ORGANIC RANKINE CYCLE WITH POSITIVE DISPLACEMENT EXPANDER AND VARIABLE WORKING FLUID COMPOSITION

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ABSTRACT

Organic Rankine Cycles are often used in the exploitation of low-temperature heat sources. The relatively small temperature differential available to these projects makes them particularly vulnerable to changing ambient conditions, especially if an air-cooled condenser is used. The authors have recently demonstrated that ~~an~~ a dynamic ORC with a variable working fluid composition, tuned to match the condensing temperature with the heat sink, can be used to achieve a considerable increase in year-round power generation under such conditions [1]. However, this assumed the expander was a turbine capable of operating at multiple pressure ratios for large scale applications. This paper will investigate if small scale ORC systems that use positive-displacement expanders with fixed expansion ratios could also benefit from this new concept. In this paper, a numerical model was firstly developed. A comprehensive analysis was then conducted for a case study. The results showed that the dynamic Organic Rankine Cycle concept can be applied to lower-power applications ~~that use~~ that use positive-displacement expanders with fixed expansion ratios and still result in improvements in year-round energy generation.

NOMENCLATURE

T	[K]	Temperature
P	[bar]	Pressure
h	[J/kg]	Enthalpy
S	[J/kg.K]	Entropy
Q	[W]	Heat Transfer Rate
W	[W]	Mechanical Work
Special characters		
φ	[%]	Improvement in Annual Energy Generation
η	[%]	First Law Efficiency
Subscripts		
1		Pump Inlet
2		Pump Outlet
2b		Regenerator Outlet (Cold)
3		Expander Inlet
4		Expander Outlet
4b		Regenerator Outlet (Hot)
ambient		Ambient
evap		Evaporator
cond		Condenser
sat		Saturation Point
dyn		Dynamic Cycle
con		Conventional Cycle
ORC		Organic Rankine Cycle

INTRODUCTION

Large amounts of energy are known to be contained in relatively low-~~enthalpy~~ temperature heat sources, such as geothermal resources, solar thermal, and waste heat from industry. The Organic Rankine Cycle is generally accepted to be the most economically viable technology to exploit these resources [2].

70% of Geothermal resources worldwide are estimated to be at a temperature of 100 to 150°C [3], which is estimated to be capable of providing 350TWh/year in Europe alone [4].

The Organic Rankine Cycle has also been considered for application to waste heat recovery [5] [6], solar thermal [7] [8], biomass [9], and even Ocean Thermal Energy conversion [10]. Most of the previous work carried out on ORC systems has focused on the case of a single-component working fluid [11].

However, several issues exist which have hereto prevented large-scale implementation of the Organic Rankine Cycle in the field.

One of the more important among these is that the low temperature difference available to drive the cycle leads to both low thermal efficiencies, and high sensitivity to changes in the temperature of either the heat source or the heat sink [12]. This is of particular severity for the case of an air-cooled condenser operating in a continental climate, where the annual variation in temperature can exceed 50 °C, which is the case considered in this paper. This problem of low efficiency has been identified before in literature [12].

The authors have previously proposed a dynamic Organic Rankine Cycle using a zeotropic mixture as its working fluid to address this challenge [1]. A zeotropic mixture has several characteristics which make it appropriate for this sort of application. Firstly, it exhibits a temperature variation, or "glide" during phase change [13], and secondly, it has bubble and dew points between those of its two constituent parts. This allows a mixture to be produced with a specific bubble or dew point by selecting the correct composition of working fluid [14]. Some research has previously been carried out into the performance of zeotropic working fluids. Most found no detriment to the first law efficiency of the cycle, and many reported increases in the utilisation of the waste heat source, the temperature glide allowing a greater temperature drop in the hot side of the evaporator while maintaining the same pinch point temperature difference [15] [16] [17].

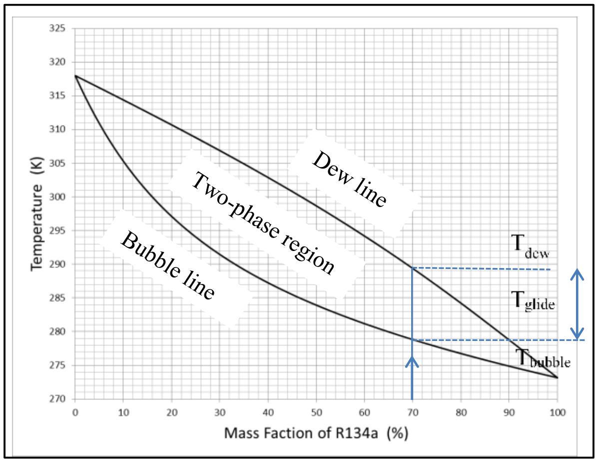


Figure 1: Bubble and Dew Curves of a mixture of R134a and R245fa at a pressure of 2.5 bar [1]

These properties allow for the concept of the dynamic cycle, which adds a composition tuning system to the conventional ORC, permitting the working fluid composition to be changed during operation, changing the bubble and dew points of the fluid and allowing the cycle to make the best use of a heat sink that varies in temperature. This paper considers a zeotropic working fluid consisting of a mixture of R245fa and R134a. R245fa was selected due to a variety of favourable properties. Its boiling point is low enough to ensure it will evaporate for the heat source temperatures we used, but high enough to ensure a condenser pressure above atmospheric for the heat sink temperatures [18]. Its critical temperature is also close to the heat source temperature, which means that a subcritical cycle with minimal superheat will have a smaller latent heat region in the evaporator, increasing utilisation of the heat source [19]. Also, by having a critical temperature that is not too far below the heat source temperature, the superheat at the expander inlet can be minimised by increasing the pressure ratio, increasing the efficiency [19].

R134a is selected as the secondary fluid, as its boiling point at the calculated condenser pressure is sufficiently different from that of R245fa to ensure adequate temperature glide for the proposed dynamic cycle.

The condenser pressure is selected to ensure that the fluid with the higher boiling point, in this case, R245fa, remains liquid on the hottest day of the year. This is also the operating condition for a conventional ORC. As the air temperature drops during the transition from summer to winter, so does the temperature of the coolant available to the cycle. This means that a fluid with a lower boiling point may be used, and still remain a liquid at the pump inlet. This is achieved by adding some R134a to the working fluid, lowering its boiling point.

The previous research from the authors considered such a system using a turbine as the expander [1]. This allowed the evaporator pressure to be increased as the boiling point of the fluid decreased, maintaining a constant superheat at the expander inlet. This showed a promising increase of 23% in annual energy production. It also analysed the feasibility of the online fluid composition tuning using a distillation column, and the economic viability of such a plant in light of the increased efficiency and capital expenditure. It was possible to conclude ~~from this~~ that the composition tuning could be carried out using simple, off-the-shelf components commonly used in the chemical industry, and that the introduction of the composition tuning would result in a higher NPV of the dynamic system for all operating periods over 3 years.

However, turbines become inefficient and expensive for lower-power applications, below a few hundred kW [20]. Such smaller systems tend to use positive displacement devices such as screw, scroll or ~~piston~~ rotary-vane expanders. These devices must be provided with a pressure ratio close to their own inbuilt volume ratio, or they will experience ~~high~~ over- or under-expansion losses, reducing their isentropic efficiency and reducing the overall efficiency of the cycle.

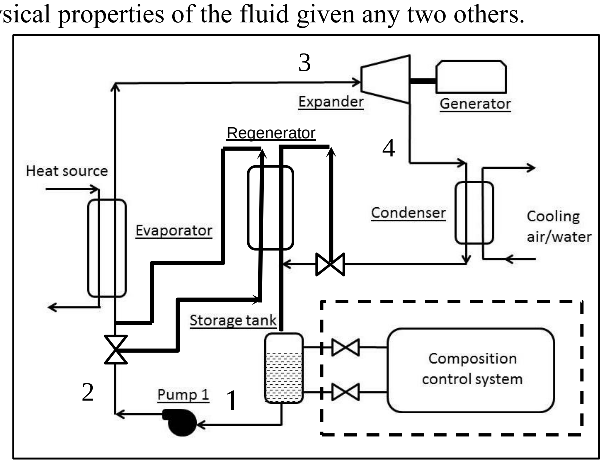
This paper investigates whether the dynamic Organic Rankine Cycle concept can still be applied to such a system, where the ~~pressure ratio~~expander's expansion ratios is ~~limited to a narrow range.~~ more or less fixed.

STEADY STATE NUMERICAL MODEL

As shown in

Figure 2~~Figure 2~~, a conventional ORC power plant has an evaporator (boiler), an expander, a condenser, a feed pump, and a liquid storage tank. In a real-world system, a composition tuning device consisting of a distillation column, storage tanks, and ancillary systems would also be present. However, for ease of analysis, this composition tuning subsystem is modelled as a black box that provides the desired fluid composition to the cycle with no significant transient or long-term effects. Previous research by the authors demonstrated that the overall parasitic power required to operate the composition tuning system was negligible in comparison to the power output of the cycle [1].

It is also assumed that there is no pressure or heat losses from piping or heat exchangers, no significant change in velocity, no change in elevation, and no effects due to compressibility. A steady-state numerical model was developed using MATLAB with REFPROP 9.1 providing the thermophysical properties of the working fluid [21]. This combination allows the use of the "refpropm" function in MATLAB, which can calculate most important thermal and physical properties of the fluid given any two others.



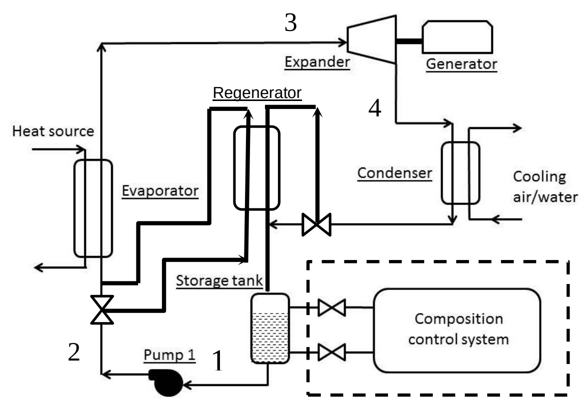


Figure 2: Schematic diagram of a dynamic ORC power plant, showing how the flow can be redirected through a regenerator if required

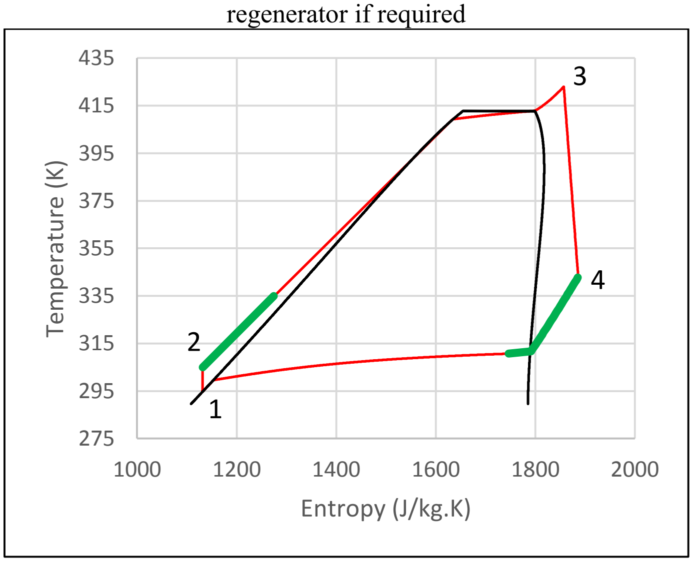


Figure 3: Temperature – Entropy diagram of a conventional zeotropic ORC cycle, with the regenerative portion marked in green

An Excel file containing ambient temperature data for Beijing, China, was then linked to this numerical code to use as a case study. From this file, the temperature of coolant available for the cycle could be obtained and used as the heat sink temperature for the power cycle, by using the "xlsread" function to create a MATLAB array.

The naming convention for points in the cycle is shown in Figure 2~~Figure 2~~ and Figure 3. The pinch point temperature difference at the condenser outlet is taken to be 5 °C which is consistent with previous research [22], [23]. An additional 2 °C of sub-cooling is also added, to ensure the working fluid was liquid at the pump inlet, giving the following equations:

$$T_1 = T_{\text{ambient}} + 5 \tag{1}$$

$$P_1 = P_{\text{sat}} \ @(T = T_1 + 2) \tag{2}$$

Knowing the temperature and the quality of the fluid allows REFPROP to determine the condenser pressure of the system for the hottest day of the year, which is also when the working fluid is composed entirely of the fluid component with the higher boiling point, in this case R245fa.

REFPROP 9.1 allows for the calculation of fluid properties on its extensive list so long as two properties are known. For example, knowing the and the pressure is enough for the program to point, dew point, enthalpy and entropy. Glide

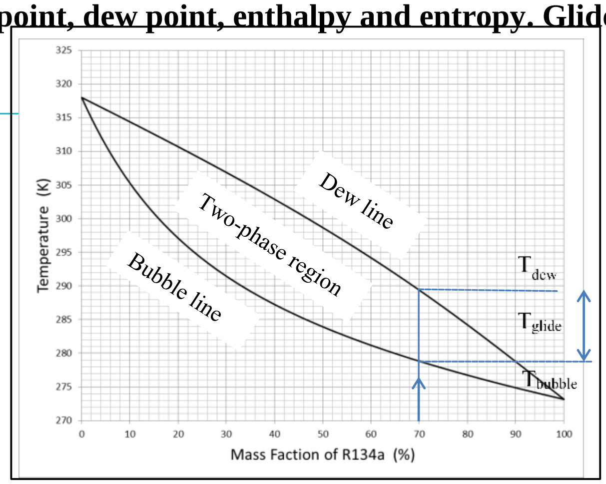


Figure 1~~Figure 1~~, can then be generated. Once the bubble point is known, the working fluid composition required to satisfy equation (2) at the desired condenser pressure can easily be calculated for any ambient temperature from the array provided to the program.

With the composition of the working fluid known, the rest of the cycle can be analysed using well-established thermodynamic techniques [24], [25]. A pinch point temperature difference at the evaporator inlet of 5 °C was used, and also a 5 °C superheat to ensure a pure vapour was fed into the expander, which allows the evaporator pressure to be calculated, using the equations

$$T_3 = T_{heat\ source} - 5 \tag{3}$$

and

$$P_{evap} = P_{sat} \ @(T_3 - 5) \tag{4}$$

The evaporator pressure was calculated using equation (4) for the fluid composition on the hottest day of the year, and held at this value year-round.

The isentropic efficiency of the pump and the expander were taken to be 90% and 70%, respectively. The isentropic efficiency of a positive displacement device is taken to remain constant as long as the expansion ratio does not change. Assuming isentropic pumping and expansion, $h_{2,\text{isentropic}}$ and $h_{4,\text{isentropic}}$ can then be obtained from REFPROP, and used to calculate the actual values, using the equations:

$$\eta_{pump} = \frac{(h_2 isentropic - h_1)}{(h_2 - h_1)} \tag{5}$$

$$\eta_{expander} = \frac{(h_3 - h_4)}{(h_3 - h_{4isentropic})} \tag{6}$$

The amount of energy transferred in the regenerator could also be calculated. Initially assuming zero enthalpy transfer, which would give a pinch point temperature difference of T_4-T_1, the program gradually increased the enthalpy transfer, monitoring the temperature difference between the hot and cold flow at 100 different points until the minimum temperature difference reached the pinch point value of 5K. The value of enthalpy transfer that gave this pinch point value could then be designated as $Q_{regenerator}$

Once this has been done, two properties are known for each of the four key points in the cycle; pump outlet, evaporator outlet, expander outlet and condenser outlet, and so Equations (7), (8), (9) and (10) can be used to calculate the efficiency of the cycle.

$$W_{pump} = h_2 - h_1 \tag{7}$$

$$W_{expander} = h_3 - h_4 \tag{8}$$

$$Q_{evaporator} = h_3 - h_2 \tag{9}$$

$$\eta_{cycle} = \left(\frac{(W_{expander}) - (W_{pump})}{Q_{evaporator} - Q_{regenerator}}\right) \tag{10}$$

This lets the efficiency of the cycle be calculated for any temperature fed to it from the excel spreadsheet. Using actual climate data in the spreadsheet allows the year-round performance of a Dynamic ORC to be calculated.

Four key metrics were analysed by the program. Firstly, the efficiency of the conventional ORC, $\eta_{con.}$ This is the efficiency of the cycle on the hottest day of the year. Secondly, the efficiency of the dynamic ORC, $\eta_{dyn.}$ This is the performance of the dynamic ORC with a given ambient temperature. Both $\eta_{con.}$ and $\eta_{dyn.}$ are calculated using equation (10).

Thirdly, the annual average efficiency of the dynamic ORC, $\overline{\eta}_{dyn}$. This is defined as

$$\overline{\eta}_{dyn.} = \frac{\sum_1^N \eta_{dyn.}}{N} \tag{11}$$

where N is the number of operational days in the year. Finally, ψ, the improvement in annual energy generation, given by

$$\psi = \frac{\overline{\eta}_{dyn.} - \eta_{con}}{\eta_{con}} \times 100\%. \tag{12}$$

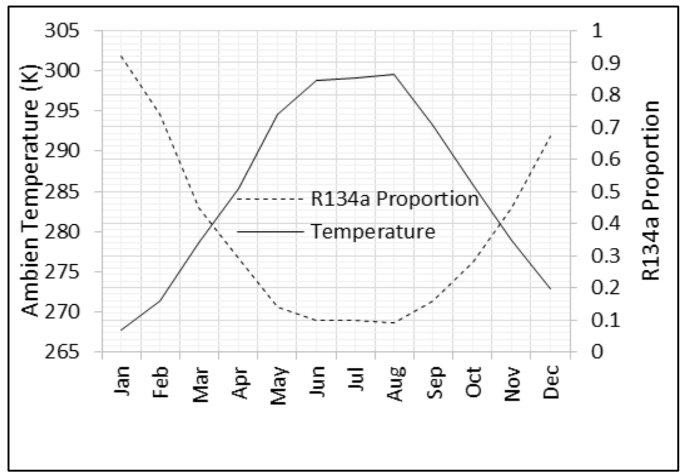
The model was validated against experimental results obtained by Kang [27] by using the same initial parameters, and produced results that were within 2% of his values for all points of the cycle as shown in the authors' previous research [24].

RESULTS

The MATLAB routine based on the equations presented in the previous section was used to analyse the case study of a dynamic ORC power plant operating under Beijing's ambient conditions for two different heat source temperatures.

Figure 4: Annual Temperature variation and associated changes in working fluid composition

~~Figure 4~~ shows the annual variation in temperature and the change in working fluid composition in necessitates. During the warmer summer months, the fluid must be entirely composed of R245fa to ensure a subcooled liquid at the pump inlet. During the colder winter months, the lower temperatures allow the working fluid to comprise a higher proportion of R134a, and still allow this pump inlet condition to be met.



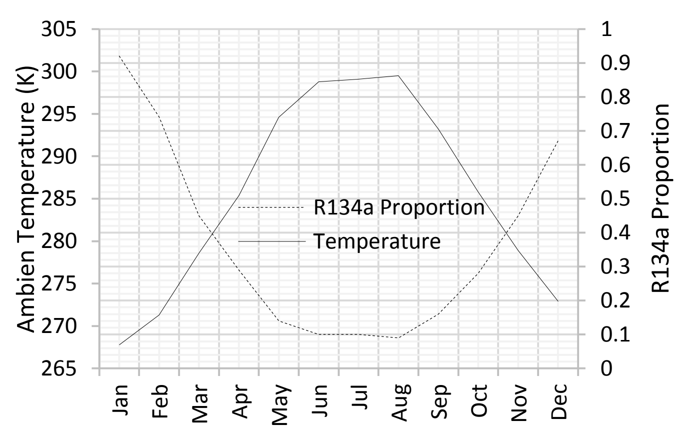


Figure 4: Annual Temperature variation and associated changes in working fluid composition

~~**Figure 4: Annual Temperature variation and associated changes in working fluid composition**~~

Figure 5~~Figure 5~~ shows the variation in efficiency over the course of the year for four cycle configurations, ~~and~~ when ~~a~~the heat source temperature is fixed at~~of~~ 100°C. Table 1~~Table 1~~ presents the annual average efficiencies of the conventional ORC and the dynamic cycle with and without a regenerator heat exchanger for comparison.

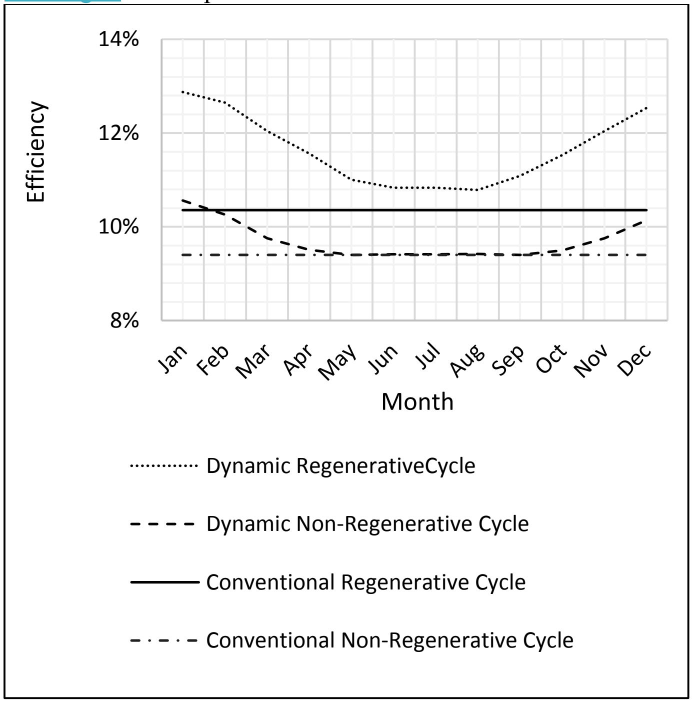


Figure 5: Average monthly efficiency of regenerative and non-regenerative cycles for a heat source temperature of 100°C

Table 1: Comparison of values of ψ for a heat source temperature of 100 celsius

Temperature	100°C	
Regenerator	Yes	No
Positive Displacement, Conventional	10.~~36~~4%	9.4~~0~~%
Positive Displacement, Dynamic cycle average	11.~~65~~7%	9.7~~1~~%

It can be seen that all of the dynamic cycles perform better during the colder months. For the positive displacement expander and no regenerator, the cycle is more efficient in the winter months, but the improvement in the annual average efficiency ψ is limited to 3.3%. This poor improvement in performance is primarily due to the fixed expansion ratio. As the temperature drops and the proportion of R134a in the working fluid is increased, the degree of superheat at the expander inlet also increases, and with no increase in expansion ratio, this also leads to an increase in superheat at the condenser inlet, increasing condenser and evaporator loading and negating much of the potential benefit of the working fluid composition shift. This is particularly noticeable during the summer months. The regenerative cycle shows a noticeable gap between the hottest day of the year, represented by the horizontal line, and the average monthly performance, represented by the dotted line. This gap is not present in the non-regenerative cycle.

The increased superheat at the expander outlet does mean that if a regenerator is included it will have a greater temperature difference to exploit between its hot and cold sides as shown in Figure 6~~Figure 6~~. The heavy green lines show the proportion of the cycle that occurs in the regenerator, and are noticeably larger in the right hand, colder condition plot. This means that the regenerator can transfer more energy into reducing the evaporator duty and increasing the efficiency of the cycle during the colder months, as demonstrated in Figure 7~~Figure 7~~. This means that the dynamic cycle can generate 12.44% more power on an annual basis than the conventional cycle when a regenerator is present.

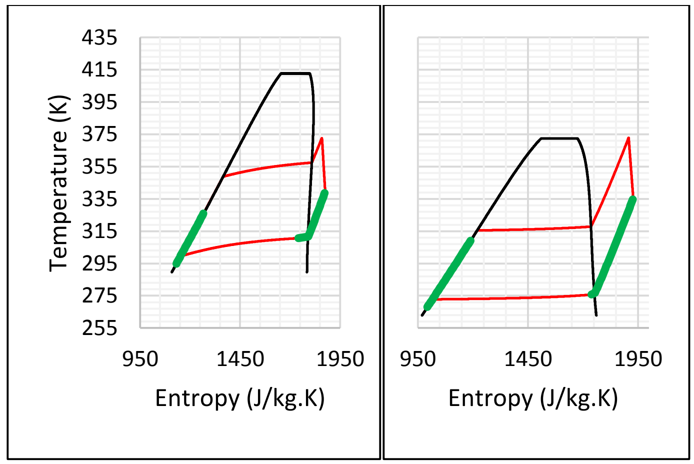


Figure 6: Comparison of T-s diagrams for warm and cold ambient conditions

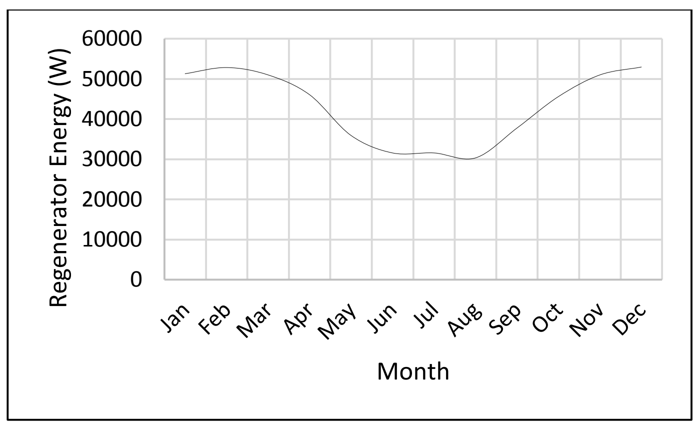


Figure 7: Regenerator Enthalpy change over the year for a positive displacement expander and a heat source temperature of 100 degrees

The trend of improved performance for the dynamic cycle is repeated for the higher heat source temperature of 150°C, as can be seen in Figure 8~~Figure 8~~ and Table 2~~Table 2~~. The dynamic cycles are more efficient in the colder months of the year, leading to increased energy production.

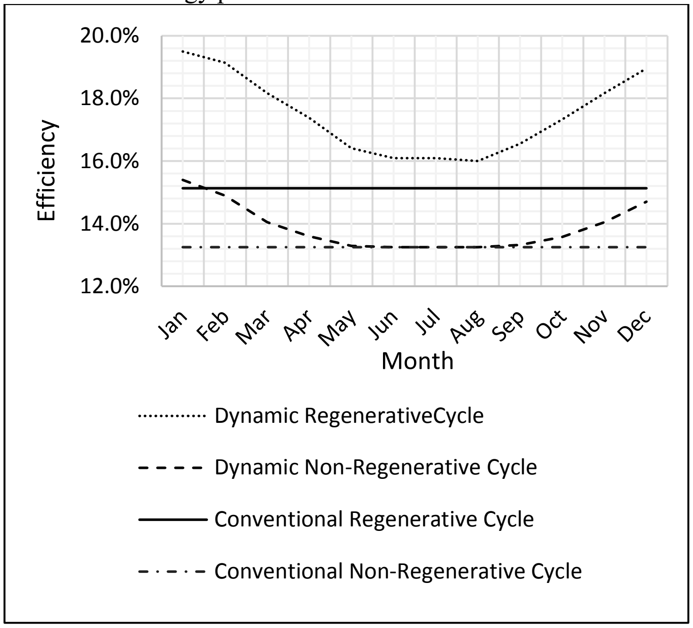


Figure 8: Average monthly efficiency of regenerative and non-regenerative cycles for a heat source temperature of 150°C

Table 2: Comparison of value of ψ for a heat source temperature of 150°C

Temperature	150°C	
Regenerator	Yes	No
Positive Displacement, Conventional	15.1~~3~~%	13.~~25~~3%
Positive Displacement, Dynamic cycle average	17.~~48~~5%	13.~~89~~9%

The annual improvement in energy generation for the positive displacement cycle is 4.8~~1~~% without a regenerator, again due to the fact that the fixed pressure ratio leads to increased superheat and more energy rejected through the condenser when the composition of the cycle shifts towards R134a. When a regenerator is installed to recover this heat, the improvement in annual energy generation increases to 15.5~~2~~%.

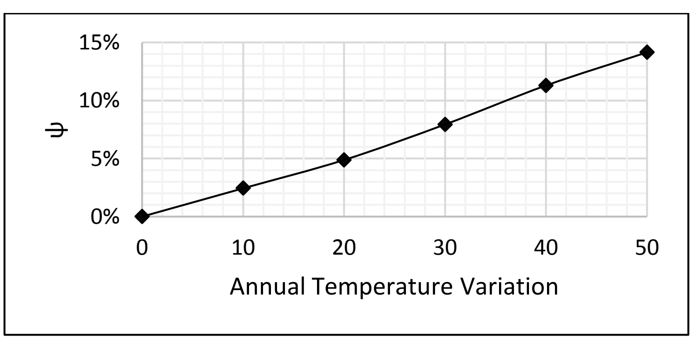


Figure 9~~10~~: Variation in ψ with changing annual variation in ambient temperature

Figure 9~~Figure 10~~ shows the variation in ψ for a cycle with a positive displacement expander ~~cycle with~~subjecting to a changing ambient ~~climate~~condition. The heat source temperature was held constant at 150°C, and the annual temperature variation was modelled as a sine wave with varying amplitude, creating an array which could be analysed using the same techniques as previously.

The increase in efficiency with increasing temperature difference is fairly linear, as was expected, as the Carnot efficiency of the cycle increases linearly with increasing temperature difference. Small deviations from a linear plot can be observed, however, as the second law efficiency of the cycle does vary according to the particular working fluid composition in operation. For lower temperature variations, only a small amount of R134a will ever need to be introduced into the system to keep the liquid pump inlet condition satisfied all year round. For higher temperature variations, there may be excess subcooling even when the working fluid is 100% R134a.

CONCLUSIONS

This paper investigates if the recently developed dynamic ORC ~~cycle~~ can be applied to small-scale systems based on positive-displacement expanders with fixed expansion ratios. The results show that ~~The~~ the dynamic ~~Organic Rankine Cycle~~ORC ~~described in this paper was shown to be~~is capable of increasing the system's annual average efficiency~~annual energy production~~ ~~from~~ for a given heat source~~,~~. However, such an improvement is~~much~~ less than that of the large scale system using turbine expanders with variable expansion ratios. Furthermore, such benefit strongly depends on heat recovery via the regenerator. The higher~~is~~ the heat regeneration, the higher~~is~~ the efficiency improvement. This is because the expander with a fixed expansion ratio ~~approximately~~ has an approximately~~a~~ constant pressure ratio between its inlet and outlet. The increase of pressure ratio between the evaporator and condenser by tuning the condensing temperature to match colder ambient condition in winter cannot be utilized by such expanders. However, with the regenerator in place, the higher discharging temperature of the expander could increase the heat recovery and consequently reduce the heat input at the evaporator, ultimately increasing the thermal efficiency.

~~so long as a regenerator was used to recover the excess superheat incurred by using a fixed evaporator and condenser pressure.~~

~~These results show that the proposed dynamic Organic Rankine Cycle is not only viable for larger-scale plants, but also for small-scale plants, with an installed capacity of less than a few hundred kW, on which scales turbines are uneconomically expensive, and positive displacement expanders are the most economical technology, particularly when the heat source temperature is 150°C and above.~~

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