EFFICENCY OF THE ARCHISOL CONCEPT

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ABSTRACT

A simplified ORC cycle is proposed for small temperature differences. The most important simplification is the replacement of the pump section by a hydraulic column. The weight difference between the downward moving liquid column and the upward moving gas column provides the driving pressure for the cycle. Because the effect of this weight difference bears resemblance to the law of Archimedes, it is called the Archisol Concept

Especially for Ocean Thermal Energy Conversion applications, it is expected to be an advantage that less mechanical components are necessary for the cycle.
The University of Wageningen has carried out some first order calculations to check the thermodynamic feasibility.
The calculation shows that it is possible to produce electrical power with the Archisol concept for both scenarios; ocean (Twater hot/cold = 27/6 °C) (i) and residual/geothermal heat (Twater hot/cold = 70/20 °C)
Typical for the ocean scenario is:

- The refrigerant cycle itself is has a limited height, about 15 m, and operates with low pressure.
- A large part of the produced power output is consumed by the pumps of the evaporator and the condenser, which makes the outcome of the calculations very dependent on the energy consumption of the pumps. Most critical in the performance is the internal pressure drop of the water in the heat exchangers itself. The calculations are based on 0.3, but fact is that water supply lines are open systems, which can and will be polluted by dirt and biofouling, which can increase the water flow resistance strongly”

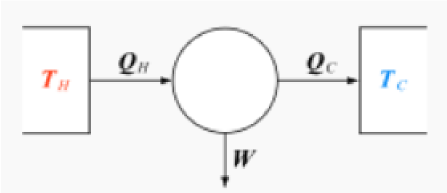
This leads to an efficiency of just over 2 % for the ORC cycle and 0,7 % for the complete process, taking secondary pump use for the water streams into account. The authors expect that the simplicity of the design could be attractive for further OTEC research and projects.

Further research is planned to demonstrate the viability of the proposed simplified ORC cycle.

1. INTRODUCTION

Power production out of waste heat recovery or natural heat sources (geothermal energy or temperature difference in deep ocean waters) can marked as durable, because they do not make (direct) use of fossil energy sources. Given the limited temperature level of the heat source, the power production will have two important characteristics:

1. According to Carnot, the efficiency of the power production using waste heat recovery or natural heat sources is limited by the temperature levels of the hot and cold source. The Carnot efficiency is defined as the ration of work W (which can be transferred into electrical power) and the hot source heat input. See figure 1.



$$\eta = \frac{W}{Q_H} = 1 - \frac{T_C}{T_H}$$

Figure 1: Definition of Carnot efficiency

Given the relative small temperature differences of these applications between the hot and cold source, relative large volumes of heat are required. The critical success factor is how efficient the power cycle can handle large heat flows. A good design reduces the hardware requirements and reduces the need for auxiliary power for pumps and fans.

2. The power cycle itself should match low temperature level of the heat- and cooling source in order to operate with acceptable pressure levels. Given the low temperature levels and looking for an continue cycle which can create a large power output, the use of the principle of the Organic Rankine Cycle is a logic choice. The ORC cycle exist of a closed refrigerant loop using the following components pump-evaporator-expander and condenser.

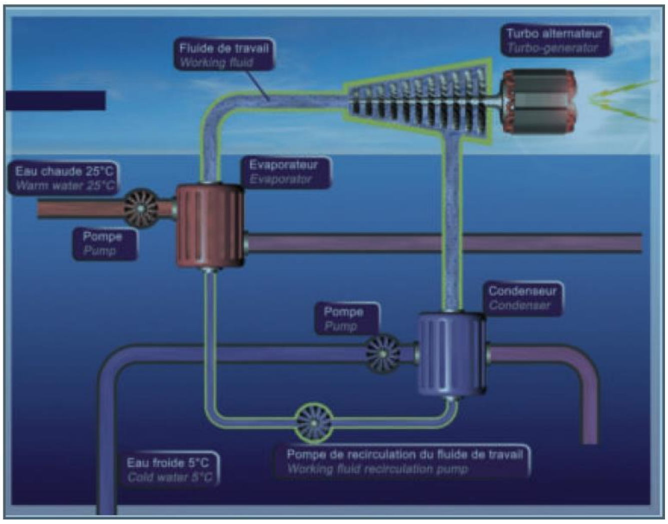


Figure 2: standard ORC cycle applied in Ocean Thermal Energy Conversion (Source: DCNS)

Archimedes Solutions has designed the so called Archisol concept. This Archisol concept is based on a modified ORC-principle. To evaluate the technical feasibility of this concept, Archimedes Solutions has asked Wageningen UR to make some first order feasibility calculations.
These first order feasibility calculations include flow- and heat losses. The auxiliary power requirements of pumps and/or fans are calculated based on usual and common used efficiencies of the components in the system.

The feasibility calculations are made for the following applications/scenario's.

1. Ocean Thermal Energy Conversion (OTEC)

OTEC uses the ocean's warm surface water with a temperature up to 27 °C to vaporize fluid and condensation of it takes place with cold seawater coming from deeper sea levels. Depending on the depth, the temperature of ocean water ranges from 6.5 to 8.5 °C. Using colder water from deep sea levels increases the efficiency of the cycle, but also requires more pumping power to transport it to the condensing heat exchanger. Table 1 gives an example of the temperature profile of deep ocean waters used as used in the calculations.

Depth (m)	Temperature (°C)
0	27
100	24
200	11
300	8.5
400	7.8
500	7
600	6.5

Table 1: Temperature distribution of deep ocean waters that is used in the calculations.

2. Geothermal/residual heat energy conversion

The application of power generation makes use of geothermal/residual heat of 70 °C water. In the most ideal situation cooling water of 20 °C is available as well. Additional a calculation will be made what the energy- and water consumption is when a cooling tower is used to produce 20 °C cooling water when the ambient temperature is 15 °C and a humidity is 70%.

2. DESCRIPTION OF THE ARCHISOL-concept

2.1 General

The Archisol-cycle is based on a standard Organic_Rankine_cycle. Where the ORC makes use of a pump to increase the refrigerant liquid pressure, the Archisol-concept uses the weight of a liquid refrigerant column itself to increase the liquid pressure. The advantage of not having a pump is that it increases the reliability and reduces the auxiliary energy to drive the pump. The disadvantage is that the installation needs sufficient height, which makes the cycle Archisol-concept less compact than the standard ORC. The authors have no knowledge of previous research where gravity is used to pressurize the liquid, so as far as they are aware this can be considered a new concept in an ORC like setup. The height of the liquid column corresponds with the pressure build up and the choice of the working fluid. The pressure build up corresponds with the available temperature difference. The Archisol concept is therefore most suitable for situations with small temperature differences and large streams of warming and cooling power.

2.2 Description of Archisol-cycle

The Archisol concept exists of a closed loop cycle, based on four sequential stages: condensing, increasing liquid pressure, evaporation and gas expansion. See figure 3. A short functional description of each stage is given below:

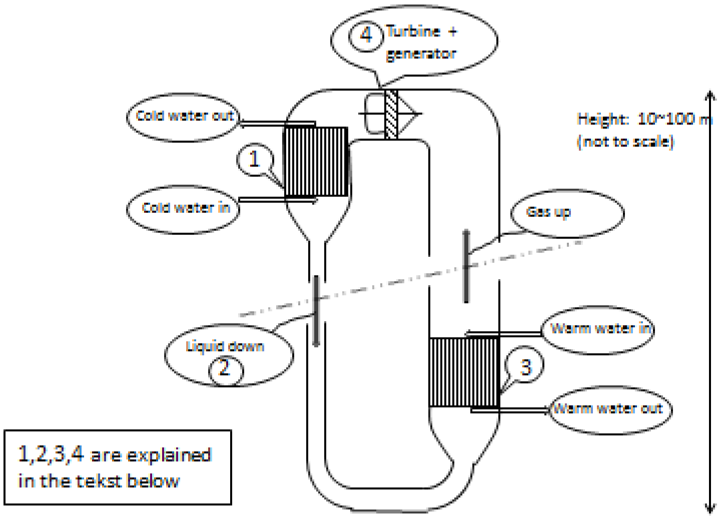


Figure 3: Cycle configuration with gas expansion

1. Condensing:

   Working fluid is condensed near to water surface in the condenser (10), using water with a low temperature pumped up from a water level deep below the surface (11).

   Condensing temperature: $$T_c = T_{\text{cold water condenser in}} + \Delta T_{\text{condenser hex}} \quad (1)$$

2. Increasing liquid pressure:

   The liquid from the condenser is pressurized by the weight of the liquid column. The height of the liquid column is self-adjusting untill there is the following equilibrium:

   Maximum liquid pressure: $$P_o = P_c + \rho_{\text{liquid}} . g . h_{\text{liquid culumn}} - \Delta P_{\text{flow losses liquid}} \quad (2)$$

3. Evaporation (/super heater)

   The working fluid is evaporated in the evaporator (6) in the lower part of the cycle, using warm water from the surface (7) pumped downwards to the evaporator. To prevent liquid formation during gas expansion in the turbine, gas should have an minimum superheating. The superheat may be controlled by controlling the liquid flow to the evaporator. The required superheat reduces the maximum evaporation temperature To.

Evaporation temperature: $T_0 = T_{\text{warm water evap in}} - \Delta T_{\text{evaporator hex}} - \Delta T_{\text{min required superheat}}$ (4)

4. Gas expansion

The pressurized gas is expanded in the expansion device (14) at the higher part cycle. Expansion of gas reduces both temperature and pressure. The ideal gas expansion process, which produces the most power, is the isentropic expansion. The practical power production can be calculated by making use of the isentropic efficiency of the expansion device. The power production is dependent on the available pressure difference and the gas flow. The pressure difference is basically the difference between the evaporation pressure and the conderser pressure, minus the additional pressure losses in the cycle. To prevent liquid formation during expansion, the gas should be superheated adequately. This limits the evaprorator pressure and thus reduces the practical availabele pressure difference.

The available pressure difference over turbine is:

$$\Delta p_{\text{turbine}} = P_o - P_c - \Delta p_{\text{flow losses gas}} - \Delta p_{\text{gas column}} \quad (4)$$
$$= P_o - P_c - \Delta p_{\text{flow losses gas}} - \rho_{\text{gas}}.g.h_{\text{cycle}}$$

The maximum theoretical power production of the gas driven turbine is calculated by single stage isentropic expansion:

$$W_{turbine\ isentropic} = \dot{m} \cdot (h_{gas\ in} - h_{gas\ out,s}) \quad (5)$$

2.4 Auxiliary systems

To drive the cycle auxiliary systems are required. Their power consumption is very important in calculating the net power production.

The two most important auxiliary systems are the pipe lines with their water pumps. These are required to transport cold water to the condenser en warm water to the evaporator.

In order to determine the electrical power consumption of these pumps, the hydraulic power has been calculated based on the flow and the pressure difference over the pump.

The pressure difference over a pump is based on the static pressure difference caused by the difference in density of the water itself plus the dynamic friction losses caused by the flow.

$\Delta p_{\text{pump}} = \Delta p_{\text{static}} + \Delta p_{\text{dynamic}}$

The static pressure difference caused by the difference in density of the water in the pipe compared with the average density out of the pipe. The used density of sea water is given in figure 4.

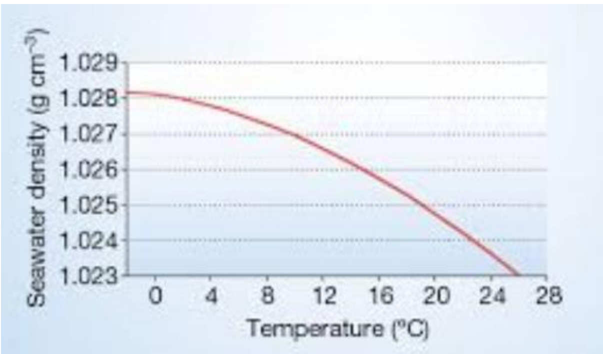


Figure 4: Salt water temperature- density relation

The dynamic pressure difference is caused by the flow losses in the pipe and in the heat exchanger. The flow losses in the pipe have been calculated by using the friction factor of a turbulent flow in a round pipe with a roughness factor (ε/d) of 0.001.
The pressure loss in the condenser and evaporator is an important design parameter, because it will strongly influence the outcome of the calculations. The calculation is made assuming a pressure loss of 0.3 bar*.

** Note that the water supply lines are open systems, which means that these can and will be polluted by dirt and organic material. This will increases internal resistance of the pipeline and the condenser and evaporator. The latter will have more impact . This all means an increase of the actual pressure drop over time compared to the design value.*

Once the pressure difference over the pump and water flow are known, the electric power consumption of these pump can be calculated when the efficiency of the pump and the used electromotor are known:

$$P_{elec\ pump} = W_{hydraulic} / (\eta_{pump} * \eta_{elecmotor}) = Q*(\Delta p_{static} + \Delta p_{dynamic}) / (\eta_{pump} * \eta_{elecmotor}) \qquad (6)$$

Thermal losses
Apart from the auxiliary power consumption these water transport systems have also thermal losses. These have been calculated, assuming the use of a 5mm PP coating on the inner diameter op the pipe. Compared to other losses, the thermal losses appeared to be small for high capacity systems. In the calculations the water pipe diameter is 1 m. The largest losses are the thermal losses of the cold water supply line and these are less than 0,1 K. This means that thermal losses are negligible for large capacity systems.

3. CALCULATION RESULTS

3.1 General

In order to calculate the power production and the auxiliary power requirements of pumps and/or fans assumptions have to be made for losses and efficiency of components. The efficiency values used in the calculation of the most important components are given in table 2. The calculation results are given in the next paragraphs of this chapter.

pump evaporator water		unit
efficiency electromotor	0.85	-
efficiency pump	0.8	-
pump condenser water		
efficiency electromotor	0.85	-
efficiency pump	0.8	-
Turbine efficiency		
turbine efficiency	0.9	-
generator efficiency	0.85	-
Heat exchanger		
Pressure drop	0,3	bar

Table 2: Main parameters required to make the efficiency calculations.

3.2 Performance calculations for ocean application

The following table gives the results for the Ocean application:

Refrigerant	**R134a**	
Power production turbine	647	kW
P elect LT water pump	318	kW
P elect HT water pump	112	kW
Net power production	**217**	**kW**
Condenser water transport pipe		
length	600	m
d-pipe	1	m
flow speed	3	m/s
Mass flow	2419	kg/s
Twater condenser in	6.6	°C
Twater condenser out	9.6	°C
Evaporator water transport pipe		
length	11.63	m
d-pipe	1	m
flow speed	3.07	m/s
Mass flow	2475	kg/s
Twater evaporator in	27.0	°C
Twater evaporator out	24.0	°C
Δp condenser water pump	0.92	bar
Δp evaporator water pump	0.31	bar
Condenser capacity Qc	28516	kW
Evaporator capacity Qo	29163	kW
Maximum possible Carnot efficiency	6.8	%
Calculated efficiency	**0.7**	**%**
Height liquid column	**12**	**m**
Max system pressure	**5.8**	**bar(a)**
Total refrigerant contents	**739**	**kg**

Table 3: Main results of efficiency calculations ocean/refrigerant R134a scenario.

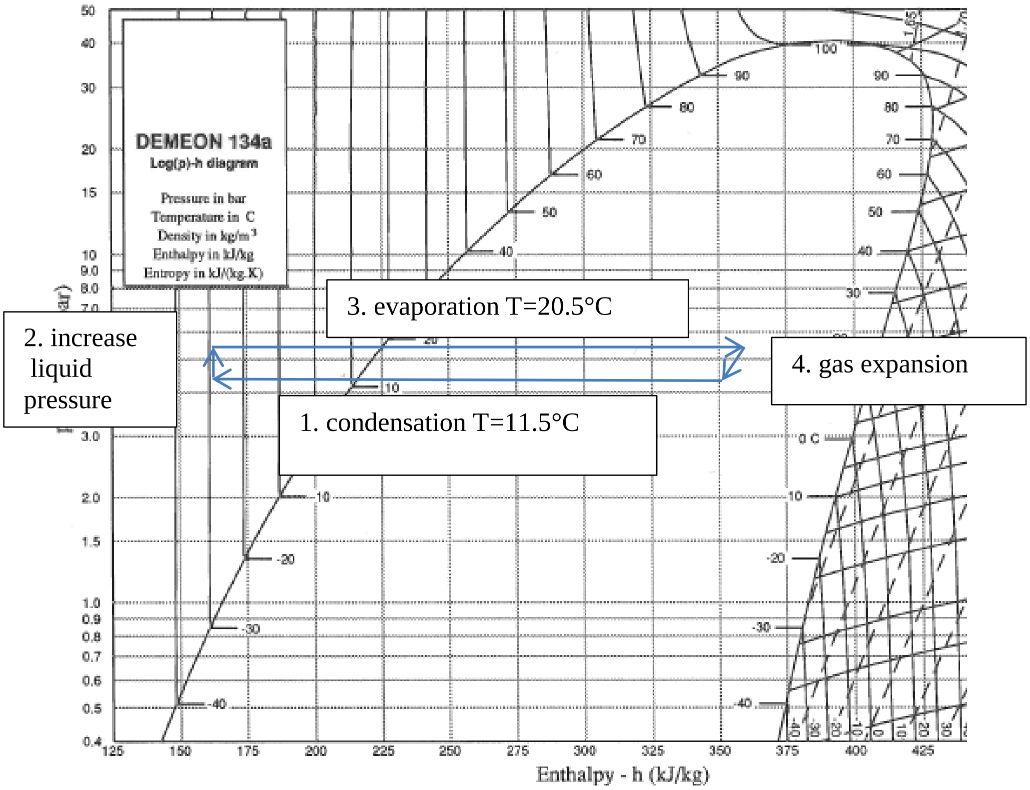


Figure 5: Process cycle of gas expansion ocean scenario indicated in h – log p diagram

The results show that a large part of the produced power is consumed as auxiliary power for the pumps. This is indirect a result of making use of a relative small temperature difference of the ocean water, with a corresponding low Carnot efficiency number. This means that relative large heat flows are required to drive the cycle. These relative high heat flows increase the investment for the heat exchangers and the use of their auxiliary power.

Result is that the outcome of the calculations is very dependent on the energy consumption of the pumps of the evaporator and the condenser. Most critical in the performance is the internal pressure drop of the water in the heat exchangers itself. This is an design parameter and is assumed to be 0,3 bar. This is possible for a clean and well-designed heat exchanger, for example an oversized shell and tube heat exchanger, but fact is that water supply lines are open systems, which can and will be polluted by dirt and other organic substances, which will increase the water flow resistance strongly.

The refrigerant cycle itself is has a limited height, about 15 m, and operates with low pressures.
The largest components of the total system are:
- the long pipeline to get cold water supply (diameter= 1 m and length= 600 m)
- the heat exchangers with their pumps

3.2 Economic considerations for ocean application

The main advantages concerning the Ocean application are considered to be:

- No working fluid pump and therefor lower investment and maintenance cost and no use of auxiliary power to drive the pump
- Compact platform design for applications at sea because of vertical placement of components
- The use of non-dangerous working fluid

The main issues are comparable to other OTEC systems:

- Overall thermodynamic efficiency is low because of the small temperature differences,
- Pump power use is high because of the long water pipelines and the pressure drop in the heat exchangers

Overall it is expected that the Archisol concept can bring economic advantages for OTEC applications, mainly due to the compacter platform design and the lower maintenance costs.

3.3 Performance calculations for residual/geothermal heat application

Refrigerant	**R134a**	
Power production turbine	2292	kW
P elect LT water pump	212	kW
P elect HT water pump	131	kW
Net power production	**1949**	**kW**
Condenser water transport pipe		
length	345	m
d-pipe	1	m
flow speed	3	m/s
Mass flow	2356	kg/s
Twater condenser in	20.00	°C
Twater condenser out	23.00	°C
Evaporator water transport pipe		
length	75	m
d-pipe	1	m
flow speed	3.07	m/s
Mass flow	2411	kg/s
Twater evaporator in	70.00	°C
Twater evaporator out	67.00	°C
Δp condenser water pump	0.61	bar
Δp evaporator water pump	0.37	bar
Condenser capacity Qc	29547	kW
Evaporator capacity Qo	31839	kW
Maximum possible Carnot efficiency	14.6	%
Calculated efficiency	**6.1**	**%**
Height liquid column	**98**	**m**
Max system pressure	**18.2**	**bar(a)**
Total refrigerant contents	**6102**	**kg**

Table 4: Main results of efficiency calculations. Scenario: residual heat/ refrigerant R134a

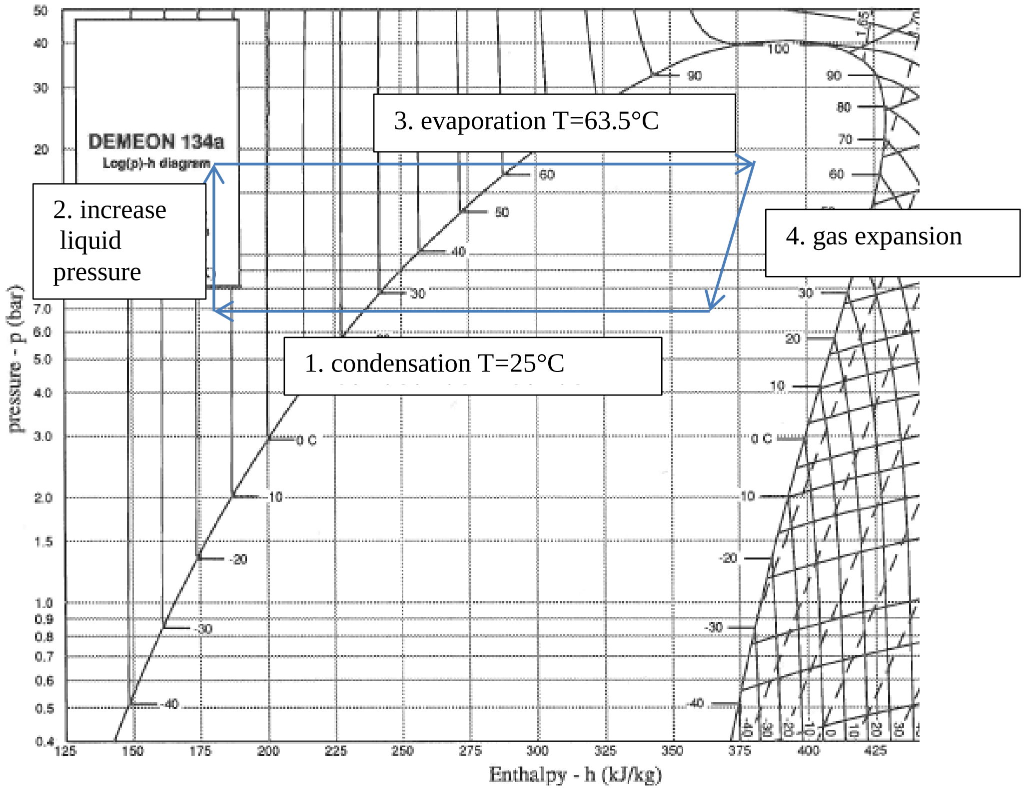


Figure 6: Process cycle of gas expansion residual heat scenario indicated in h – log p diagram

Compared to the results of the ocean application, here a smaller part of the produced power is consumed as auxiliary power for the pumps. This is a result of making use of a larger temperature difference, with a corresponding high Carnot efficiency number.
The outcome of the calculations is dependent of the energy consumption of the pumps of the evaporator and the condenser, but is here less critical than compared to the ocean application. Critical issue in this application is the presence of cooling water for the condenser. If not, air has to be used and that increases the investment and the auxiliary energy use..

The refrigerant cycle itself has a significant, height, about 100m and operates with moderate pressure.
The largest components of the total system are:
- the pipeline of the cold water supply (diameter= 1 m and length= 345m)
- the pipeline of the warm water supply (diameter= 1 m and length= 75 m)
- the heat exchangers with their pumps

The cold water supply line is quite long because of the fact that the water in and outflow should not be close to each other and the fact that the condenser is situated at a height of minimal 98 m. In order to reduce the pump power requirements the potential energy of the water flow has to be recovered by using a down flow pipe.

4. Conclusion

This article describes the feasibility study of the Archisol concept for both the ocean application and the geothermal heat application. The Archisol-concept is an ORC without a liquid pump. The functionality of the pump is replaced by the column weight of the liquid refrigerant.
The feasibility study has been made for a large capacity system, because auxiliary power consumption is a deciding factor. The ocean application has an lower efficiency than the geothermal application, 0.7% against 6.1%. This is mainly caused by the relative small difference between the surface and deep water temperature (27/6.6°C). The main risk is the increase of auxiliary power consumption by pollution of the open seawater heat exchangers. This is however common for OTEC applications. When comparing the Archisol concept to existing OTEC applications the simplified cycle design (no working fluid pump required) and a compact platform design are expected to be advantageous.

The residual/geothermal application has a better thermodynamic feasibility, especially when large scale cooling water is available. The use of ambient air for cooling the condenser reduces the efficiency or consumes large quantities of water.

Reference:

Kempener, Ruud (IRENA), Neumann Frank (IMIEU) OCEAN THERMAL
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