transfer characteristics improve considerably and therefore increases the boiler capacity. However, for water to reach 120°C the oil tank would have to be exchanged for a sealed unit that can handle the saturation pressure of water at 120°C and this unit might be classified as a pressure vessel.
3. The simplest solution to increase the boiler capacity would be to replace the oil with a heat transfer fluid such as Dowtherm. This requires no modifications to the system while the improved heat transfer characteristics would bring about an increase in boiler capacity.

The super-heater performance was satisfactory, ensuring that no liquid entered the scroll expander. It should be noted that any changes made to increase the boiler heat transfer capacity would also increase the super-heater capacity. Since the scroll expander can handle some liquid the super-heater is not a critical component in the experimental system. The electricity output of the scroll expander is very poor, measuring less than 40 W while the expected output was 1 kW. The cause of this poor output is the circular dependence of the magnetic field and the output voltage on one another. A simple way to improve the output would be to replace the run capacitor with a much larger one. This would allow a larger current to be drawn with a smaller voltage drop, improving the power output. A better solution however would be to re-manufacture the scroll expander housing and covert the hermetic scroll expander to a semi-hermetic scroll expander. A magnetic coupling is recommended since it ensures no leaks. The new configuration would allow for the measurement of mechanical work directly on the housing of the electric motor. The electric motor should be replaced by a DC motor or an induction motor with access to the stator winding terminals. If the stator winding terminals can be accessed the stator windings can be excited by a different power source, offering control of the magnetic field and therefore control of the output voltage.

The FWH is not a critical system component, it serves to improve the overall performance of the system. The FWH capacity is slightly higher than expected, reducing the required boiler and condenser capacity even more. If the refrigerant volume flow rate is increased the FWH capacity would also increase and thus this components performance would still be satisfactory. The natural convection condenser performed well according to the design extracting the required heat without any problems. If the volume flow rate of the refrigerant is increased however the outlet temperature of the condenser might rise. A rise in outlet temperature would result in a decrease in system efficiency. However the efficiency could easily be improved by installing a fan in the condenser plenum. This would increase the cooling capacity and lower the outlet temperature and pressure of the condenser, increasing the amount of energy available to the scroll expander and thus increasing the system efficiency.

References

Baek, J., Groll, E. & Lawless, P., 2005. Piston-cylinder work producing expansion device in a transcritical carbon dioxide cycle. Part I: experimental investigation. *International Journal of Refrigeration,* 28(2), pp. 141-151.

Eskom Holdings SOC Limited, 2012. *Electricity Technologies.* [Online] Available at: http://www.eskom.co.za/AboutElectricity/ElectricityTechnologies/Pages/Electricity_Technologies.aspx [Accessed 25 9 2014].

Eskom Holdings SOC Limited, 2012. *Tarriff History.* [Online] Available at: http://www.eskom.co.za/CustomerCare/TariffsAndCharges/Pages/Tariff_History.aspx [Accessed 19 6 2013].

Haiqing, G., Yitai, M. & Minxia, L., 2006. Some design features of CO swing piston expander. *Applied Thermal Engineering,* 26(2), pp. 237-243.

Kim, H. J., Ahn, J. M., Park, I. & Rha, P. C., 2007. Scroll expander for power generation from a low-grade steam source. *Proceedings of the Institution of Mechanical Engineers, Part A: Journal of Power and Energy,* 221(5), pp. 705-711.

Marion, M., Voicu, I. & Tiffonnet, A.-L., 2012. Study and optimization of a solar subcritical organic Rankine cycle. *Renewable Energy,* 24 May, Issue 48, pp. 100-109.

Mathias, J. A. et al., 2009. Experimental Testing of Gerotor and Scroll Expanders Used in, and Energetic and Exergetic Modeling of, an Organic Rankine Cycle. *Journal of Energy Resources Technology,* 131(1), p. 012201.

Mohd.Tahir, M., Yamada, N. & Hoshino, T., 2010. Efficiency of Compact Oraginc Rankine Cycle System with Rotary-Vane-Type Expander for Low-Temperature Waste Heat Recovery. *International Journal of Environmental Science And Engineering,* 2(1), pp. 11-16.

NIST, 2013. *National Institute of Standards and Technology Chemistry Webbook.* [Online] Available at: http://webbook.nist.gov/chemistry/ [Accessed 16 February 2014].

Schuster, A., Karellas, S., Kakaras, E. & Spliethoff, H., 2009. Energetic and economic investigation of Organic Rankine Cycle applications. *Applied Thermal Engineering,* 29(8), pp. 1809-1817.

Subiantoro, A. & Ooi, K., 2010. Design analysis of the Revolving Vane expander in a transcritical carbon dioxide refrigeration system. *International Journal of Refrigeration,* 33(4), pp. 675-685.

The Editors of Encyclopædia Britannica, 2014. *Steam Engine (Machine).* [Online] Available at: http://www.britannica.com/EBchecked/topic/564472/steam-engine [Accessed 21 10 2014].

Wang, X. et al., 2010. Performance evaluation of a low-temperature solar Rankine cycle system utilizing R245fa. *Solar Energy,* 84(3), pp. 353-364.

Yagoub, W., Doherty, P. & Riffat, S., 2006. Solar energy-gas driven micro-CHP system for an office building. *Applied Thermal Engineering,* 26(14), pp. 1604-1610.

Yamamoto, T., 2001. Design and testing of the Organic Rankine Cycle. *Energy,* 26(3), pp. 239-251.

Yang, B. et al., 2009. Experimental investigation on the internal working process of a CO2 rotary vane expander. *Applied Thermal Engineering,* 29(11), pp. 2289-2296.

Zhang, B. et al., 2007. Development of a double acting free piston expander for power recovery in transcritical CO2 cycle. *Applied Thermal Engineering,* 27(8), pp. 1629-1636.