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Experimental evaluation of an autonomous low-temperature solar Rankine cycle system for reverse osmosis desalination

D. Manolakos[a]*, G. Papadakis[a], S. Kyritsis[a], K. Bouzianas[b]

[a]*Agricultural University of Athens, Department of Agricultural Engineering, Farm Structures Laboratory, 75, Iera Odos street 11855 Athens, Greece*
Tel. +30 (210) 5294033; Fax +30 (210) 5294023; email: dman@aua.gr
[b]*Hellas Energy K. Bouzianas P. Moschovitis & Co., 10, Saint George Square 11257 Athens, Greece*

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Abstract

The present paper regards the experimental evaluation of the performance under laboratory conditions, of a low-temperature solar organic Rankine cycle system for reverse osmosis (RO) desalination. The operation principle of the system is given briefly below. Thermal energy produced by a solar collectors' array evaporates the refrigerant (HFC-134a) in the evaporator surface of Rankine engine. The super-heated vapour is driven to the expander where the generated mechanical work produced from expansion drives the RO unit high-pressure pump. The vapour at the expander's outlet is directed to the condenser and condensates. The saturated liquid at the condenser outlet is then pressurised using a piston-diaphragm pump and the thermodynamic cycle is repeated. The design of the system has already been done and presented in [1]. For manufacturing the prototype system, the design results have been used. In this paper the experimental results derived from the laboratory tests are illustrated. The next research step is the evaluation of the system performance on site, under real climatic conditions. The main difference of the above two experimental cases is that in laboratory tests the thermal energy source used is an electric heater of 100 kW, capable to operate at partial thermal load, which substitutes and simulates the behaviour of solar collectors.

Keywords: Solar energy; Rankine cycle; RO desalination

*Corresponding author.

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1. Description of the system

The low-temperature solar organic Rankine cycle (LTSORC) system for RO desalination consists of the following sub-systems and components (Fig. 1):

1) High efficiency vacuum tube solar collector array
2) Circulator
3) Evaporator
4) Condenser
5) Expanders
6) HFC-134a pump
7) RO unit
8) Seawater reservoir
9) Fresh water reservoir
10) RO energy recovery system

The system operation is described briefly below.

Thermal energy produced by the solar array (1) preheats and evaporates the refrigerant (HFC-134a) in the preheater–evaporator surfaces (3). The super-heated vapour is driven to the expanders (5) where the generated mechanical work drives the RO high-pressure pump, cooling water pump and HFC-134a pump (6). The expanders are in reverse running scroll type compressors. The sub-cooled liquid at the condenser outlet is pressurised by the piston-diaphragm pump (6). An energy

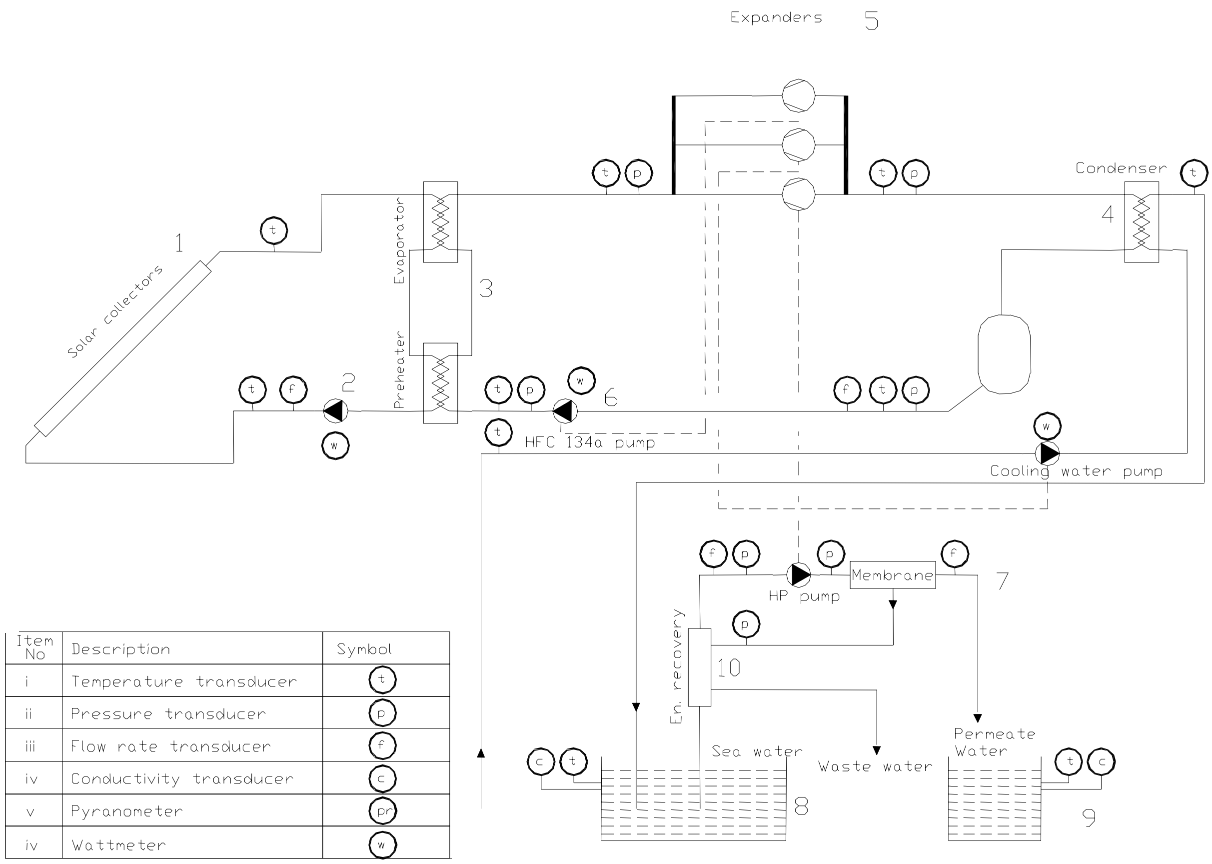

Fig. 1. Schematic representation of the system.

recovery system (10) is coupled to the RO unit thus declining significantly the energy consumption. It consists of two Dunfoss APP pumps connected in the same shaft. The excess energy needed for the desalination process is supplied by one of the expanders through a pulley. The capacity of the RO unit is 280 l/h. Fig. 2 presents the Rankine engine.

2. Subsystems and components

2.1. Solar collectors array

The SOLAMAX direct flow evacuated solar energy collector manufactured by Thermomax Ltd. has been considered. In Table 1 the basic data of the collectors array are presented.

Fig. 2. The Rankine engine.

2.2. Solar collector circulation pump and cooling pump

Table 2 summarizes the technical characteristics of both identical pumps.

2.3. Heat exchangers

2.3.1. Preheater and evaporator

Table 3 provides the basic technical characteristics of preheater and evaporator.

2.3.2. Condenser

The selected condenser characteristics are presented in Table 4.

2.5. HFC-134a pump

Table 5 presents the Freon pump characteristics.

Table 1
Characteristics of the solar collector array

Manufacturer	Thermomax Ltd.
Type	SOLAMAX
No. of tubes/collector	30
Capacity, l	6
No. of collectors	56
No. of collectors connected in series	2
Slope, º	40
Maximum flow rate, l/min	196.8
Inlet temperature, ºC	70
Outlet temperature, ºC	77.3

Table 2
Collector circulator characteristics

Pump type	Centrifugal-1 stage
Flow rate, l/min	200
Head, m	18
Efficiency, %	75
Nominal power, kW	1.85
Rotation speed, rpm	2850

Table 3
Preheater and evaporator characteristics

	Preheater	Evaporator
Manufacturer	CIAT	CIAT
Model	EXL-1440	EXL-1440
Dimensions, L×w×h, mm	129×265×528	129×265×528
Duty, kW	40	65
Fluids, hot/cold	Water/HFC-134a	Water/HFC-134a
Pressure drop, hot/cold, mmWG	237/2140	237/2140
Type	Counter flow	Counter flow
Flow rate hot/cold, m^3/h, kg/h	11 m^3/h	11 m^3/h

Table 4
Condenser characteristics

Manufacturer	CIAT	
Model	EXL-1440	
Dimensions L×W×H, mm	129×265×528	
Duty, kW	100	
	Tube side	Shell side
Fluid	Seawater	HFC-134a
Inlet/outlet temperature, ºC	25/33	—
Discharge temperature, ºC	—	75
Condensing temperature, ºC	—	34.8
Pressure drop, mmWG	399	—
Flow rate	11 m^3/h	1700 kg/h

Table 5
HFC-134a pump characteristics

Type	Piston diaphragm
Flow rate, l/min	33
Suction pressure, bar	9
Discharge pressure, bar	22
Motor power, kW	1.8
RPM	441

2.6. Expander

The Sanden TRS 105 model compressor is used after suitable modifications.

2.7. RO-desalination plant with energy recovery system

The RO system consists of four membranes connected two by two in series and series group in parallel. Fig. 3 illustrates schematically the layout of the RO system. The system has four 1000 psi membrane vessels with 2.5″ diameter made of fiber glass. In each membrane vessel there is one Filmtec membrane, type SW30-2540.

A Danfoss pump APP2.2 is used as a high-pressure pump. To form the energy recovery unit, a Danfoss pump APP1.8 in reverse running is used. Both pumps are made of duplex stainless steel and they are fixed in the same shaft, while a pulley is adjusted in between. The diameter of the pulley of the energy recovery unit is the same as that of the expander. Thus a transformation ratio 1:1 is achieved. The high pressure pump with energy recovery needs approximately 1 kW as input power if seawater with salinity 40,000 ppm at a temperature of 20ºC is considered. This excess amount of energy needed is provided by the expander which is coupled with a belt with the energy recovery unit.

When the high-pressure pump coupled with a reverse running pump runs at 3000 rpm, the flow rate of the APP2.2 pump is 2 m^3/h, and the flow rate of APP1.8 pump is 1.7 m^3/h.

With the above configuration, the product flow rate is expected to be approximately 280 l/h. Fig. 4 illustrates the RO unit while Fig. 5 the high pressure pump with the energy recovery system.

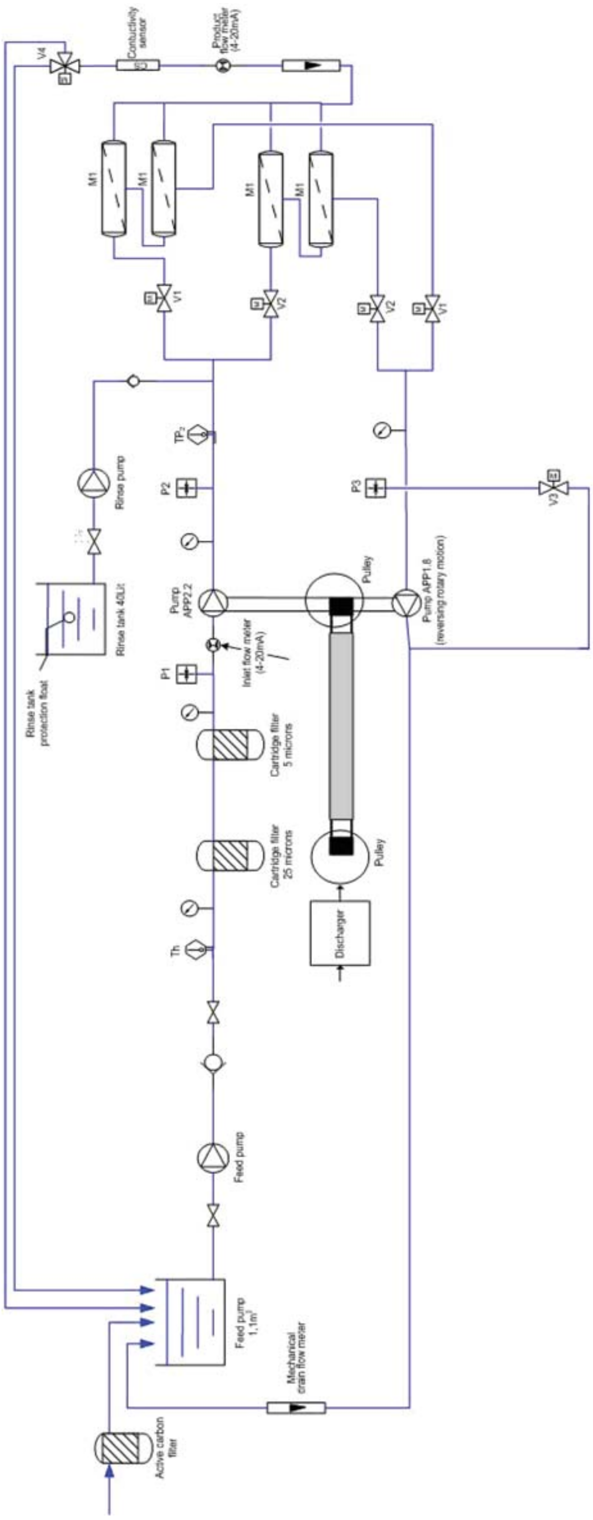

Fig. 3. Layout of the RO unit with the energy recovery system.

Fig. 4. The RO unit.

3. Laboratory test results

In this section the first results derived from the laboratory tests of the system are presented. The first research step was to asses the behavior of scroll type expander and the Rankine engine as well. Scroll type expander is the component of the system generating mechanical energy to supply the high-pressure pump of the RO unit. To this context, the expander was connected first to an electric brake, consisting of a 7.5 HP motor controlled by a special inverter (with vector speed mode) capable to keep constant a pre-selected rotation speed. The measurements were realized keeping the inlet–outlet pressure difference of the expander *DP*, constant at a *DP* = 13.0 bar at various rotation speed of the expander shaft. The hot water temperature of the electric heater varied from 65 to 70°C.

Fig. 5. The energy recovery unit.

3.1. Instrumentation and recorded data

In each rotation speed of the expander a series of data was recorded. The data of concern was the following: M_{exp} — torque in the expander shaft, Nm (using a torque meter); n_{exp} — rotation speed of the expander shaft, rpm (using a speedometer); P_{in} — pressure at the inlet of the expander, bar (using pressure transducer); P_{out} — pressure at the outlet of the expander, bar (using pressure transducer); n_{fr} — rotation speed of the Freon pump, rpm (using speedometer); T_1 — temperature of the super-heated vapor at the inlet of the expander, °C (using a thermometer); T_2 — temperature of the super-heated vapor at the outlet of the expander, °C (using a thermometer); T_3 — condensation temperature, °C (using a thermometer); T_w — thermal water temperature, °C (using a thermometer).

From the above gathered variables it is possible to calculate a set of results able to give a global assessment of both scroll expander and the Rankine engine behavior at specific operation conditions.

3.2. Data processing

In view to assess the system behavior, the following quantities are calculated from the measured variables (sub-section 3.1).

3.2.1. Power produced by the expander

The power produced by the expander, W_{exp}

represents the total power produced by the system and is given by the formula below:

$$W_{exp} = M_{exp}\left(\pi n_{exp}/30\right) \quad (1)$$

The continuous line in Fig. 6 presents the variation of torque as a function of the rotation speed, while the continuous line in Fig. 7 shows the power produced by the expander, and thus by the system, vs. rotation speed.

3.2.2. Power absorbed by the Freon pump

In general the equation below can be applied to calculate the power absorbed by the Freon pump, W_{fp}:

$$W_{fp} = Q_f\left(P_{in} - P_{out}\right) \quad (2)$$

The flow rate Q_f can be calculated from the rotation speed of the Freon pump n_{fr}, given that

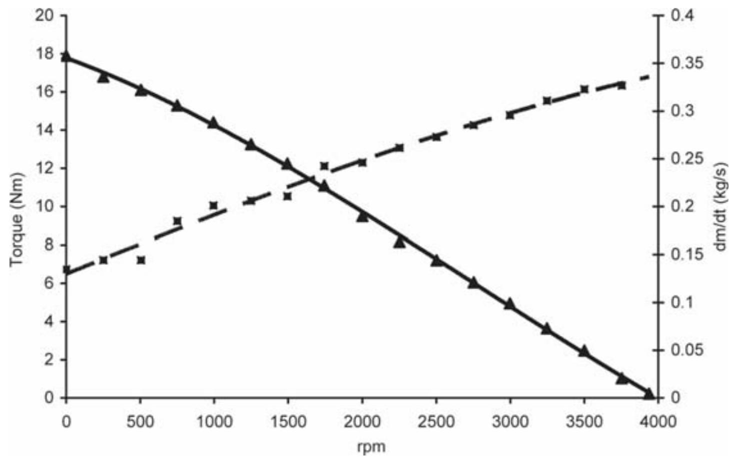

Fig. 6. Variation of torque (continuous line) and mass flow of HFC-134a (dashed line) as a function of the rotation speed.

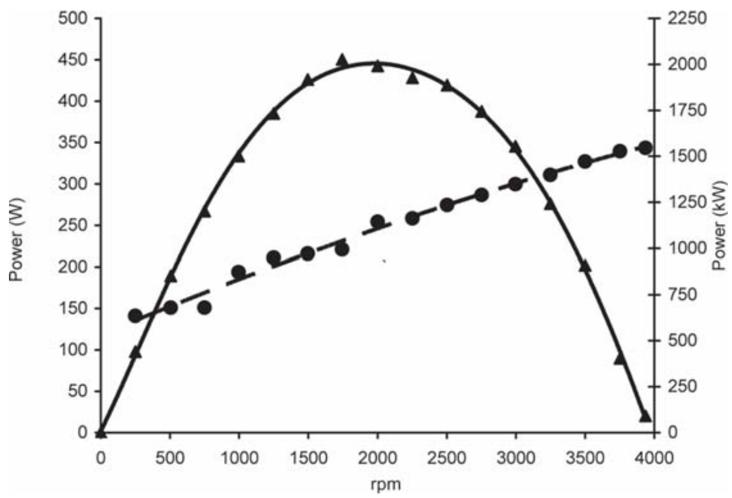

Fig. 7. Variation of power generated by the expander (continuous line, right y axis) and power absorbed by the Freon pump (dashed line, left y axis) as a function of the rotation speed.

the Freon flow rate pumped per revolution is equal to 1.247×10^{-6} m^3/s/rev.

To calculate the refrigerant mass flow m_f (kg/s), it is just needed to calculate the specific volume v (m^3/kg) at the condensation temperature. The product $Q_f v$ equals to the mass flow. The dashed line in Fig. 6 illustrates the variation of mass flow as a function of the rotation speed, while the dashed line in Fig. 7 shows the power absorbed by the Freon pump vs. the rotation speed.

3.2.3. Efficiency of the expander

The expander efficiency can be derived from the following equations:

$$\eta_{exp} = W_{exp} / W_{in} \tag{3}$$

$$W_{in} = m_f \left(h_{in} - h_{out} \right) \tag{4}$$

where h_{in} is the specific enthalpy of the superheated vapor at expander inlet and h_{out} is the same property at the expander outlet.

h_{in} and h_{out} can be easily calculated from the pair of temperature and pressure in the exact thermodynamic state. In Fig. 8 the expander efficiency as a function of the rotation speed is presented (continuous line).

3.2.4. Overall efficiency of the system

The overall efficiency can be calculated from the following formula:

$$\eta_t = \left(W_{exp} - W_{fp} \right) / W_t \tag{5}$$

W_t reflects to the total power generated by the system. Considering that the total amount of thermal energy produced is exploiting by the Rankine engine this quantity can be calculated as:

$$W_t = m_f \left(h_{in} - h_{con} \right) \tag{6}$$

where h_{con} is the value of specific enthalpy at condensation temperature.

The dashed line in Fig. 8 shows the variation of the overall system efficiency as a function of the rotation speed of the expander. Fig. 9 presents the power produced by the system as a function of the rotation speed.

4. Conclusions

From the elaboration of the results derived from the laboratory experiments some very important conclusions can be concluded. More specifically:

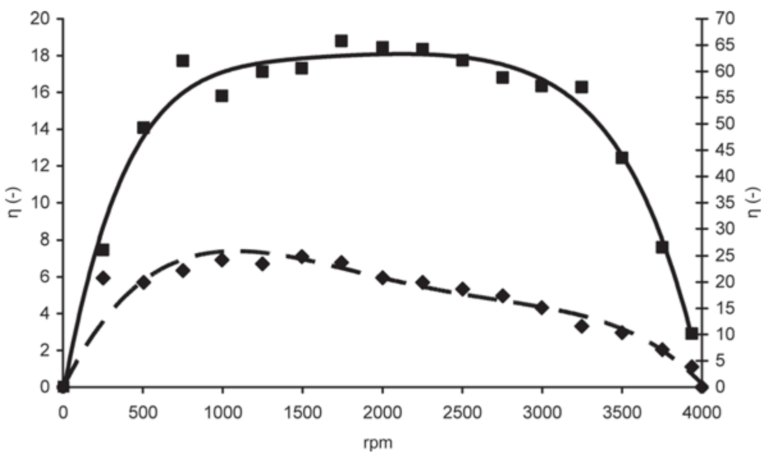

Fig. 8. Variation of the expander efficiency (continuous line, right axis) and overall system efficiency (dashed line, left axis) as a function of the rotation speed.

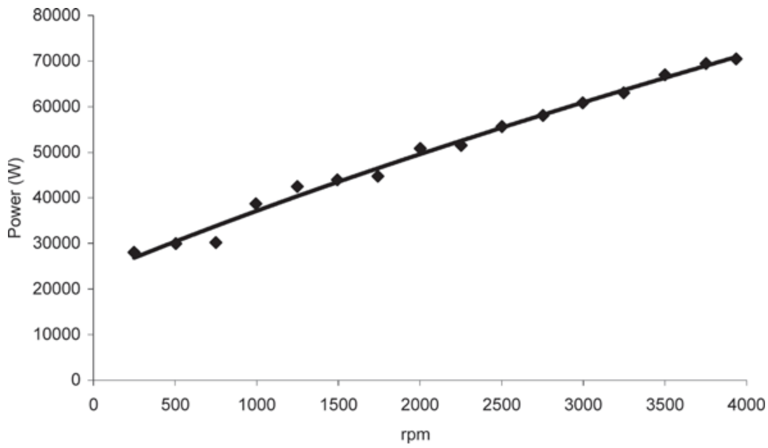

Fig. 9. The overall thermal power generated by the system as a function of the rotation speed.

- The variation of torque as a function of the rotation speed has a maximum value of 18 Nm at 0 rpm.
- The expander has an overall maximum efficiency of about 65%. This efficiency seems to be preserved constant at a wide operation range, varying from approximately 1000 to 3000 rpm. The observed plateau of this efficiency implies a high performance at a considerably wide range.
- The maximum power generation is about 2.05 kW achieved near 2000 rpm. This power is quiet enough to drive the RO unit and assure the initially planned fresh water production and it indicates that the system should be regulated in such a way as to operate around this value of the rotation speed.
- The power consumed by the Freon pump is considerably low and at 2000 rpm reaches some 250 W.
- The maximum overall system efficiency is 4%. Although this value of efficiency seems considerably low at first sight, it is high enough taking into account the selected range of temperature of the refrigerant. In principle, the efficiency of Rankine cycle depends on the evaporation–condensation temperature difference. The higher the difference, the higher the efficiency. The theoretical efficiency of the ideal Rankine cycle at the experimental operation conditions is not above 10%, while that of Carnot, which is equivalent to the maximum possible, is not more than 14%.

All the above results refer to a constant pressure difference of 13 bar. The system behaviour changes significantly with the variation of this variable.

Acknowledgements

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References

[1] D. Manolakos, G. Papadakis, E.Sh. Mohamed, S. Kyritsis and K. Bouzianas, Design of an autonomous low-temperature solar rankine cycle system for reverse osmosis desalination, Desalination, 183 (2005) 73–80.