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Jean Castaing-Lasvignottes, Alexandre Dijoux, Frantz Sinama, Bertrand Clauzade, Olivier Marc Marc. A presentation of the ocean thermal energy conversion prototype in La Reunion. OTEC Symposium 2017, Sep 2017, Saint-Pierre, Réunion. <hal-02999786>

HAL Id: hal-02999786

<https://hal.science/hal-02999786v1>

Submitted on 11 Nov 2020

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A presentation of the ocean thermal energy conversion prototype in La Reunion.

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ABSTRACT

The electricity produced in tropical islands has to face the continuous growing demand of all the sectors and is mainly generated by fossil fuels with no possibility to connect the grid to other bordering countries. This is the case of the island of La Réunion, a French overseas territory situated in the Indian Ocean, off the east coast of Madagascar (Latitude 21S, Longitude 55.5E). La Réunion has launched a very ambitious energy program to reach the electricity autonomy by 2030. This context has contributed to the research of new ways of electricity generation by renewables. Among the various techniques to produce basic electricity, the one consisting in using Ocean Thermal Energy Conversion (OTEC) knows a very important growth worldwide. An onshore prototype of an Organic Rankine Cycle (ORC) driven by the temperature difference between seawater from the depth and from the surface has been designed and built and is under test in order to study the opportunity of producing electricity by this mean. This paper presents the whole installation and particularly focuses on heat exchangers that are key figures in such an ORC. Ammonia is the working fluid and the level of magnitude of exchanged heat in the condenser or the evaporator is about 500 kW for an aimed electricity production of about 15 kW. An experimental point is presented and analyzed in terms of temperature, pressure and of course energy balances. Results reveal quite close to those expected with an obtained efficiency of 2.5%, representing a thermodynamic efficiency (when compared to Carnot ideal cycle) of 35 %.

Keywords: Organic Rankine Cycle; Ocean Thermal Energy Conversion; experimentation; prototype; energy analysis

1 INTRODUCTION

Ocean Thermal Energy Conversion (OTEC) provides an interesting alternative to conventional renewable energy with the utilization of ORC. Using the temperature difference between the warm water at the upper surface and the deeper colder water (about 5 °C at 1000 m), it is possible to generate electricity. In this application ammonia is used as the working fluid. As represented in Fig. 1, the warm water from the ocean surface is pumped into the evaporator (E) where the working fluid receives the heat and vaporizes (1-2). The steam produced is expanded (2-3) in the turbine (T) and then condenses (3-4) at low pressure in the condenser (C) where the heat is exchanged to the cold source (cold water from depth). The fluid that has been condensed is returned to the evaporator (4-1) passing through the pump (P).

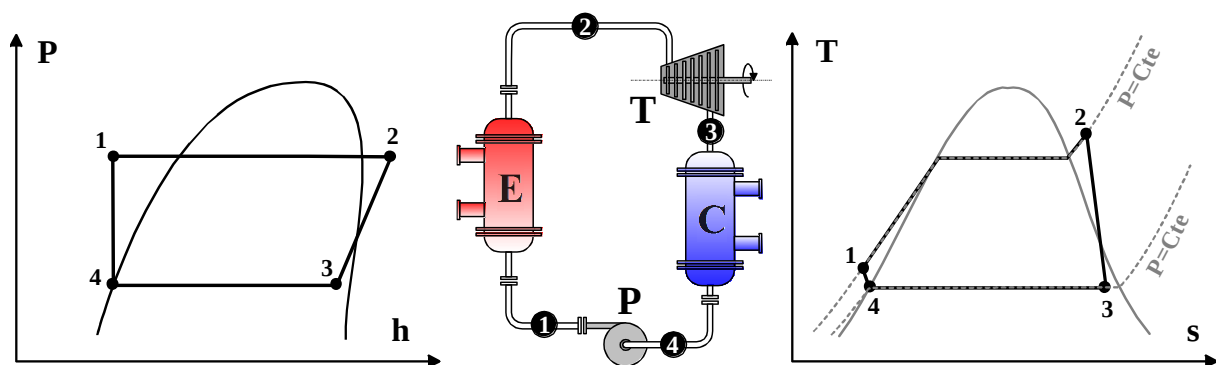


Fig. 1. Schematic of a closed Rankine cycle for OTEC and representation on a $\ln(P)$ - h and a T - s diagram

OTEC plants are particularly adapted to tropical zones [1], [2], [3] where surface water absorbs solar energy with a temperature water varying over a year between 23 and 30 °C, depending on the location (between 23°N and 23°S) [4]. Many island countries in the world have a very high potential for OTEC application in Pacific, Atlantic or Indian oceans [5], [6]. La Reunion is located in the sub-tropical area in the southern hemisphere (200 km West of Mauritius in the Indian Ocean).

Even if the solar contribution is free, the efficiency of such cycles suffers from the small temperature difference between the warm and the cold sources. Even in the Carnot ideal case, 8% is a maximum and unfortunately, obtained experimental values are much lower, generally close to 2-3% [7]. Nevertheless, OTEC plants can be combined with other applications such as Sea Water Air Conditioning for buildings (SWAC) [8], desalination [9] or aquaculture [10], [11]. Most of the work required to increase these efficiencies must be principally done on heat exchangers, which are of great importance in the energy balances and on OR Cycles performances. To face these challenges, a few pilot plants have been built all over the world to test the feasibility in terms of fluid, temperatures and heat exchanger configurations. This paper will firstly deal with the state of the art in terms of actual or passed plants working in the OTEC domain. Secondly, the OTEC onshore prototype situated in La Réunion (an overseas French department) will be presented in details. In the end, obtained experimental results will be discussed, in terms of energy balance and thermal efficiency.

2 BACKGROUND ON EXPERIMENTAL OTEC FACILITIES

The first formulation of OTEC was expressed by d'Arsonval, a French scientist in 1881. The first experimental test was demonstrated by another famous French scientist, Claude in Belgium in 1928 using the cooling water from a steel plant at 30°C as warm source and the water of the Meuse River at 10°C as cold source [24]. In this test, 50 kW power output was generated by an open cycle [12]. The success of this experimentation conducted Claude to test this installation on the sea in Cuba. The installation produced electricity but no net power due to a poor site selection and a mismatch of the power and seawater systems [13]. Many other OTEC plants have been built since that time in the world (Fig. 2).



Fig. 2. Location of OTEC plants in the world

The Table 1 summarizes and synthetizes the experimental facilities built since 1928, focusing in particular on operating conditions, obtained performances and technology used. Many other countries are nowadays beginning investigations in the OTEC field such as Brazil [14], India [15], Colombia [16] as well as in many islands [1], [2], [3] and [17].

Table 1
Summary of OTEC experimentations in the world since 1928

Country	Years	Installation Type	Cycle Type	Working Fluid	Warm / Cold	HEX type	Gross Power kW	Thermal efficiency (%)	Ref
					Range of temperature (°C)				
Belgium	1928	Onshore	Open	Seawater	30 / 10	Flash evaporation	50		[24], [29]
Cuba	1930	Onshore	Open	Seawater	27 / 13	Flash evaporation	22	1.3	[30]
USA	1979	Offshore	Closed	NH ₃	26.1 / 5.7	Titanium plate heat exchangers	53		[23]
Japan	1977 - 1980	Lab-based	Closed	R114	26-42/ 5-24	Plate heat exchangers	0.001 – 1.9		[33], [34]
Japan	1980	Offshore	Closed	R22	23.6 / 9.6	Plate-fin heat exchangers	1		[24]
Japan	1981 - 1984	Onshore	Closed	R22	29.8 / 5.6	Shell-and-tubes type with titanium tubes	120		[7], [33], [36]

Japan	1982 - 1984	Onshore	Closed	NH ₃	28.5 / 12	Plate evaporator / shell-and-tubes condenser	50		[37]
Japan	1985	Onshore	Closed	NH ₃	28 / 7	Plate heat exchangers	75		[38]
USA	1993 - 1998	Onshore	Open	Seawater	26 / 6	Flash evaporation	255		[30]
Japan	1995	Onshore	Kalina & Uehera	Water-NH ₃	28 / 7	Plate heat exchangers	4.5		[33], [40], [43]
Japan	2003-2009	Onshore	Uehera	Water-NH ₃	28 / 7	Plate heat exchangers	30		[33], [40], [43]
Fidji	2012	Lab-based	Closed	R134-a	24-30 / 4.5-5	Shell-and-tubes	0.015	1.4	[45]
Japan	2013	Onshore	Closed	R134-a	25.9-29 / 8.5-9	Plate heat exchangers	50		[44]
China	2013	Lab-based	Reheat	Water-NH ₃	24-30 / 5-15	horizontal tubular rising-film generator	0.052	0.75	[46]
USA	2014	Onshore	Closed	NH ₃	24-28 / 5-7	Brazed Aluminum plate heat exchangers	100		[32]
South Korea	2014	Onshore	Closed	R32	26 / 5	Semi-welded-type heat exchangers	20	2	[47]
La Réunion (France)	2011	Onshore	Closed	NH ₃	23-28 / 5	Plate heat exchangers Shell-and-tubes	15	2.5	This paper

3 LA REUNION EXPERIMENTAL PROTOTYPE

An OTEC (Ocean Thermal Energy Conversion) experimental installation was designed and built by a French naval firm, NAVAL ENERGIES, and experimented at Reunion University (Fig. 1). Sized to obtain 15 kW gross power, its objective is to test the Rankine cycle components and in particular heat exchangers (condenser and evaporator). The experimental installation is not connected to the ocean and a heat pump simulates hot and cold water sources. As shown in Fig. 3, the whole installation is composed of an experimental room containing the thermodynamic Rankine cycle and the heat pump and a second room for driving the machine, containing all the control facilities and the data acquisition system.



Fig. 3. The OTEC onshore prototype

3.1 Ammonia loop

There exists many papers dealing with the choice of the working fluid [7], taking into account thermodynamic performances, technological, safety, and environmental aspects. Despite its toxicology, ammonia was chosen. Its pressure/temperature saturation curve is very well adapted to OTEC. First, the saturation pressure level of R717 corresponding to the low-grade temperature of OTEC, mechanically enables the use of this fluid with a low pressure much higher than atmospheric pressure (about 6-7 bar) preventing from introducing air in the working fluid loop.

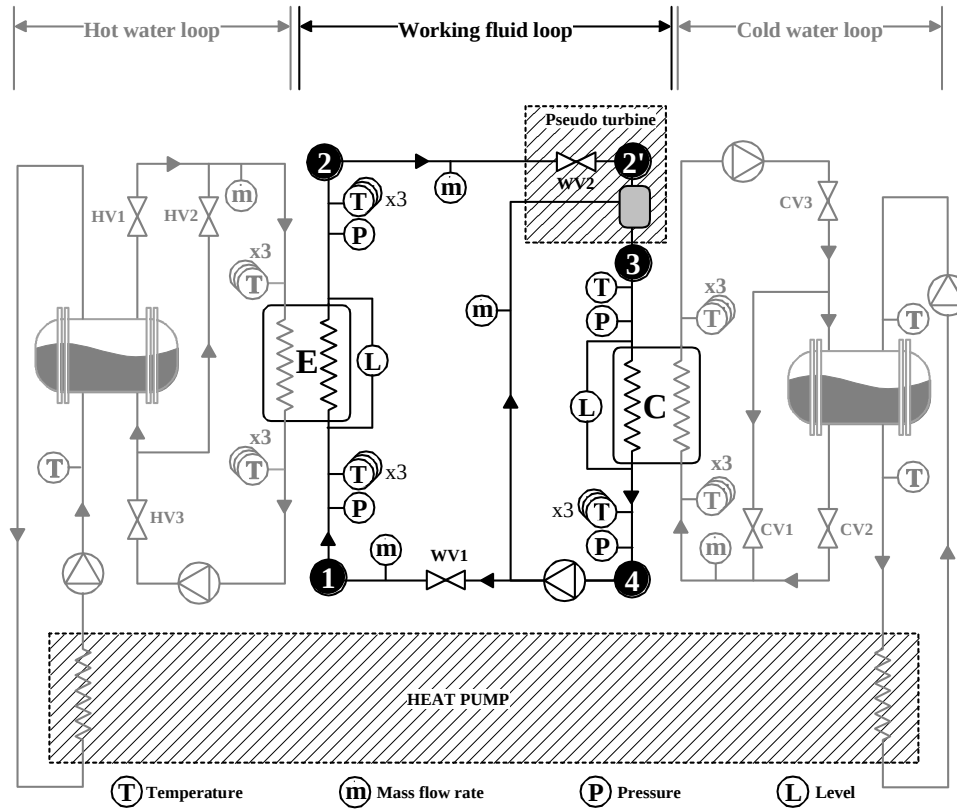


Fig. 4. Global schematic diagram of the OTEC experimental installation

Moreover, for a hot water temperature of 28 °C, the saturation pressure is not too high (around 9 bar) and does not require important technical skills. Secondly, the ammonia has a high latent heat of vaporization (~ 1370 kJ/kg at 1 bar), that allows lower mass flow rate and lower electric pump consumption than other fluids [18], [19]. In the end, ammonia is an environmental friendly fluid and do not contribute to the Ozone depletion (ODP=0) nor global warming (GWP=0) [20].

Fig. 4 shows the global scheme of the OTEC onshore prototype. The level of thermal power and therefore of gross power that are implemented here were chosen to be representative. So, a few hundred kW of exchanged thermal power at the condenser or the evaporator are then required to run the machine in order to be able to produce an amount of a dozen kW of electricity. For mainly two reasons, a turbine was not installed on this prototype. The first one concerns the final use of this test facility which is a hundred times smaller than a real OTEC plant. The objective here is mostly to test various technologies of heat exchanger and to accumulate knowledge in these particular and essential components that are submitted to a very small temperature difference. The second one is related to the lack of manufactured turbines in this low range of power. As our purpose is much more focused on heat exchanger behavior than on turbine design or development, another choice was done here to simulate a Rankine cycle despite everything. So, a pseudo-turbine, which reproduces the same thermodynamic transformation (see Fig. 4), has been designed to replace the turbine. This latter is composed of an expansion valve and a desuperheater.

First, R717 is evaporated thanks to hot water. Then, it is expanded in an isenthalpic way by the WV2 valve (point 2 to point 2' in Fig. 4). A little amount of cold liquid is injected in the warm steam in order to decrease its temperature and to bring back the fluid under the saturation curve. These transformations of expansion and desuperheating gives the same fluid properties as if it had been submitted to an isentropic expansion in a turbine. The working fluid is then condensed thanks to cold water and brought back toward the evaporator with the pump.

3.2 Heat exchangers

In terms of evaporators, four different technologies are tested on this onshore prototype: a flooded shell-and-(smooth) tubes one, a flooded shell-and-(enhanced) tubes one, a shell-and-tubes falling film one and a vertical plate one. These technologies have been chosen depending on their tubes technology, their material and their exchange area. The results presented hereafter are obtained with the flooded shell-and-tube enhanced one (Fig.

5 a). It is composed of 159 enhanced tubes of 19.05 mm diameter, distributed in two passes. The evaporator is 4.9 m long, and the shell has a diameter of 0.71 m.

The condenser is a shell-and-tube one (Fig. 5 b). It is composed of 280 smooth tubes of 19.05 mm diameter, distributed in four passes. The condenser is 3.06 m long and has a diameter of 0.5 m. The evaporator has a higher exchange surface than the condenser. The ammonia quantity needed for the evaporator is very important (see table 2) because every tubes row has to be flooded, thus allowing a much easier control of the stability of experiments. Finally, the ammonia pump is a centrifugal with a wet-rotor one, allowing a mass flow rate ranging from 0.120 to 0.450 kg/s.



Fig. 5. The flooded shell-and-tube evaporator (a), the shell-and-tube condenser (b)

As shown in the Fig. 4, each water loop is composed of two pumps and a liquid receiver. This last one allows having different water mass-flow rates in the heat pump and in the heat exchangers. For the two water loops, the water circulates through the heat exchangers with mass flow rates ranging from 20 to 50 kg/s. In the heat pump, the water circulates with a nominal mass flow rate of 47 kg/s for the hot water, and 35 kg/s for the cold water. Pumps are single-speed centrifugal types. Concerning operating conditions, the heat pump has been designed to produce hot water from 23 °C to 28 °C, and cold water around 5 °C. With 120 kg of ammonia as working fluid, it generates a refrigerating capacity ranging from 450 to 530 kW.

3.3 Control and data acquisition

As drawn on the Fig. 4, pressure and temperature sensors have been placed at the input and at the output of the main components of the ammonia loop. For the hot and cold water loop, temperature sensors have been placed at the input and at the output of each heat exchangers and just after the heat pump. In addition, three flowmeters have been placed in the working fluid loop, one for the liquid entering the evaporator, one for steam leaving the evaporator and one for the desuperheating mass flow (liquid). Two other flowmeters are situated at the input of the ammonia/water heat exchanger of each water loop. All the sensors enable to calculate the state of ammonia at each point (pressure, temperature, quality, density, enthalpy and entropy) in order to carry out energy balances and to characterize heat exchanger behavior. The main sensors specifications and their uncertainty are listed in the Table 2. All the experimental data from the sensors are registered thanks to an acquisition computer with a 1 Hz frequency.

Table 2

Specification of the sensors used in the experimental installation (see Fig. 4)

Sensor	Number	Description	Manufacturer	Type	Accuracy
T	25	Temperature sensor	Endress Hauser	Three wires PT-100	$\Delta T = \pm(0,15 + 0,002 \cdot T)$ with T in °C
P	6	Pressure transmitter	Cerabar M	0-16 bars	± 32 mbar

			Endress Hauser PROMASS F		
$\dot{m}$	5	Mass flow meter	Endress Hauser	Coriolis	$\pm 0.5 \%$
L	2	Level	Kubler	Bypass level indicator with Level transmitter	0.2 mm

4 RESULTS AND DISCUSSION

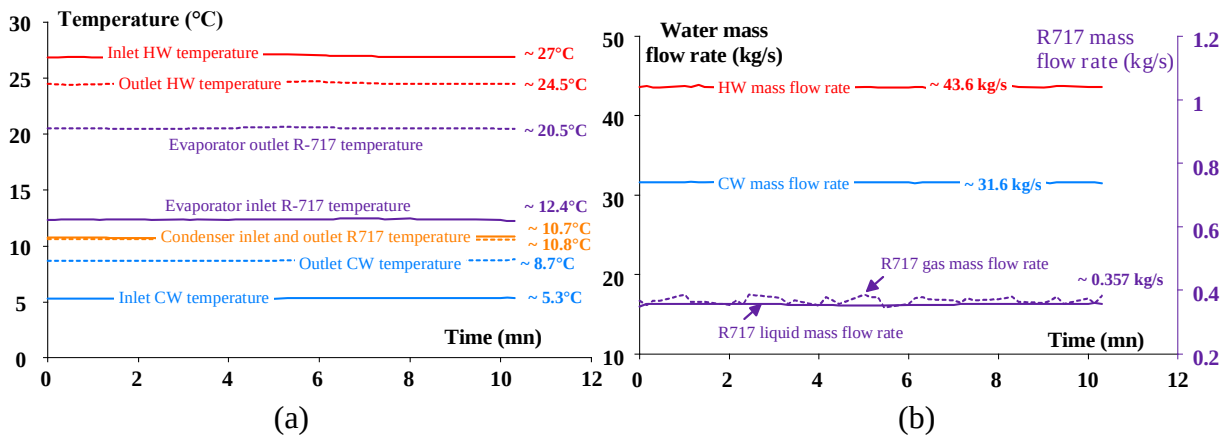
4.1 Test conditions

Several tests have been conducted on the experimental installation. As the hot seawater and the cold seawater temperatures are constant on a daily basis, steady state tests were performed. All the results that are presented in this section have been registered after a period where all variables were stable under pre-defined limits ($\pm 0.2^\circ\text{C}$ for temperature, ± 0.1 bar for pressure, ± 0.2 kg/s for Water Mass flow rate and ± 0.005 kg/s for NH_3 mass flow rate). The registration lasts a minimum of 20 minutes in order to control the stability and to obtain averaged values.

Each test is defined by five specific experimental conditions whose values are summarized in Table 3 for the presented test point. Fig. 6 presents the evolution of the main temperatures of ammonia and of heat transfer fluids, the different mass flow rates and the working fluid evaporation and condensation pressures versus time. On these three pictures, it can be clearly seen that steady state conditions are achieved. Only the gas mass flow rate of ammonia is subject to a very light variation. It is mainly due to the difficulty in measuring the steam velocity, added to the fact that lower density in the case of vapor increases the sensitivity of velocity to mass flow rate. The Fig. 6a shows that the hot water temperature undergoes a decrease of 2.5°C when passing through the heat exchanger. Then, the evaporator inlet ammonia temperature being under the saturation temperature (12.4°C), the working fluid first warms up, and then evaporates.

Table 3
Experimental conditions

Parameter	Value
HW temperature	27.0°C
HW mass flow rate	43.6 kg/s
CW temperature	5.3°C
HW mass flow rate	31.6 kg/s
R717 mass flow rate	0.357 kg/s



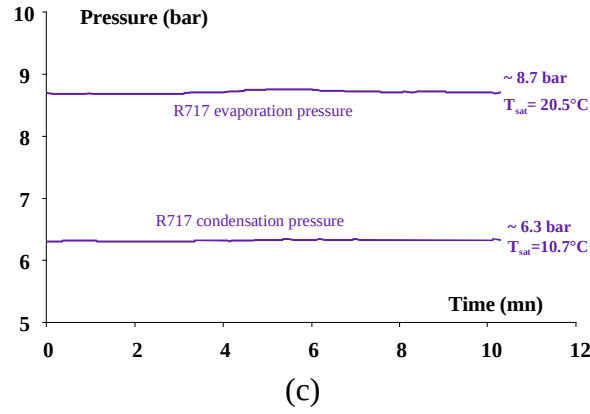


Fig. 6. Evolution of the temperatures (a), mass flow rate (b) and pressure (c) vs time

Moreover, the saturation temperature calculated from the evaporation pressure (represented on the Fig. 6c) corresponds to the outlet temperature of ammonia leaving the evaporator. This is mainly due to the evaporator which is flooded and whose ammonia level is maintained sufficiently high so as to cover all the heat exchanger tubes. This allows to say that the working fluid is not overheated in the evaporator. Contrarily to the evaporator, the inlet and outlet temperature of ammonia are pretty the same in the condenser and are almost equal to the saturation temperature calculated from the condensation pressure (Fig. 6c). This tends to demonstrate that ammonia is not completely dry when entering the condenser and that there is not any subcooling at the outlet of the condenser.

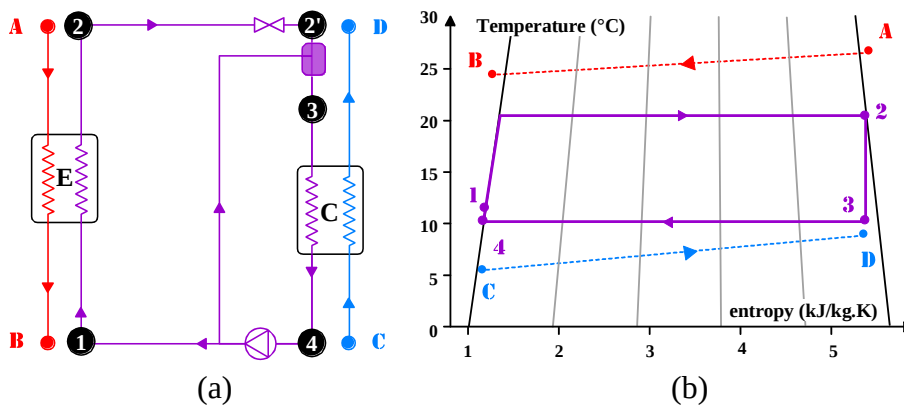


Fig. 7. Real cycle and heat transfer fluid temperature represented on a T-s diagram.

Temperature of ammonia and heat transfer fluid are represented on a T-s diagram in Fig. 7 where temperatures difference can be clearly identified.

4.2 Energetic analysis

Before all, consistency of results have been analyzed, first comparing power for each heat exchanger on ammonia side and water side. This can be seen on Fig. 8 where differences are representing about 1% for each component. On the other hand, applying the energy balance on all the components of the whole system (evaporator, condenser and pump) gives a remainder of only -0.093 kW. The relative low values of uncertainty obtained are clearly due to the high quality of the prototype in terms of design, building and measurements.

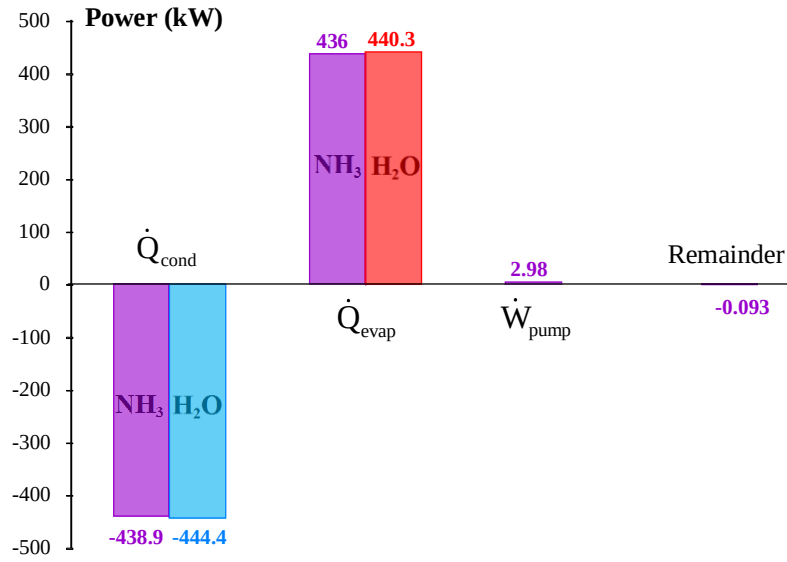


Fig. 8. Energetic balance of the system.

As explained in the beginning, this machine has been initially built to test different kind of heat exchanger, in particular evaporators. Because of the reduced size of the prototype, a real turbine able to produce the required power was not available. Nevertheless, applying first law to the actual cycle (Fig. 9a) and considering a virtual cycle as drawn in Fig. 9b, one can calculate the gross power that would have lead the equivalent Organic Rankine Cycle.

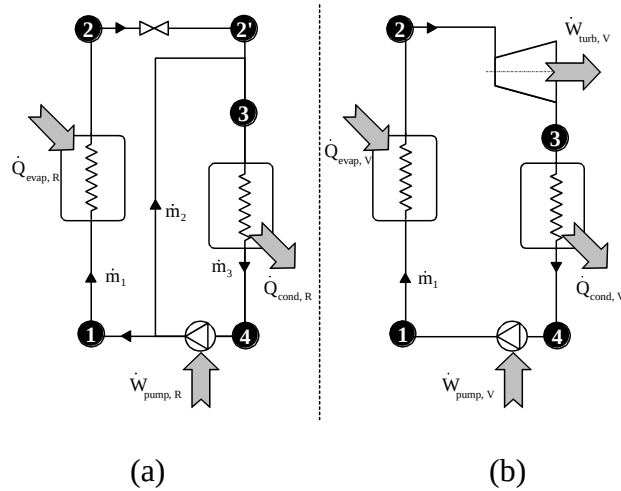


Fig. 9. Real (a) and virtual (b) cycles

Firstly, energy balance is written for the two cycles as in Eq. 1 and taking into account all heat and work contributions. Secondly, a certain number of hypothesis are made in Eq. 2 to solve the problem. Values used in the presented example are summarized in Table 4.

$$\begin{cases} \dot{W}_{pump, R} + \dot{Q}_{evap, R} + \dot{Q}_{cond, R} + \dot{Q}_{rem, R} = 0 \\ \dot{W}_{pump, V} + \dot{W}_{turb, V} + \dot{Q}_{evap, V} + \dot{Q}_{cond, V} + \dot{Q}_{rem, V} = 0 \end{cases} \quad (1)$$

$$\left\{ \begin{array}{l} \dot{Q}_{evap, V} = \dot{Q}_{evap, R} = \dot{m}_1 \cdot (h_2 - h_1) \\ \dot{W}_{pump, V} = \dot{m}_1 \cdot (h_1 - h_4) ; \dot{W}_{pump, R} = \dot{m}_3 \cdot (h_1 - h_4) \\ \dot{W}_{turb, V} = \dot{m}_1 \cdot (h_3 - h_2) \\ \dot{Q}_{cond, V} = \dot{m}_1 \cdot (h_4 - h_3) ; \dot{Q}_{cond, R} = \dot{m}_3 \cdot (h_4 - h_3) \end{array} \right. \quad (2)$$

Table 4

Experimental conditions and energetic analysis of the real and the virtual cycle

Point	Temperature (°C)	Pressure (bar)	Power (kW)	Real	Virtual
1	12.37	8.83	$\dot{Q}_{evap}$	436.04	436.04
2	20.54	8.72	$\dot{Q}_{cond}$	-438.92	-425.17
2'	15.7	6.89	$\dot{W}_{turb}$	0	-13.74
3	10.81	6.33	$\dot{W}_{pump}$	2.98	2.89
4	10.67	6.4	$\dot{Q}_{rem}$	-0.093	-0.093

In the end, the gross power produced by the turbine is 13.74 kW and the net power considering the pump consumption is 10.86 kW, giving an efficiency of approximately 2.49 %.

5 CONCLUSION

As presented in the first part, research on OTEC by mean of Organic Rankine Cycle is continuously growing in the world. NAVAL ENERGIES, a French group, intends to develop this activity in the tropical area and in particular in oversea French departments such as La Martinique (north hemisphere) [21] and La Reunion (south hemisphere). These volcanic Islands benefit from favorable geographic conditions (high depth quite close to the cost) and high surface temperature. The power level for scale 1 power plant is identified to reach approximately 100 MW. Projects in that direction are under study [22].

To achieve this goal, knowledge has to be capitalized in particular on heat exchanger that are key figures in OR Cycles. Indeed, when only 20 to 25 °C separates low and high temperature sources, a great attention has to be paid in reducing heat exchanger pinches to benefits from the maximum of this temperature difference. So, a prototype has been built from 2009 to 2011 and tested since that time. Different heat exchangers technologies are being investigated and studied. This paper intends to present the global prototype in terms of power, flow rate and temperature levels that can be obtained and also all the data acquisition system that has been built up to perform experiments as accurate as possible. An example is given here with a flooded evaporator tested under real conditions. The results analysis concludes to a quite good consistency of values in term of uncertainty (about 1% in heat flux) and of heat losses (a few tens of Watts).

Because no turbine is available in this size, a pseudo turbine has been thought off in order to replace a real turbine behavior by mean of an expansion valve and a desuperheater. It allows then driving the prototype under real conditions and also to determine the gross power that would have given a virtual turbine undergoing the same transformation. The analysis concludes to a net efficiency of 2.49% that can be considered as low but that must be compared to 7.23 %, the value obtained from Carnot analysis and with temperature sources of 27 °C and 5.3 °C. Nevertheless, this is a rather high value when compared to the other experiments carried out all over the world (see Table 1) and in the end, the thermodynamic efficiency of this cycle is approximately 35 %.

No doubt that studies have to be done on heat exchangers in order to reduce pinches and benefits as much as possible of this temperature differences. Work in that direction is actually under progress, comparing in particular heat exchanger technologies in terms of heat transfer coefficients, level of power achieved and efficiencies.

ACKNOWLEDGMENTS

The whole prototype has been designed and built by the NAVAL ENERGIES Group and funded by this latter and La Région Réunion. The Prototype is hosted by the IUT and maintenance is realized by SOREFROID, a local society. The authors gratefully acknowledge the support provided by all these contributors and by Jerome Vigneron, our technician in charge of most tests.

NOMENCLATURE

P	pressure (Pa)	<i>Subscripts</i>	
h	specific enthalpy (J/kg)	R	real
s	specific entropy (J/kg.K)	V	virtual
$\dot{W}$	mechanical power (W)	pump	pump
$\dot{Q}$	heat exchanged (W)	turb	turbine
$\dot{m}$	mass flow rate (kg/s)	evap	evaporator
T	temperature (K)	cond	condenser

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