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Power generation using waste heat recovery by organic Rankine cycle in oil and gas sector in Egypt: A case study

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A R T I C L E I N F O

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A B S T R A C T

ORC (organic Rankine cycle) is a promising technology for conversion of heat into useful work. This study utilizes the ORC in an existing gas treatment plant in Egypt, as a case study, to recover the waste heat and convert it into electricity. A simulation model using Aspen HYSYS v7.1 has been built up for the case study. Two different cycles, the basic and the regenerative cycles, have been studied. Various working fluids have been investigated using different parameters such as net work produced, efficiency, volumetric flow rate and the irreversibility. To be more confident about the best working fluid, a capital cost and profitability analysis has been performed for the most two promising working fluids. The simulation has shown that regenerative cycle using either benzene or cyclohexane is the most promising choice. However, the capital cost and profitability study has shown that benzene is more suitable as working fluid than cyclohexane. Finally, an optimization study on the parameters indicates that the turbo expander inlet pressure of 4.1 MPa and temperature of 290 °C–300 °C are the most appropriate working conditions.

1. Introduction

In recent years, energy security debates and emission regulations have pushed the whole world to explore alternative forms of energy. Whether it is solar, wind, geothermal energy, fuel cells or bio-fuel, something must be done to relinquish our dependency on fossil fuels. Therefore, it is of the utmost importance to explore innovative measures to improve the fuel economy and efficiency, and simultaneously decrease the emissions [1,2].

Many industrial processes require large quantities of thermal energy, much of which is eventually exhausted to the environment, either to the atmosphere or to water. Recovering this waste heat represents the largest opportunity for reducing industrial energy consumption [3]. Industrial waste heat refers to energy that is generated in industrial processes without being put to practical use. Sources of waste heat include hot combustion gases discharged to the atmosphere, heated products exiting industrial processes, and heat transfer from hot equipment surfaces. The exact quantity of industrial waste heat is poorly quantified, but various studies have estimated that as much as 20–50% of industrial energy consumption is ultimately discharged as waste heat. While some waste heat losses from industrial processes are inevitable, facilities can reduce these losses by improving equipment efficiency or installing waste heat recovery technologies. Waste heat recovery entails capturing and reusing the waste heat in industrial processes for heating or for generating mechanical or electrical work. Examples for using the waste heat include generating electricity, preheating combustion air, preheating furnace loads, absorption cooling, and space heating [4]. More than 50% of the total heat generated in industry is the low grade heat in the range of 100–220 °C. The ability to recover this type of energy and convert it to electricity can reduce fossil fuel consumption and alleviate many environmental problems [5].

In recent years, organic Rankine cycle (ORC) has become a field of intense research and appears as a promising technology for conversion of low grade heat into useful work or electricity [6–8]. Unlike in the steam power cycle, where vapor steam is the working fluid, organic Rankine cycles employ refrigerants or hydrocarbons [9–11]. The economics of a Rankine system is strictly linked to the thermodynamic properties of the working fluid. A bad choice could lead to a low efficient and an expensive plant. Properties of a good fluid are: low specific volumes, high efficiency, moderate pressures in the heat exchangers, low cost, low toxicity, low ODP (Ozone Depletion Potential) and low GWP (Global Worming Potential) among others [12]. The effects of the working fluids types and thermal physical properties on ORC performance have been reviewed recently by Bao and Zhao [13]. In addition, they summarized pure and mixed working fluids researches including the

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discussion of the working fluids screening results, the comparison of the pure and mixed working fluids and the advantages and disadvantages of ORCs mixtures.

More attention was paid to the technology in the last few years. Many researches focused on working fluid selection and parametric optimization of ORC from different aspects. Some examples are reviewed below.

Tchanche et al. [12] assessed comparatively the theoretical performances as well as the thermodynamic and environmental properties of few fluids for use in low temperature solar ORC systems. Efficiencies, volume flow rate, mass flow rate, pressure ratio, toxicity, flammability, ODP and GWP were used for comparison. Of the 20 fluids investigated, R134a appeared as the most suitable for small scale solar applications. R152a, R600a, R600 and R290 offered attractive performances but needed safety precautions, owing to their flammability.

Wang et al. [14] studied the effects of key thermodynamic design parameters for ORC, including turbine inlet pressure, turbine inlet temperature, pinch temperature difference and approach temperature difference in heat recovery vapor generator on the net power output and surface areas of both the vapor generator and the condenser. Working fluids used were R123, R245fa and isobutene. By parametric optimization, the ORC system with isobutane showed the best system performance compared to other fluids investigated.

Wang et al. [15] analyzed the performance of different working fluids operating in specific regions using a thermodynamic model built in Matlab together with REFPROP database. Nine different pure organic working fluids were investigated. The outcomes indicated that R11, R141b, R113 and R123 manifest slightly higher thermodynamic performances than others.

Rayegan and Tao [16] developed a procedure to compare capabilities of working fluids when they are employed in solar Rankine cycles with similar working conditions. A procedure to compare ORC working fluids based on their molecular components, temperature-entropy diagram and fluid effects on the thermal efficiency, net power generated, vapor expansion ratio, and exergy efficiency of the Rankine cycles has been proposed. They recognized fluids with the best cycle performance in two different temperature levels within two different categories of fluids: refrigerants and non-refrigerants. They suggested 11 fluids (acetone, benzene, butane, cis-butene, cyclohexane, Difluoromethane, isobutene, iso-pentane, R-245ca, R-245fa, and trans-butene) to be employed in solar ORCs that use low or medium temperature solar collectors. They indicated that the use of the regenerative ORC instead of the basic cycle reduces the irreversibility of a solar ORC. At the two temperature levels studied, higher molecular complexity results in more effective regenerative cycles except for cyclo-hydrocarbons.

Lai et al. [17] investigated potential pure working fluids for high temperature ORC processes using the molecular based equations of state BACKONE and PC-SAFT. The fluids considered were alkanes, linear siloxanes and aromates. The results showed that cyclopentane was the best working fluid for all cases studied.

Saleh et al. [18] performed a thermodynamic screening of 31 pure component working fluids for ORCs at low temperatures between 30 °C and 100 °C using BACKONE equation of state. The fluids were alkanes, fluorinated alkanes, ethers and fluorinated. The highest values for thermal efficiency were obtained for the high boiling substances with overhanging saturated vapor line in subcritical processes.

Finally, the evaluation of the environmental impact of ORC was presented by Liu et al. [19]. They applied life-cycle assessment to an ORC power-plant for waste-heat-recovery. The life-cycle analysis was divided into construction, operation and decommissioning phases using 7 different working fluids. The GWP, acidification potential, eutrophication potential, human toxicity potential, solid waste potential, and soot and dust potentials were investigated. The results showed that the construction phase contributes mostly to the GWP and eutrophication potential. GWP was the most serious environmental impact followed by human toxicity potential among all the environmental impacts.

Currently, the heat recovery is a method rarely implemented in the oil & gas industry. The upstream treatment facilities are full of chimneys discharging high temperature gases; thus, the associated thermal power is lost. Waste heat could be recovered for the generation of thermal, mechanical or electrical energy.

The purpose of this paper is to present the opportunities to recover waste energy and use it for producing electricity. This is presented by implementing the organic Rankine cycle for a case study in the oil and gas industry in Egypt.

2. Methodology

An existing upstream gas treatment facilities “Hapy Plant” located in Port Said area, Egypt treats natural gas extracted from a number of offshore wells. Natural gas is sent through two sea lines to the plant:

- Low pressure: 0.55 MPa, with daily flow rate of about 650,000 Sm^3
- High pressure: 5 MPa, with daily flow rate of about 1,250,000 Sm^3

Gas compression from 0.55 MPa to the network delivery pressure (5–5.5 MPa) is obtained via turbo-machines. The plant accommodates two trains, one on duty and one on stand-by, consisting of a two stages intercooled compressor driven by GE PGT10 DLN gas turbine. Once compressed, natural gas coming from low pressure sea line is collected with high pressure gas and sent to the treatment facilities.

Table 1 shows temperature and flow rate of gas turbine exhaust gases in design conditions (20 °C, 100% load) and at actual conditions (10 °C, 70% load). The gas turbine exhausts temperature is about 499 °C at design conditions, but actually the exhausts temperature measured in the chimney is about 416 °C. Finally, the exhaust flow is typically between 39 kg/s and 44 kg/s, depending on ambient temperature. Average annual electric power consumption in the gas treatment plant is of 3000 MWh. Annual power consumption profile is quite flat.

Design and operative data of the plant have been collected, and a process simulation model using Aspen HYSYS software V7.1 has been developed and validated with field data. The model has been modified to evaluate performances of each working fluid for the organic Rankine cycle to determine the most suitable working fluid. The model also has been evaluated to determine the best operating condition to maximize the output power. Finally, an economical analysis has been conducted to evaluate the feasibility of this improvement.

Table 1
Exhaust gas flow rate and temperature at actual and design conditions.

	Actual	Design
Load	70%	100%
Ambient temp (°C)	10	20
Exhaust temp (°C)	416	499
Exhaust flow (kg/s)	43.7	41

2.1. Simulation model

Two models have been built and evaluated. The first one is the Basic Cycle (Fig. 1A) and the other one is the Regenerative Cycle (Fig. 1B). The two cycles are similar and the only difference is that the Regenerative cycle contains one more heat exchanger (HE-003).

The Basic cycle consists of:

1 Waste Heat Oil Heat Exchanger (HE-001).
2 Evaporator (HE-002).
3 Thermal Oil Pump (P-001).
4 Turbo Expander (K-100).
5 Air Cooler (AC-001).
6 Working Fluid Pump (P-002).

The Regenerative Cycle consists of:

1 Waste Heat Oil Heat Exchanger (HE-001).
2 Evaporator (HE-002).
3 Regenerator (HE-003).
4 Thermal Oil Pump (P-001).
5 Turbo Expander (K-100).
6 Air Cooler (AC-001).
7 Working Fluid Pump (P-002).

Table 2 contains the parameters used in building the simulation model [16,20].

2.2. Preselected working fluids

Starting from the previous work in the literature [16], the REFPROP 8.0 database, which contains 117 organic fluids, was used as a reference. Table 3 shows the final list of fluids to be studied. The choice of the fluids from the database was according to the following criteria:

1 Chlorine-containing fluids that are not ozone-safe fluids were discarded.
2 Dry or isentropic working fluids are more appropriate for ORC systems. This is because dry or isentropic fluids are in the superheated zone after the isentropic expansion. This eliminates the concern of liquid droplets existence in the turbo expander.
3 Since air cooler was used for condensing the working fluids (maximum ambient air temperature is 40 °C with minimum approach 10 °C – 15 °C), other fluids with normal boiling temperatures lower than 55 °C, were eliminated.

The heat absorption process in an ORC may end in a saturated vapor state or superheated vapor state. Generally, superheating in an ORC increases the thermal efficiency of the cycle with a very low

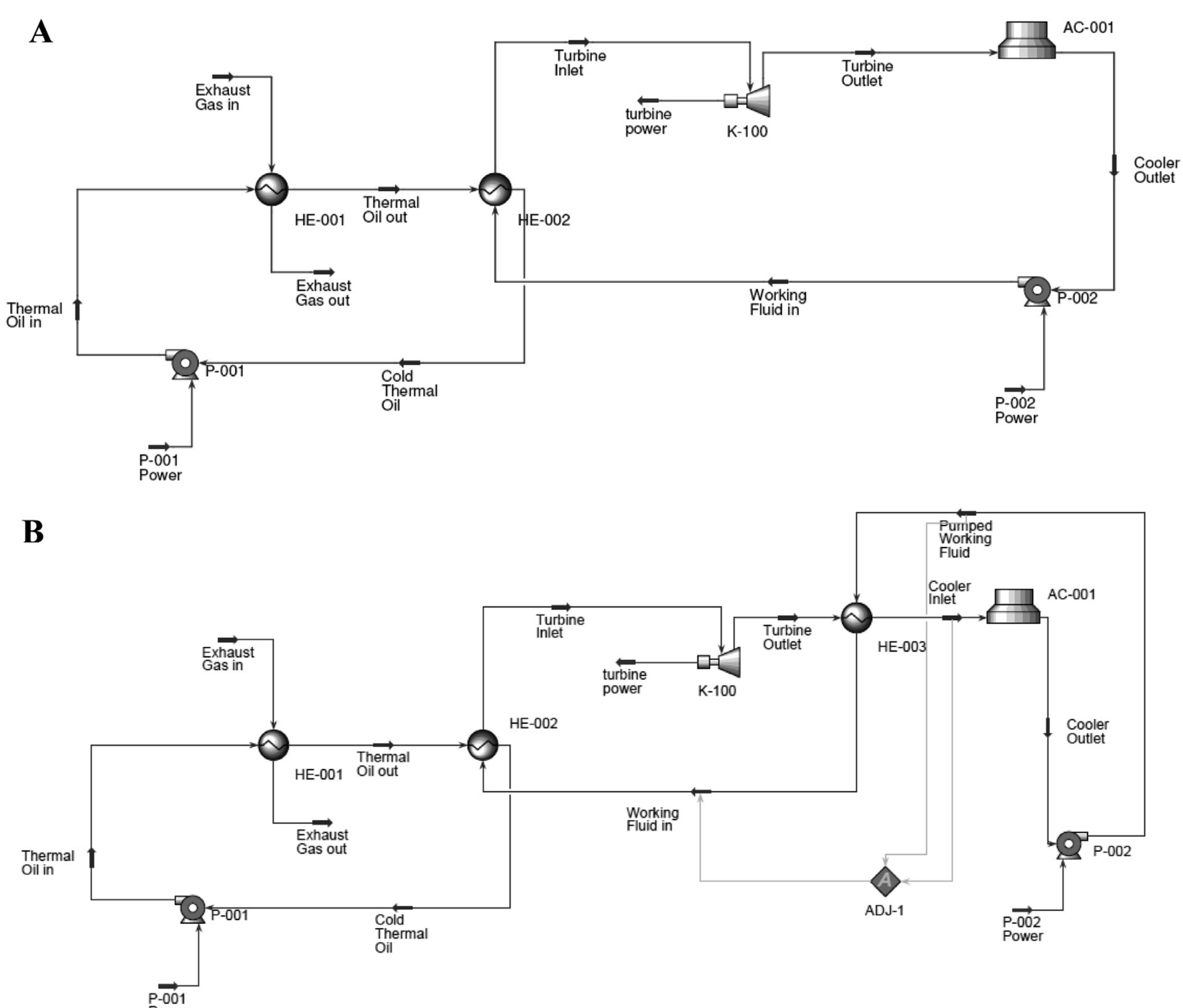

Fig. 1. A-Basic cycle Configuration; B-Regenerative cycle Configuration.

Table 2
Parameters used in building the simulation model.

Parameter	Value/constrain
Property package	Peng–Robinson
Heat exchanger efficiency	90%
Minimum approach for heat exchangers	10 °C
Heat exchanger pressure drop	25 kPa
Pump efficiency	80%
Turbo expander adiabatic efficiency	80%
Air cooler efficiency	90%
Air cooler pressure drop	25 kPa
Air cooler minimum working pressure	5 kPa abs
Ambient air temperature	25 °C
Exhaust gas inlet temperature	420 °C
Exhaust gas outlet temperature	≥150 °C
Exhaust gas flow rate	44 kg/s

slope but decreases the exergy efficiency of the cycle [16]. Consequently, superheated cycles are never recommended unless in order to gain more power at the expense of losing efficiency. Because of these reasons, the saturated Rankine cycle was investigated in this study instead of the superheated cycle, except for the water for sake of comparison. All the working fluids were preselected as dry or isentropic fluids, except water which is a wet fluid. Therefore, in the case of using water as working fluid, the superheating is required to prevent liquid drops existence inside the turbo expander.

As the higher pressure ratio across the turbo expander leads to a higher power output, it is preferred to expand higher and lower pressure limits of the cycle. However, there are always some practical restrictions. Near critical pressure, small changes in temperature are equivalent to large changes in pressure, which makes the system unstable. Therefore, a reasonable distance between the higher limit of the cycle and the critical point of the fluid has to be considered [16]. However, there is no unique interpretation of the reasonable distance from the critical point. Drescher and Bruggemann [20] suggested setting the higher pressure limit of the cycle 0.1 MPa lower than critical pressure. Delgadeo-Torres and Garcia-Rodriguez [21] considered the higher temperature of the cycle to be 10 °C − 15 °C lower than critical temperature. In the present study, these two criteria have been followed, the main criteria is setting the higher pressure limit of the cycle 0.1 MPa lower than critical pressure and checking the higher temperature of the cycle. If the difference between the higher temperature of the cycle and the critical temperature is equal or higher than 15 °C, this pressure limit is kept otherwise the second criteria is followed.

3. Analysis

The first law of thermodynamics is applied to the individual components of the cycle and the second law of thermodynamics is applied to the whole cycle to determine heat transfer, work input and output, and irreversibility of the cycle. The first law of thermodynamics for steady flow processes when potential and kinetic energy changes are negligible can be expressed as:

$$Q^{\circ} - P = m^{\circ}(h_{out} - h_{in}) \tag{1}$$

where

Q°: Heat transfer rate
P: Power exchange
m°: Mass flow rate [kg/s]
h_{out}: Outgoing specific enthalpy [kJ/kg]
h_{in}: Incoming specific enthalpy [kJ/kg]

Table 3
The preselected fluid to be evaluated.

Working fluid	P_{cr} (MPa)	T_{cr} (°C)	Max P_{eva} (MPa)	Max T_{eva} (°C)	Normal boiling temp. (°C)
Benzene	4.894	288.9	4.067	274	80.17
Cyclohexane	4.075	280.5	3.665	272	80.73
Decane	2.103	344.5	1.896	337	174.1
Dodecane	1.817	384.9	1.723	381	216.3
Heptane	2.736	267	2.41	258	98.43
Hexane	3.034	234.7	2.68	226	68.73
Isohexane	3.04	224.5	2.682	216	60.26
Nonane	2.281	321.4	2.059	314	150.8
Octane	2.497	296.2	2.2	287	125.7
Toluene	4.126	318.6	3.576	307	110.6
p-Xylene	3.5	343	3.0	330	139
Water	22.1	374.1	18.8	360	100

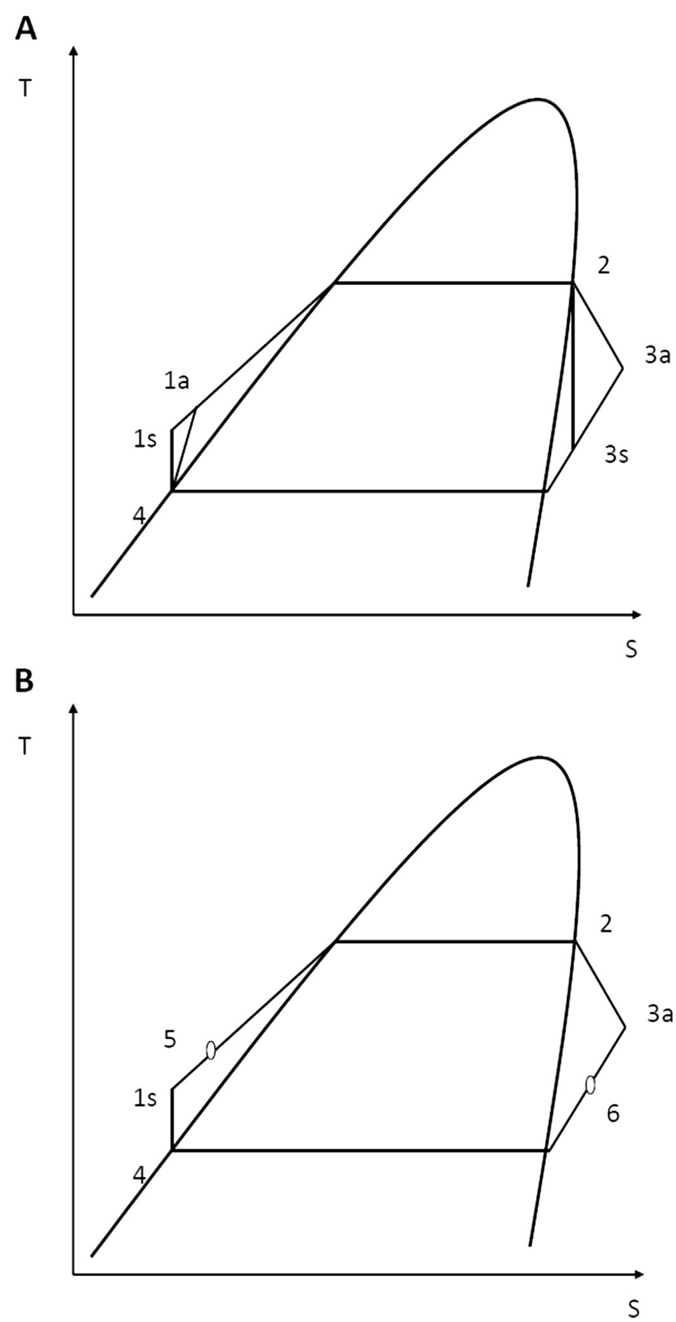

Fig. 2. A-Actual saturated basic ORC; B-Actual saturated regenerative ORC.

The irreversibility rate for a cycle in steady flow condition can be expressed as:

$$I = T_0 S_{gen} = T_0 m^{\circ} \sum \left(q_j / T_j \right) = \mathrm{Ex_{in}} - \mathrm{Ex_{out}} \tag{2}$$

where

I: Irreversibility
S_{gen}: The entropy generation rate [kW/K]
$\mathrm{Ex_{in}}$: Incoming exergy flows
$\mathrm{Ex_{out}}$: Outgoing exergy flows
q_j: The specific heat of the jth component of the cycle [kJ/kg]
T_j: The temperature of the jth component of the cycle [K]

3.1. Basic cycle

Fig. 2A shows a general representation of the actual saturated basic Rankine cycle in the T–S diagram. State 1a is the actual exit state of the pump, while 3a is the actual exit state of the turbo expander and 1s and 3s are the corresponding states for the isentropic cases.

Heat transfer and power in each component of the cycle are calculated by applying the first law of thermodynamics on them.

Referring to Fig. 1A, the following equations can be generated.

- Waste Heat Oil Heat Exchanger (HE-001).

$$q_{HE001} = h_5 - h_6 \tag{3}$$

- Thermal Oil Pump (P-001).

$$p_{P001} = h_6 - h_7 \tag{4}$$

- Evaporator (HE-002).

$$q_{HE002} = h_2 - h_{1a} \tag{5}$$

- Turbo Expander (K-100).

$$p_t = h_2 - h_{3a} \tag{6}$$

- Air Cooler (AC-001).

$$q_{co} = h_{3a} - h_4 \tag{7}$$

- Working Fluid Pump (P-002).

$$p_{P002} = h_{1a} - h_4 \tag{8}$$

where

q_{HE001}-Absolute values of specific heat in Waste Heat Oil Heat Exchanger (HE-001).
p_{P001}-Absolute values of pump specific work (P-001).
q_{eva}-Absolute values of specific heat in evaporator (HE-002).
w_t-Absolute values of turbo expander specific work (K-100).
q_{co}-Absolute values of specific heat in Air cooler (AC001).
p_{P002}-Absolute values of pump specific work (P-002).

3.2. Regenerative cycle

Fig. 2B shows a general representation of the actual saturated regenerative Rankine cycle in the T–S diagram. States 1a is the actual exit state for the pump, while 3a is the actual exit state for the turbo expander.

Heat transfer and power in each component of the cycle are calculated by applying the first law of thermodynamics on them. Referring to Fig. 1B, the following equations were generated:

- Waste Heat Oil Heat Exchanger (HE-001).

$$q_{HE001} = h_5 - h_6 \tag{9}$$

- Thermal Oil Pump (P-001).

$$p_{P001} = h_6 - h_7 \tag{10}$$

- Evaporator (HE-002).

$$q_{eva} = h_2 - h_{1a} \tag{11}$$

- Regenerator (HE-003).

$$q_{reg} = h_{3a} - h_8 \tag{12}$$

- Turbo Expander (K-100).

$$p_t = h_2 - h_{3a} \tag{13}$$

- Air Cooler (AC-001).

$$q_{co} = h_8 - h_4 \tag{14}$$

- Working Fluid Pump (P-002).

$$p_{P002} = h_9 - h_4 \tag{15}$$

where

q_{HE001}: absolute values of specific heat in Waste Heat Oil Heat Exchanger (HE-001).
p_{P001}: absolute values of pump specific work (P-001).
q_{eva}: absolute values of specific heat in evaporator (HE-002).
q_{reg}: absolute values of specific heat in regenerator (HE-003).
p_t: absolute values of turbo expander specific work (K-100).
q_{co}: absolute values of specific heat in Air cooler (AC001).
p_{P002}: absolute values of pump specific work (P-002).
h_{1a}: specific enthalpy for stream (working fluid in).
h_2: specific enthalpy for stream (turbo expander inlet).
h_{3a}: specific enthalpy for stream (turbo expander outlet).
h_4: specific enthalpy for stream (cooler outlet).
h_5: specific enthalpy for stream (thermal oil out).
h_6: specific enthalpy for stream (thermal oil in).
h_7: specific enthalpy for stream (cold thermal oil).
h_8: specific enthalpy for stream (cooler inlet).
h_9: specific enthalpy for stream (pumped working fluid).

3.3. Parameters of comparison

The following parameters and equations were used to compare between the working fluids under evaluation (Table 3)

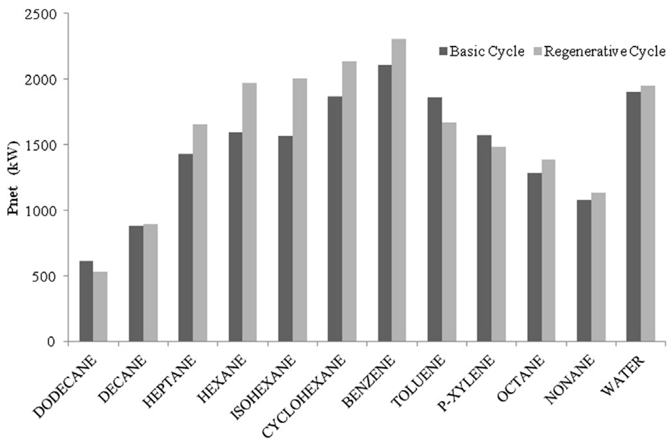

Fig. 3. The net work output for different working fluids in the two cycles.

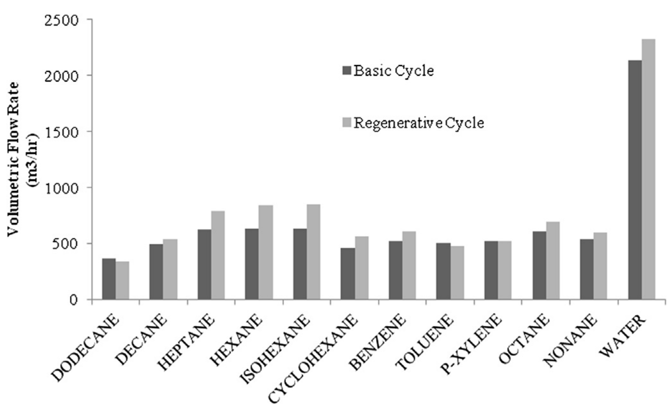

Fig. 4. The volumetric flow rate for different working fluids in the two cycles.

and between the two cycles (Basic and Regenerative) under investigation.

a) The net Power output (P_{net}):

P_{net} = turbo expander power − working fluid pump power − Thermal Oil Pump power − air cooler power.

b) Efficiency (η_{th}):

The factor which is always the center of attention among different factors in a Rankine cycle is the efficiency or the first law efficiency of the cycle. For a specific working fluid and particular amount of input heat rate, the higher the thermal efficiency the higher the net power output. Through the comparison between different working fluids in the Rankine cycle, the net power output should be considered along with the thermal efficiency.

The efficiency is calculated as the ratio of the net power output to the total rate of heat gained from the exhaust gas.

$$\eta_{th} = P_{net}/Q^{\circ}_{in} \tag{16}$$

where Q°_{in} is the total rate of heat gained from the exhaust gas.

$$Q^{\circ}_{in} = m^{\circ}(h_{Ei} - h_{Eo}) \tag{17}$$

where

Q_{in}: The heat gained from the exhaust gas, kW
m°: The mass flow rate for the exhaust gas. kg/s
h_{Ei}: specific enthalpy for the exhaust gas in, kJ/kg
h_{Eo}: specific enthalpy for the exhaust gas out, kJ/kg

c) Irreversibility:

The irreversibility rate for a cycle in steady state steady flow condition can be expressed as [16]:

$$I_{tot} = T_0 m^{\circ}\left[-\frac{h_2 - h_1}{T_H} - \frac{h_4 - h_3}{T_L}\right] \tag{18}$$

where

I_{tot}: Irreversibility, kW
T_0: Surrounding Temperature, K
m°: Fluid Mass Flow Rate, kg/s
h_1 : Mass enthalpy for the stream (working fluid in), kJ/kg
h_2: Mass enthalpy for the stream (turbo expander inlet), kJ/kg
h_3: Mass enthalpy for the stream (turbo expander outlet/cooler inlet), kJ/kg
h_4: Mass enthalpy for the stream (cooler outlet), kJ/kg
T_H: Heat source temperature, K
T_L: Cooling medium temperature, K

d) Volumetric Flow Rate:

The volumetric flow rate is the volume of fluid which passes through the turbo expander per unit time. This is a very important parameter since it affects the turbo expander size and, as a consequence, the cost of the system.

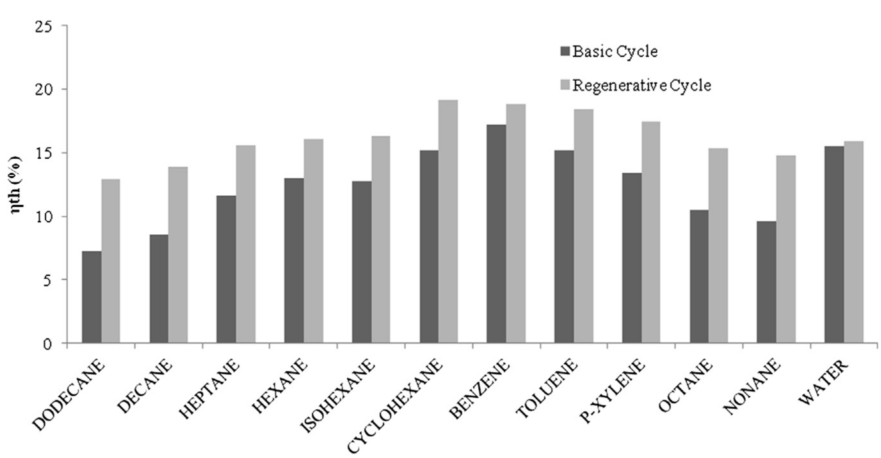

Fig. 5. The cycle efficiency for different working fluids in the two cycles.

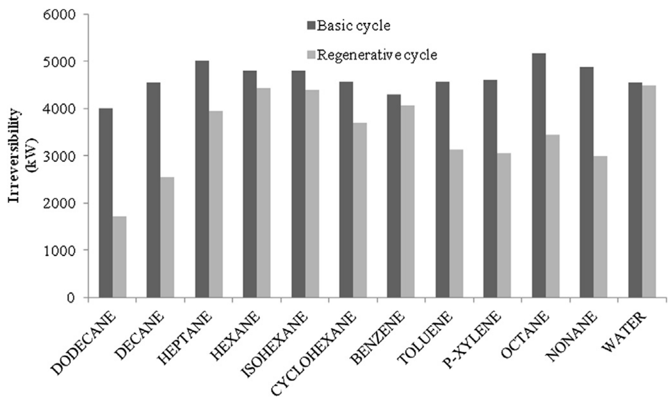

Fig. 6. The Irreversibility for different working fluids in the two cycles.

Table 4
Capital cost of benzene regenerative cycle and cyclohexane regenerative cycle.

Fluid	Capital cost, $
Benzene	2,614,400
Cyclohexane	2,432,000

3.4. *Capital cost and profitability*

Capital cost and profitability study has been carried out on the best two working fluids to be confident about the best working fluid. The capital cost estimation has been carried out using Aspen Economic Analyzer software V 7.1. Aspen Economic Analyzer software uses different input data such as pressure, temperature and flow rate of each equipment to preliminary size it and, consequently, determining its capital cost from its database. The following parameters have been used to economically compare between the best two working fluids:

a) Rate of Return

Rate of return on investment is ordinarily expressed on an annual percentage basis. The yearly profit divided by the total initial investment necessary multiplied by 100 represents the standard percent return on investment [22].

$$\text{Rate of Return} = ((\text{Return} - \text{Capital})/\text{Capital}) \times 100\% \quad (19)$$

b) Net Present Value

Net cash flows and net benefits are important for the project developers. NPV (Net Present Value), method is a powerful indicator of the viability of the projects and can be determined from the following equation [23]:

$$\text{NPV} = \sum_{i=1}^{n} (B - C)_i a_i \quad (20)$$

where NPV is the net present value, B is the benefit, C is the cost and “a” is the discount rate.

The discount rate, a, can be calculated as:

$$a = \frac{1}{(1+i)^p} \quad (21)$$

where i is the interest rate and p is the period.

4. Results and discussion

4.1. *The net power output*

Fig. 3 shows the P_{net} resulting from simulation of the 11 organic fluids comparing to water for both the basic and regenerative. Comparing the results in Fig. 3 with the physical properties of the

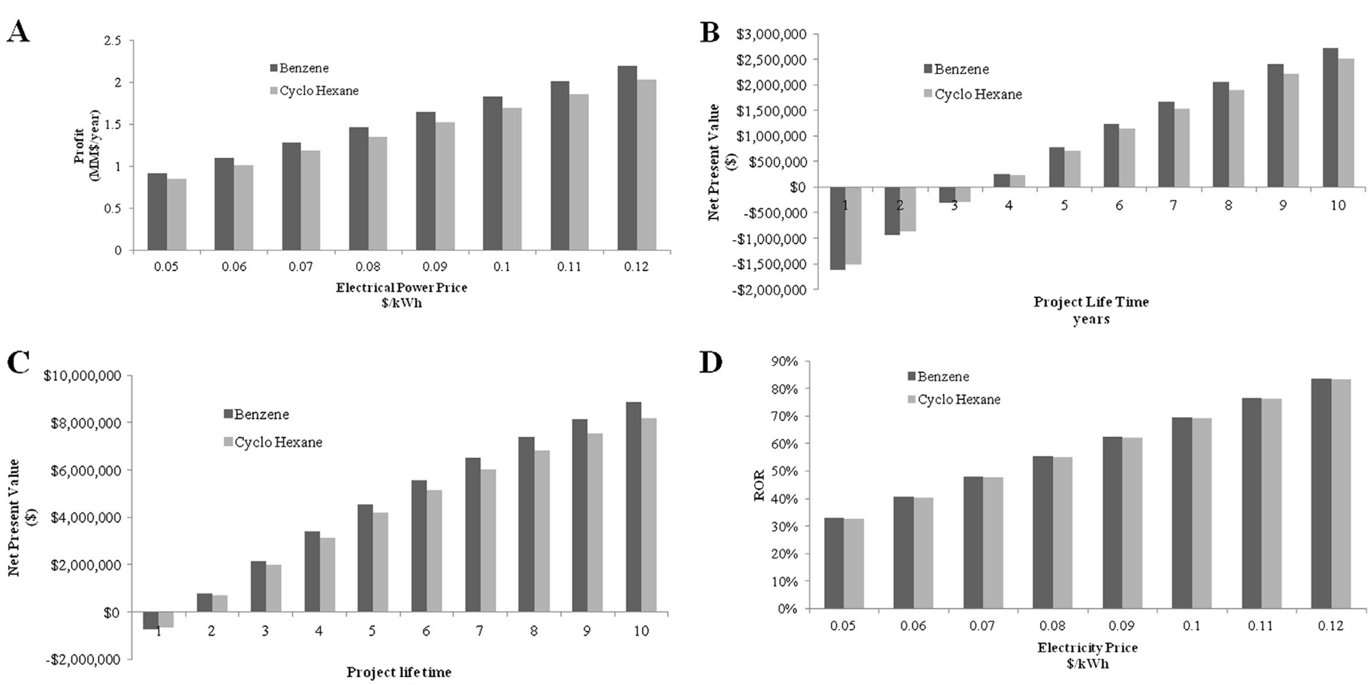

Fig. 7. A-Profit for each working fluid against electrical power price; B-Net Present Value for each working fluid based on electrical power price of 0.05 $/kWh; C-Net Present Value for each working fluid based on electrical power price 0.11 $/kWh; D-Rate of Return for 10 years for each working fluid against electricity price.

working fluids under investigation shows that P_{net} is dependable on the critical pressure of the working fluid. As the critical pressure increases, the differential pressure across the turbo expander increases. As a consequence, P_{net} produced increases. In addition, the critical temperature has an impact on the P_{net} produced since the critical temperature of the working fluid should not exceed the exhaust gas inlet temperature in order to make use of all the available energy.

The only exception for this is water; where superheating is required in order to prevent water condensation inside the turbo expander. The superheating causes the turbo expander inlet pressure to be about 2.1 MPa.

In addition, from Fig. 3, the net power output for the regenerative cycle is mostly greater than that for the basic cycle. These results comply with previous studies in the literature [16]. The net power output for decane, dodecane, nonane, toluene, p-xylene and water are less in the regenerative cycle than in the basic cycle. This can be explained in view of that the exhaust gas out temperature has been increased more than 150 °C in order to keep the minimum approach in the Waste Heat Oil Heat Exchanger (HE-001) $\geq$ 10 °C. As a consequence, the waste heat recovered has decreased and its effect comes over the effect of using the Regenerator (HE-003). Therefore, the net work output has decreased. For cyclo-hexane, heptane and octane, the effect of using the regenerative cycle is more than the effect of decreasing in the waste heat recovered.

4.2. Volumetric flow rate

For the design of the Rankine cycle, an important factor is the volumetric flow rate at the inlet of the turbo expander. This factor determines the design and cost of the turbo expander, which account for almost 30% of the whole system (11) [24]. So fluids with a low value of volume flow rate are required.

As indicated in Fig. 4, the volumetric flow rate for each organic fluid is much less than that for water. This will result in the decrease of the size of the turbo expander in case of using the organic fluid compared to that required in case of using water. Consequently, the capital cost in case of using organic fluids is less than that in the case of using water.

4.3. Efficiency

From Fig. 5, the efficiency of the regenerative cycle is higher than that of the basic cycle. This is expected since the heat removed by the air cooler is reduced by using the Regenerator (HE-003). These results are in line with previous studies done by others [16].

4.4. Irreversibility

The irreversibility for all working fluids is related to the efficiency of the cycle. This means that if the improvement in the efficiency due to the regenerative cycle is large, this will be reflected and the irreversibility will be low and vice versa. This is obvious in Fig. 6, where the irreversibility of the regenerative cycle is lower than that of the basic cycle.

From the previous results it is evident that:

1) The Regenerative Cycle is the most suitable cycle for the case since:
   - It produces higher P_{net} than the basic cycle.
   - Its efficiency is higher than that of the basic cycle.
   - Its irreversibility is lower than that of the basic cycle.
2) Benzene and cyclohexane are the most promising working fluids to be used.

Complete details of the Aspen HYSYS output for the optimized regenerative cycle with benzene as working fluid is represented in Supplementary Materials.

4.5. Capital cost and profitability analysis

In order to be more confident about the best working fluid, a capital cost and profitability analysis has been carried out on the best two working fluids (benzene and cyclohexane) for the regenerative cycles based on the previous studied parameters. The capital cost and profitability analysis has been conducted using Aspen Process Economic Analyzer V. 7.1, integrated with the process simulator. Table 4 summarizes the cost estimated for the regenerative cycle while using either benzene or cyclohexane (the most promising fluids). The analysis shows that the cost of cycle using benzene is slightly higher than that using cyclohexane.

Fig. 7A–D summarize the profit, the net present value (NPV) for electrical power price 0.05 $/kWh or 0.11 $/kWh, and the rate of

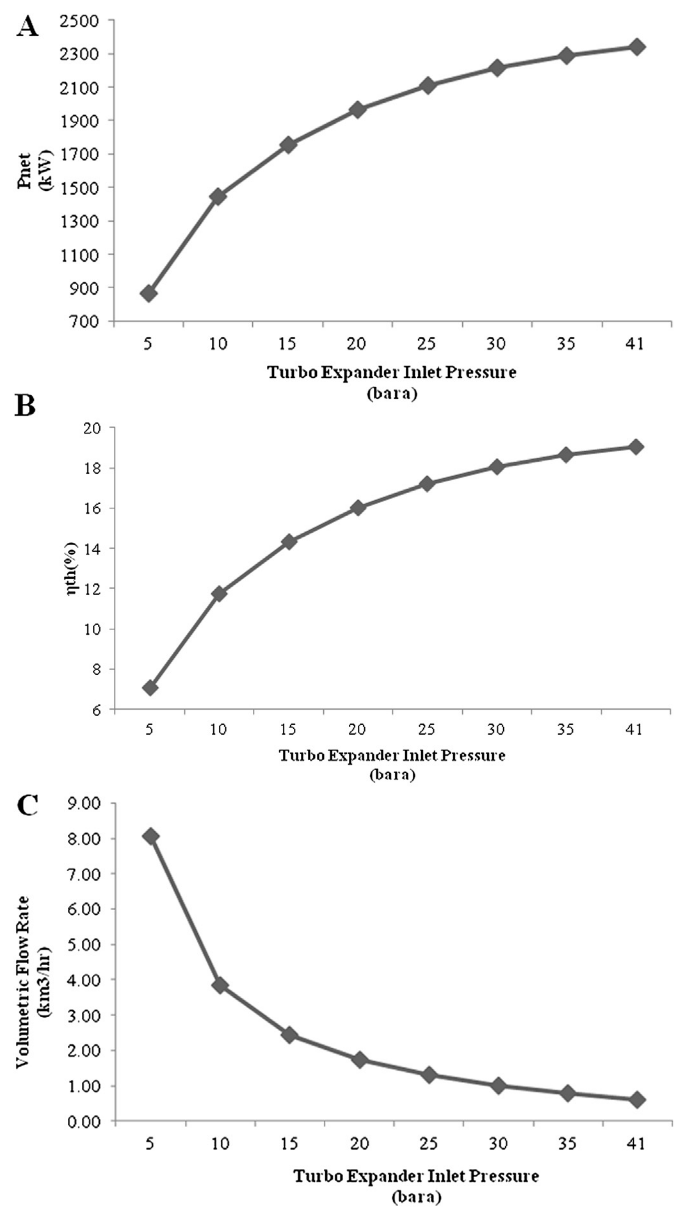

Fig. 8. A-Net power produced at different Turbo Expander inlet pressure; B-Cycle efficiency at different Turbo Expander inlet pressures; C-Volumetric flow rate at different Turbo Expander inlet pressure.

return (RoR) against the electrical power price. From the figures, the waste heat recovery system using organic Rankine cycle is a profitable project. Moreover, benzene can be considered as the best working fluid suitable for the case study.

4.6. Cycle optimization

The parametric analysis of the most two pronounced parameters on the economics of the ORC has been studied. These parameters are:

- Turbo Expander inlet pressure
- Turbo Expander inlet temperature at the maximum working pressure

4.6.1. Turbo expander inlet pressure

From Fig. 8A– C, the net power produced and the cycle efficiency increase with increasing the expansion ratio (the ratio between the inlet and the outlet pressures). On the contrary of the previous, the volumetric flow rate decreases with increasing the turbo expander inlet pressure. In addition, an inlet pressure of 4.1 MPa is the most suitable for the case under study. This conclusion has been approved by conducting a capital cost and profitability as shown in Fig. 9A–D. These figures present the capital cost of the regenerative cycle, the capital cost per each kW produced and the net present value for 10 years project life time, respectively at different turbo expander inlet pressures. Fig 9D presents the net present value for different turbo expander inlet pressures through project life time with electricity price of 0.05 $/kWh.

4.6.2. Turbo expander inlet temperature

From Fig. 10A, the net power produced slightly increases with superheating the working fluid up to 290 °C. This could be understood in view of increasing the working fluid temperature at the inlet of the evaporator. This leads to reducing the heat recovered from the exhaust gas. From Fig. 10B, the volumetric flow rate increases due to the superheating but actually the mass flow rate is decreasing. Fig. 11A–C presents the capital cost of the regenerative

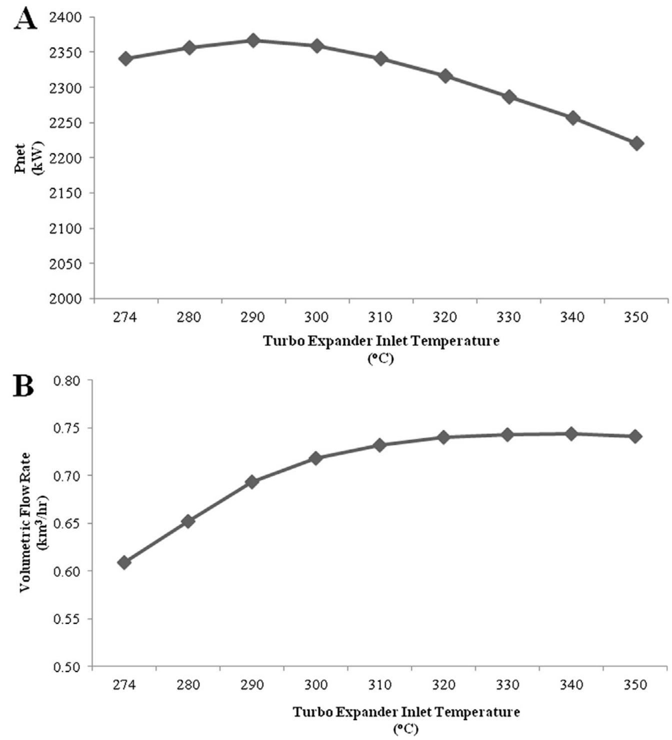

Fig. 10. A-Net power produced at different Turbo Expander inlet temperature; B-Flow rate at different Turbo Expander inlet temperatures.

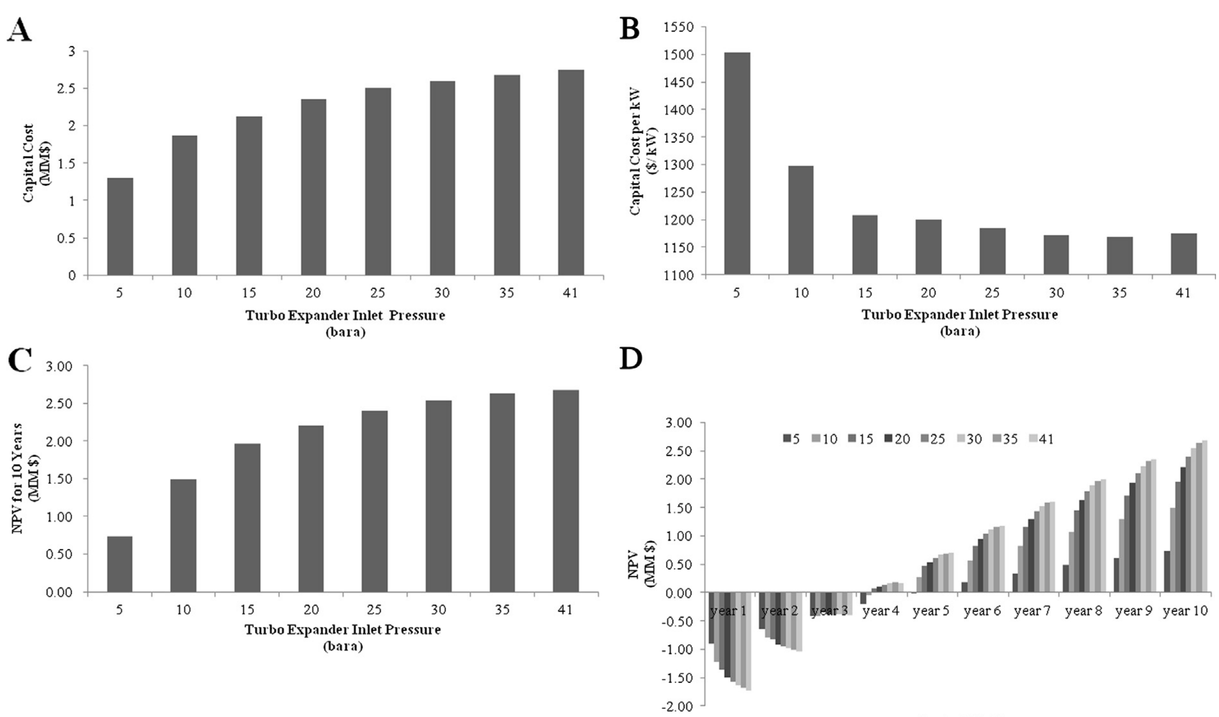

Fig. 9. A-Capital cost at different Turbo Expander inlet pressures; B-Capital Cost per kW at different Turbo Expander inlet pressures; C-Net present Value for 10 years at different Turbo Expander inlet pressures with electricity price 0.05 $/kWh; D-Net present Value for different Turbo Expander inlet pressures through project life time with electricity price of 0.05 $/kWh.

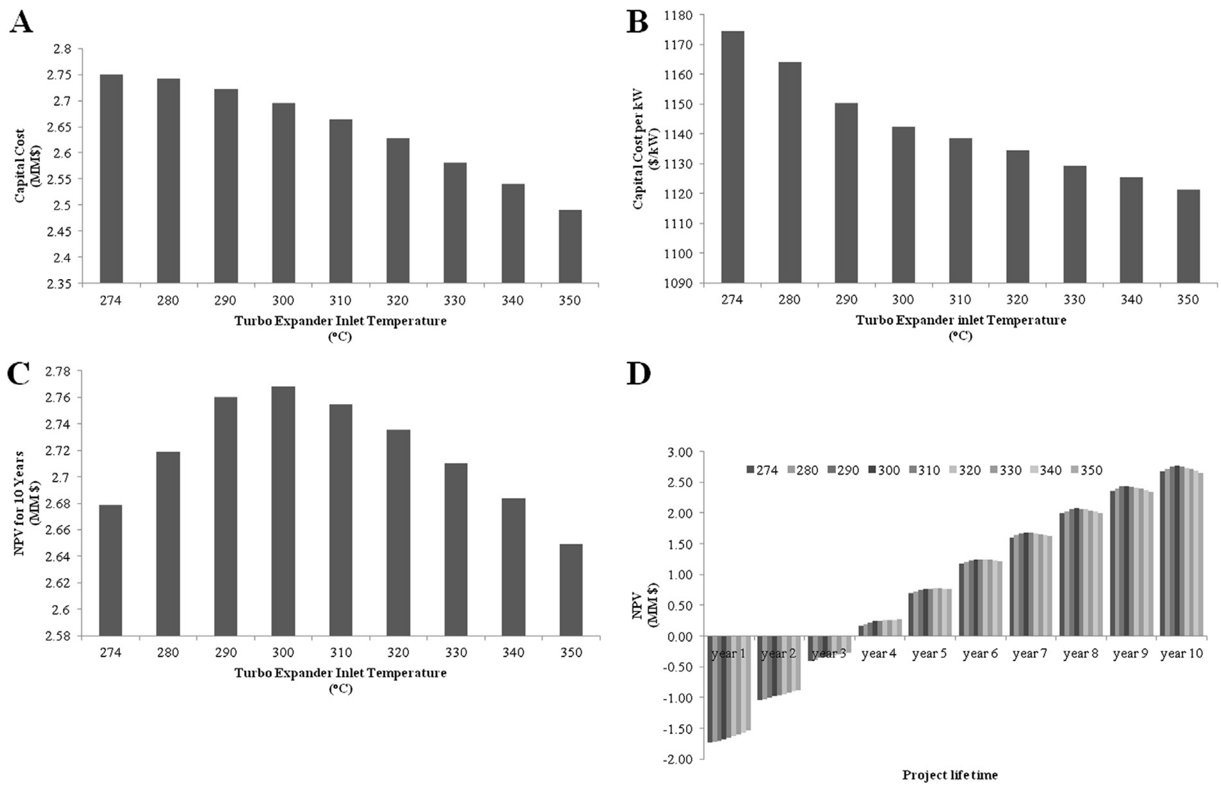

Fig. 11. A-Capital Cost at different Turbo Expander inlet temperatures; B-Capital Cost per kW at different Turbo Expander inlet temperatures; C-Net present Value for 10 years at different Turbo Expander inlet temperature with electricity price 0.05 $/kWh; D-Net present value for different Turbo Expander inlet temperatures through project life time with electricity price of 0.05 $/kWh.

cycle, the capital cost per each kW produced and the net present value for 10 years project life time respectively at different turbo expander inlet temperatures. Fig 11D presents the net present value for different turbo expander inlet temperatures through project life time with electricity price of 0.05 $/kWh. From Fig. 11C and D, it is concluded that the net present value of the project for 10 years life time is slightly higher at 300 °C compared to 290 °C. This leads to that the most suitable turbo expander inlet temperature is between 290 °C and 300 °C.

5. Conclusions

Thermodynamic and economical analyses were performed to utilize the waste heat disposed from an existing Egyptian gas treatment plant using organic Rankine cycle. Two different cycles (Basic and Regenerative) were simulated using Aspen HYSYS v 7.1. Various working fluids have been studied. The study compared different parameters such as net power produced, efficiency, volumetric flow rate and irreversibility. From the study, it is concluded that the regenerative cycle is the most suitable cycle for the case study using either benzene or cyclohexane.

In addition, a capital cost and profitability analysis was performed for the best two working fluids (benzene and cyclohexane) resulted from the previous results compare between these two fluids. The analysis showed that benzene is slightly better as working fluid.

Finally, optimization study using economical indicators was conducted to determine the most suitable working condition. The optimization study results showed that turbo expander inlet pressure of 4.1 MPa and temperature between 290 °C and 300 °C are the most appropriate working conditions for the case under study. The waste heat recovery using the organic Rankine cycle is a profitable process. The study shows that the payback period will not exceed 4 years. It is recommended that similar approach can be carried on other industries in order to contribute to energy saving and optimization of the Egypt industry.

Notations

a	discount rate
B	benefit
C	cost
Ex	exergy
h	mass specific enthalpy [kJ/kg]
i	the interest rate
I	irreversibility
m°	mass flow rate [kg/s]
p	mass specific work
p	the period
P	power
q	specific heat [kJ/kg]
Q°	heat transfer rate
S_{gen}	entropy generation rate [kW/K]
T	temperature [K]
T_0	the surrounding temperature [K]

Latin

η	efficiency

Abbreviations

NPV	net present value
ORC	organic Rankine cycle
ODP	ozone depletion potential
GWP	global worming potential
ROR	rate of return

Subscripts

abs	absolute
co	air cooler (AC001)
cr	critical
Ei	the exhaust gas in
Eo	the exhaust gas out
eva	evaporator (HE-002)
H	heat source
HE001	Waste Heat Oil Heat Exchanger (HE-001)
in	incoming
j	*j*th component of the cycle
L	cooling medium
out	outgoing
P001	pump (P-001)
P002	pump (P-002)
reg	regenerator (HE-003)
t	turbo expander (K-100)
th	thermal
tot	total
1	stream (working fluid in)
1a	stream (working fluid in)
2	stream (turbo expander inlet)
3	stream (turbo expander outlet/cooler inlet)
3a	stream (turbo expander outlet)
4	stream (cooler outlet)
5	stream (thermal oil out)
6	stream (thermal oil in)
7	stream (cold thermal oil)
8	stream (cooler inlet)
9	stream (pumped working fluid)

Appendix A. Supplementary data

Supplementary data related to this article can be found at http://dx.doi.org/10.1016/j.energy.2013.11.011.

References

[1] Sham DK. Analysis of exhaust waste heat recovery techniques from stationary power generation engines using organic Rankine cycles. M.S. Dissertation. USA: Mississippi State University; 2008.

[2] Srinivasan KK, Mago PJ, Krishnan SR. Analysis of exhaust waste heat recovery from a dual fuel low temperature combustion engine using an Organic Rankine Cycle. Energy 2010;35:2387–99.

[3] Roos CJ. An overview of industrial waste heat recovery technologies for moderate temperatures less than 1000°F [Online], www.chpcenternw.org/NwChpDocs/AnOverviewOfIndustrialWasteHeatRecoveryTechForModerateTemps.pdf; 2009 [last access: March 2013].

[4] BCS Inc.. Waste heat recovery: technology and opportunities in U.S. Industry [last access: March 2013], www.gulfcoastcleanenergy.org/Portals/24/Reports_studies/WHR_DOE%20ITP.pdf; 2008 [Online].

[5] Wang ZQ, Zhou NJ, Guo J, Wang XY. Fluid selection and parametric optimization of organic Rankine cycle using low temperature waste heat. Energy 2012;40:107–15.

[6] Mago PJ, Chamra LM, Srinivasan K, Somayaji C. An examination of regenerative organic Rankine cycles using dry fluids. App Therm Eng 2008;28:998–1007.

[7] He C, Liu C, Gao H, Xie H, Li Y, Wu S, et al. The optimal evaporation temperature and working fluids for subcritical organic Rankine cycle. Energy 2012;38:136–43.

[8] Desai NB, Bandyopadhyay S. Process integration of organic Rankine cycle. Energy 2009;34:1674–86.

[9] Vélez F, Segovia JJ, Martín MC, Antolín G, Chejne F, Quijano A. A technical, economical and market review of organic Rankine cycles for the conversion of low-grade heat for power generation. Renew Sustain Energy Rev 2012;16:4175–89.

[10] Badr O, Probert SD, O'Callaghan PW. Selecting a working fluid for a Rankine cycle. Appl Energy 1985;21:1–42.

[11] Papadopoulos AI, Stijepovic M, Linke P. On the systematic design and selection of optimal working fluids for Organic Rankine Cycles. App Therm Eng 2010;30:760–9.

[12] Tchanche BF, Papadakis G, Lambrinos G, Frangoudakis A. Fluid selection for a low-temperature solar organic Rankine cycle. App Therm Eng 2009;29:2468–76.

[13] Bao J, Zhao L. A review of working fluid and expander selections for organic Rankine cycle. Renew Sustain Energy Rev 2013;24:325–42.

[14] Wang J, Yan Z, Wang M, Ma S, Dai Y. Thermodynamic analysis and optimization of an (organic Rankine cycle) ORC using low grade heat source. Energy 2013;49:356–65.

[15] Wang EH, Zhang HG, Fan BY, Ouyang MG, Zhao Y, Mu QH. Study of working fluid selection of organic Rankine cycle (ORC) for engine waste heat recovery. Energy 2011;36:3406–18.

[16] Rayegan R, Tao YX. A procedure to select working fluids for solar organic Rankine cycles (ORCs). Renew Energy 2011;36:659–70.

[17] Lai NA, Wendland M, Fischer J. Working fluids for high-temperature organic Rankine cycles. Energy 2011;36:199–211.

[18] Saleh B, Koglbauer G, Wendland M, Fischer J. Working fluids for low-temperature organic Rankine cycles. Energy 2007;3:1210–21.

[19] Liu C, He C, Gao H, Xie H, Li Y, Wu S, et al. The environmental impact of organic Rankine cycle for waste heat recovery through life-cycle assessment. Energy 2013;56:144–54.

[20] Drescher U, Bruggemann D. Fluid selection for the organic Rankine cycle (ORC) in biomass power and heat plants. App Therm Eng 2007;27:223–8.

[21] Delgadeo-Torres AM, Garcia-Rodriguez L. Preliminary assessment of solar organic Rankine cycles for driving a desalination system. Desalination 2007;216:252–75.

[22] Peters MS, Timmerhaus KD. Plant design an economics for chemical engineers. 4th ed. New York, USA: McGraw-Hill; 1991.

[23] Pulat E, Etemoglu AB, Can M. Waste-heat recovery potential in Turkish textile industry: case study for city of Bursa. Renew Sustain Energy Rev 2009;13:663–72.

[24] Wang XD, Zhao L. Analysis of azeotropic mixtures used in low-temperature solar Rankine cycles for power generation. Sol Energy 2009;83:605–13.